2009

Engineering Report

Team 9: Clipper Windpower Retractable Wind Turbine Blade

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Summary

This design project is partnered with Clipper Windpower, a local wind energy company that manufactures wind turbines for both land and sea installations all around the world. Harnessing the wind can be an eco-friendly way to generate electricity, but it is very dependent upon local weather conditions. When wind speeds drop below optimal levels, the ability of a wind turbine to produce electricity drops sharply, which is the problem Clipper asked us to help solve. A successful solution to this problem would improve the power output of the turbine when the wind is slow without compromising other aspects of their existing design.

The amount of electrical power generated by a wind turbine at a given wind speed is directly proportional to the square of the blade length, so the way to increase the power output at low wind speeds is to increase the length of the blades. The problem with simply employing longer blades is that a longer blade creates more lift at all wind speeds and the extra lift becomes a safety issue for the structure during high winds. Longer blades create much greater bending stresses concentrated at the hub of the turbine, which reduce the lifetime of the blade and raise the risk of catastrophic failure of the whole turbine. At the moment, wind turbine designers must choose between low power output at low wind speeds or high stresses at high wind speeds.

Our group has identified and designed a solution which will allow for higher power output in low wind conditions while preventing the high stresses at high wind speeds. This is accomplished with a tip extension blade that moves in and out of the tip of the original blade. The new design will enable the turbine to produce the maximum amount of electricity in lower wind conditions without having to compromise the structural integrity of the turbine.

The tip extension will operate by extending outward during low wind conditions to increase the length of the blade and consequently increasing the amount of electricity generated. When the wind speed is high enough for the turbine to reach maximum power output without the extra blade length, the tip extension will retract into the main blade to reduce stresses applied to the structure.

The development of this design concept took considerable effort. Thus far, our team has performed extensive technical and design analysis, performed fiberglass bonding and rope stretch tests and analysis, constructed computer models, performed FEA analysis, and constructed a proof-of-concept model. These efforts are explained in great detail in the body of this report. The results of this work have brought us close to demonstrating design feasibility with the exception of providing physical proof of the viability of utilizing rope in our design as well as proving through FEA analysis that an altered structural system will support wind loads. If these results are favorable, design feasibility will be established.

To complete this project, the following action items have been identified as critical to project completion and success. Fray testing of rope is scheduled to determine wear characteristics in order to prove that it can be used in our design. FEA analysis on the alternative structure must be finalized to ensure its strength and structural soundness. Finally, a scaled prototype is scheduled to be completed to test and verify the performance metrics identified for this design. Using these performance results to predict how our design will behave on Clipper’s turbine will bring our project to completion.

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1 Introduction

Wind turbines are machines designed to generate electricity by using the wind as a power source. They employ the same principles as airplane propellers but in an opposite fashion. Unlike airplane propellers which use a motor to spin a propeller and generate thrust, wind turbines use thrust from the wind to turn the propeller (or turbine), which turns a motor that generates electricity. Although wind turbines are a great way to harness energy, there are some drawbacks. Most notably, if there is no wind to harness, these turbines cannot produce electricity. When wind speeds are slower than ideal, turbines will produce electricity at a significantly lower efficiency than at ideal wind speeds. There are many companies around the world that produce these turbines, on large and small scales. One company in particular, Clipper Windpower, has been looking into ways to improve this loss in efficiency in their large turbines.

1.1 Purpose

Clipper Windpower approached our team with a project purpose: design a mechanism to improve low speed wind efficiency of their flagship product, the Liberty wind turbine, while keeping high speed wind loads and increased weight to a minimum. Clipper has already designed their turbines to perform at the highest efficiency under optimal wind speeds, but as wind speeds drop, so does the efficiency. When wind speeds increase, the efficiency stays the same because a braking mechanism is employed to prevent the turbine from spinning at an unsafe speed, but the wind loads that bend the turbine blades still increase. This is why Clipper has not used the solution of simply lengthening the blades to solve the efficiency problem. As a team, we were charged with devising a mechanism that allowed the turbine blades to change their shape while the turbine was operating. This mechanism had to employ the concept of increasing the blade's size for low speed winds and decreasing its size for high speed winds.

After extensive research, technical and design considerations, calculations and formation of a decision matrix (**Appendix 4**) during the fall of 2008, a method known as a "tip extension" was agreed upon as the overall solution to this design problem. This tip extension is a smaller secondary blade, housed within each main turbine blade. This tip extension can extend out and retract into the main blade, thereby increasing and decreasing the blade length.

1.2 Scope

Deciding on a design concept for this project allowed for a clear definition of the project scope. Our team will be designing a mechanism that will move a smaller secondary "tip extension" blade in and out of the end of each main turbine blade. This design will be tested and demonstrated on a prototype at 1:2 scale for testing and analysis purposes, with a truncated length of 3 meters. Due to the gigantic size of Clipper's Liberty turbine blades, which are 43.2 meters long, we will work on a smaller scale with a truncation of length[1] to stay within our allocated budget and timeline. Although not required for our project, we also plan to use FEA analysis of our CAD models and calculations from our scaled prototype test results to extrapolate performance characteristics we expect our design to deliver on a full scale.

[1] The purpose for this truncation will be explained in the results section of this report.

1.3 Design Goals

Scaling down our performance characteristics, we plan to use our prototype to prove design feasibility through the following performance characteristics:

- → Maintain the ability to operate the mechanism at a maximum acceleration of 6 g's
- → Limit total mass increase to 50 kg with added mechanism attached
- → Move the "tip extension" blade 1.5 m in 20 sec
- → Lower the minimum wind speed required for full capacity operation from 15 m/s to 12 m/s

These performance characteristics are benchmarked against Clipper's actual Liberty turbine and define the conditions our mechanism will be operating under. When spinning at its maximum speed, the turbine will produce six times the gravitational acceleration on our mechanism, and the design must withstand these loads[2]. The limit of mass increase was specified by Clipper Windpower and is critical to the amount of bending stress added to the main turbine blades. The speed of movement of the tip extension was considered because the extension must deploy in an expedient amount of time. The minimum wind speed requirement was determined by Clipper. Reaching full capacity at this new, slower wind speed will allow the turbine to increase the efficiency of electricity production.

2 Technical Considerations

The technical part of this project needed to consider many of the physical aspects involved with adding a mechanism near the end of a turbine blade. These technical aspects address the project purpose and performance characteristics, such as the increase in efficiency, the forces placed on the mechanism it must overcome to operate, the effects of adding weight to the turbine blades, and the loads placed on the ropes and pulleys. These technical considerations have a heavy influence on the design considerations. These technical considerations helped to more clearly define what challenges this project faces and what it can achieve.

2.1 Increase in Efficiency

Increasing the efficiency for Clipper's Liberty turbine is vital in order to achieve an increase in electrical power output produced by low wind speeds. In order to attain this increase in output, the amount of lift acting on the turbine blades must increase. Enhancing the lift can be accomplished by extending the length of the lifting body or by expanding the chord length of the turbine blade, as pictured in **Figure 2.1**. Lift is governed by the Equation **(2.1)**:

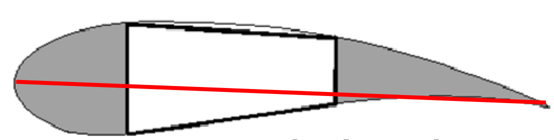


Figure 2.1 – Chord length of a blade

[2]How we determined this acceleration is explained in detail in the technical considerations section of this report.

$$L = q_\infty S \cdot C_L \qquad \textbf{(2.1)}$$

Where:

$q_\infty = \frac{1}{2}\rho_\infty V_\infty^{\ 2}$ = Dynamic Pressure
S = Blade Plainform Area
C_L = Lift Coefficient
L = Resulting Lift
ρ_∞ = Air Density
V_∞ = Wind Velocity

These calculations are based on some key assumptions. It is assumed that the dynamic pressure and lift coefficient for the tip extension are about the same as for the main blade, and that they remain constant. This assumption is validated since the lift coefficient is independent of the size of the airfoil, and is strictly based on the airfoil geometry. Dynamic pressure will also stay constant because it is reliant on air density and wind velocity, which we assume flows across the entire blade at a uniform speed. Since both the tip extension and the main blade should have comparable wing sections, their lift coefficients will be similar.

It was also assumed that a turning turbine blade, with an appropriate twist along its airfoil, can be assumed to act similar to a wing, because the incorporated twist is supposed to account for the increase tangential velocity as position moves towards the tip of the blade.

By increasing the length of the turbine blade, an increase in area is obtained, which increases the total amount of lift force generated by the blade. Similarly, an increase in the chord length of the turbine blade produces an increase in the total area. Through the use of flaps and slats, similar to those used on commercial aircrafts, an increase in chord length can be obtained.

The relationship between the length of the blade and the corresponding lift increase is important in order to correlate them successfully into the power generation equation provided by Clipper Windpower:

$$P = \frac{1}{2}\rho V^3 A C_P \qquad \textbf{(2.2)}$$

Where:

$A = \frac{\pi D^2}{4}$ (Swept Area)
D = Rotor Diameter
ρ = Air Density
V = Wind Speed
C_p = Coefficient of Performance

In order to obtain the desired increase in efficiency of a lifting body at low wind speeds, an increase in swept area will be required assuming constant wind speed, air density, and coefficient of performance. This increased area proves to be problematic when facing high wind speeds since more lift may be produced than the windmill was designed to withstand. Therefore it is very important to find a balance between an increase in efficiency and increased loads on the windmill at various wind speeds. Preliminary calculations can be found in **Table 2.1** below, where highlighted cells illustrate the percent increase in area necessary to achieve maximum power at wind speeds of 12.5 m/s and 12 m/s, as opposed to 15 m/s on Clipper's current design. The final full scale design is a tip extension of 10m, which approximately corresponds to a maximum power output at wind speeds around 12 m/s.

Blade Length (m)	Cp	RPM	Angle of Attack (rad)	Cl
44	0.248418	15.5	0.262	1.645

% Increase in Swept Area	Area (m^2)	Tip Extension Length (m)	Tip Velocity (m/s)	Avg. Tip Velocity (m/s)	P @12.5m/s (W)	P @12 m/s (W)
0	6082.12	0.00	71.42	71.42	1779450.62	1574344.02
5	6386.22	1.09	73.18	72.30	1868423.15	1653061.22
10	6690.33	2.15	74.90	73.16	1957395.68	1731778.43
15	6994.44	3.18	76.59	74.00	2046368.21	1810495.63
20	7298.54	4.20	78.24	74.83	2135340.74	1889212.83
25	7602.65	5.19	79.85	75.63	2224313.27	1967930.03
30	7906.75	6.17	81.43	76.42	2313285.80	2046647.23
35	8210.86	7.12	82.98	77.20	2402258.34	2125364.43
40	8514.97	8.06	84.50	77.96	2491230.87	2204081.63
45	8819.07	8.98	86.00	78.71	2580203.40	2282798.83
50	9123.18	9.89	87.47	79.44	2669175.93	2361516.03
55	9427.28	10.78	88.92	80.17	2758148.46	2440233.24
60	9731.39	11.66	90.34	80.88	2847120.99	2518950.44
65	10035.50	12.52	91.74	81.58	2936093.52	2597667.64
70	10339.60	13.37	93.12	82.27	3025066.05	2676384.84
75	10643.71	14.21	94.48	82.95	3114038.58	2755102.04

Table 2.1 – Results of power calculations for various increases in tip length at two wind speeds

It is important to understand why increasing the area is the best way to raise the power output at lower wind speeds. The coefficient of performance is assumed to be held constant since this term is based upon how much useful mechanical energy is provided to the rotor from the passing wind. In order to increase power by increasing the coefficient of performance a drastic redesign of the entire rotor would be required because ideal rotors would produce zero drag. This would prove to be an extremely difficult, costly path to pursue.

2.2 Force Required to Hold and Move Tip Extension Mechanism

Due to the rotational velocity and the sheer size of the turbine blades that need to be considered, there are large accelerations that must be overcome in order to design a successful tip-extension mechanism. When looking into the forces the tip extension needed to overcome, specs from Clipper Windpower's Liberty Turbine were referenced. These specs include a static blade length of 43.2 meters with a maximum radial velocity of 15 revolutions per minute.

Using these specifications provided to us by Clipper Windpower, the theoretical accelerations and corresponding forces felt by a mass at the tip of the blade could be calculated.

2.2.1 Centripetal Forces

The most significant force that acts on the tip-extension is the centripetal acceleration due to the rotational velocity and the length of the turbine blade. Due to the magnitude of the centripetal acceleration, this is the primary acceleration that the mechanism is required to overcome in order for smooth and safe operation to occur.

The centripetal acceleration is illustrated in **Figure 2.2** and is defined as:

$$a_C = \omega^2 R \tag{2.3}$$

Where:

$$\omega = \text{Radial Velocity } \left({Rad}/{Sec} \right)$$
$$R = \text{Radius of Roatation } (\text{Length})$$

The direction of the centripetal acceleration is always parallel to the radius vector of circular motion and is caused by the mass continually changing directions in order to travel in a circle. Through the use of Equation **(2.3)**, the acceleration that the tip mass will experience can be calculated. It is vital that our design is based on the highest acceleration case. For the Liberty wind turbine, this is at a distance of 43.2 meters with a rotational velocity of 15 rpm on its downswing. In the downswing position, the blade encounters centripetal acceleration as well as gravitational acceleration. In the upswing position, the gravitational acceleration is subtracted from the centripetal acceleration. At any blade position between these two spots, the acceleration will be a value in between the two described maximum accelerations, since gravitational acceleration is always in the downward direction. This means the maximum acceleration will be on the downswing, and the equation for the force will be:

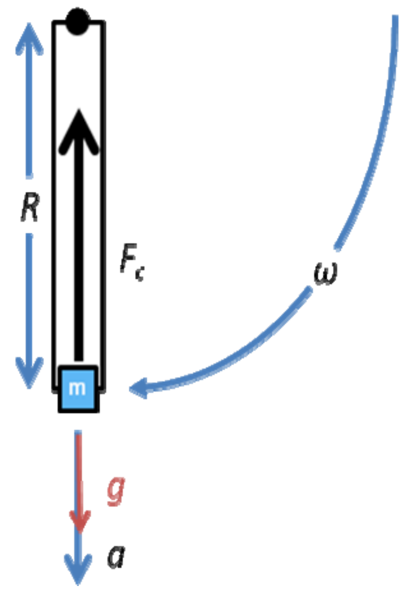


Figure 2.2

$$F_C = (A+1)mg \tag{2.4}$$

Where:

$$A = \frac{a_C}{g} = \text{ Centripetal acceleration as a multiple of } g$$
$$m = \text{Mass of Object}$$
$$g = \text{Gravitational Acceleration}$$

Plugging in the values mentioned earlier into Equations (2.3) and (2.4) a resulting centripetal acceleration can be obtained:

$$a_C = \left(15\,\tfrac{\text{rev}}{\text{min}} \cdot \tfrac{2\pi\text{ rad}}{\text{rev}} \cdot \tfrac{1\text{ min}}{60\text{ sec}}\right)^2 \cdot (43.2\text{ m}) = 106.6\,\tfrac{\text{m}}{\text{s}^2}$$

$$A+1 = \frac{106.6\,\tfrac{\text{m}}{\text{s}^2}}{9.8\,\tfrac{\text{m}}{\text{s}^2}} \approx 12$$

The resulting centripetal acceleration at the downswing of the turbine blade comes out to be approximately 12 times that of gravity. This establishes that in order for the tip extension design to work properly, it must be designed to operate under this acceleration.

2.2.2 Frictional Forces

An additional force that the tip extension blade will experience is that of friction. This force is much less significant than centripetal acceleration; however it still plays an important part in the design. A pictorial description is shown be found in **Figure 2.3**.

The frictional force felt by the blade is defined by the equation:

$$F_f = N \cdot \mu_d \tag{2.5}$$

Where:

N = Normal Force

μ_d = Coefficient of Dynamic Friction

In this case, N is defined as any tangential accelerations produced in addition to gravity. The blade will feel no gravitational acceleration when pointing directly toward and away from level ground and full gravitational acceleration at positions 90° from that. This is due to the normal force always moving in a circular direction while the gravitational force is always fixed in the downward direction. With an approximate μ_d of 0.002 based upon average coefficients of dynamic friction rolling along fiberglass surfaces, the resulting frictional force is going to be insignificant in comparison to the centripetal force, and it can be neglected and accounted for by the factors of safety that are being applied throughout the system.

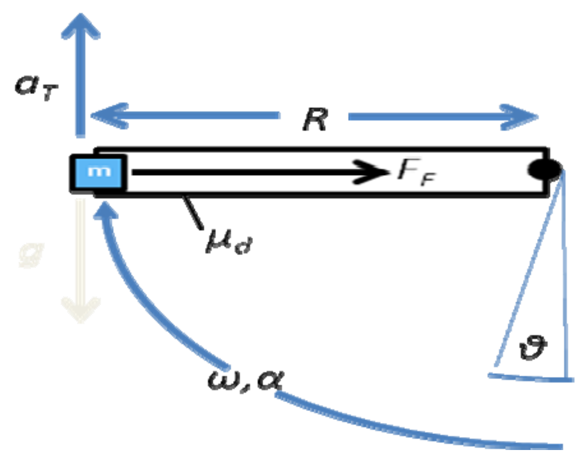


Figure 2.3

2.2.3 Total Force Required to Move Extension

The two forces that the tip extension must overcome are those produced by the centripetal acceleration and the friction between the tip extension rollers and the railing system. The following equation computes the total force that the tip-extension design is required to overcome, which is:

$$F_{\text{Total}} = \left(F_f + F_C\right) \cdot \text{FOS} \tag{2.6}$$

Where:

F_f = Frictional Force

F_C = Centripetal Force

FOS = Factor of Safety

With an approximate tip mass of 600 kg and an acceleration of $106.6\frac{m}{s^2}$, the force that is required for our tip extension to move is 63,960 N or 14,378 lbf. Using a common factor of safety of 1.5 gives a desired force to overcome of 21,567 lbf. For the polymer fiber rope with a force reduction through the proposed pulley system, a factor of safety of 9 is applied to the mechanism in order to reduce the chance of rope failure. The FOS of 9 provides a force of 100,000 lbf that can easily be met through the use of polymer fiber ropes. Further information regarding polymer fiber ropes can be found in **Section 3.4.1** of this report.

2.3 Added Weights and Moments

In order to achieve a successful tip extension mechanism, adding mass to the turbine blade is unavoidable. The addition of any weight to the system will in turn increase stresses throughout the turbine blade. This increase in stress is due to the system behaving as a cantilever beam would. A tip mass on a cantilever beam produces bending moments throughout, which results in stresses. The increases in stress and strain mean that reinforcements may be required to maintain the structural integrity of the blade. The equation for stress within a beam is given by the following equation:

$$\sigma = \frac{M \cdot r}{I} \quad \textbf{(2.7)}$$

Where:

$$M = F \cdot d \ \left(\text{Bending Moment}\right) \quad \textbf{(2.8)}$$

$r =$ Distance to the Outer Surface

$I =$ Second Moment of Area

As can be seen in **Figure 2.4**, the higher stress concentrations will be located along the length of the supporting box beam. This is where potential reinforcements will be located if necessary. Potential moments caused by the additional mass provided by the tip extension mechanism can be found in **Figure 2.5** below.

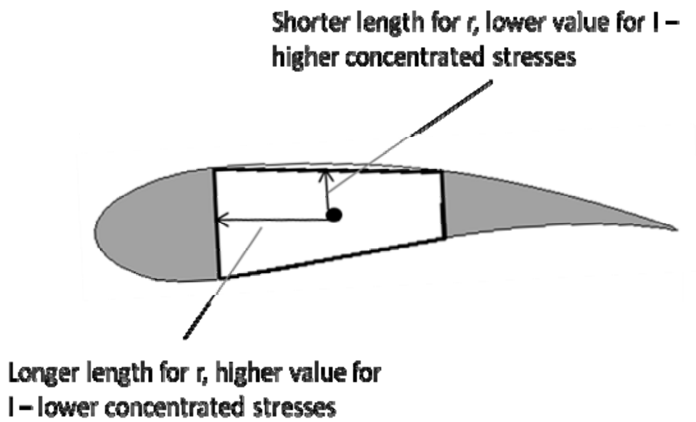
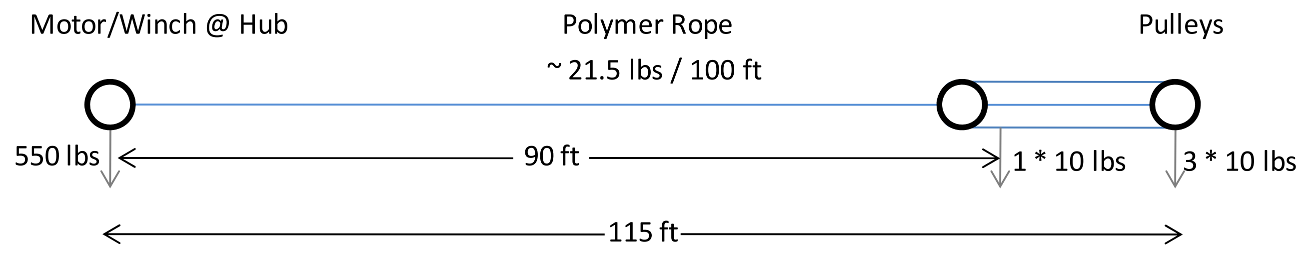


Figure 2.4 - Stress Concentrations in the Box Beam



Types of Polymer Rope Considered

Vectran – 30.1 lbs / 100 ft @ 1 inch diameter (break strength = 105,000 lbs)
Spectra – 23.3 lbs / 100 ft @ 1 inch diameter (break strength = 80,000 lbs)
Dyneema – 21.5 lbs / 100 ft @ 1 inch diameter (break strength = 105,000 lbs)
Technora – 30.1 lbs / 100 ft @ 1 inch diameter (break strength = 125,000 lbs)

Figure 2.5 – Added moment due to additional mass provided from tip extension mechanism.

Plugging in these numbers into Equation **(2.8)**, the resulting moment added to the turbine blade felt at the hub is as follows:

$$M = (90\text{ ft})\left(\frac{21.5\text{ lbs}}{100\text{ ft}}\right)\left(\frac{400}{100}\text{ ft}\right) + (115\text{ ft})(30\text{ lbs}) + (90\text{ ft})(10\text{ lbs})$$

$$M = 12,090\text{ ft}\cdot\text{lbs}$$

With an approximate mass of the tip extension of 600 kg (1,320 lb) taken into consideration, this moment is increased by 151,800 ft·lb. With a box beam cross section of 2 by 1 meters (~6.6 by 3.3 ft), this moment produces an increase in stress corresponding to Equation **(2.7)**:

$$\sigma = \frac{(12,090\text{ ft}\cdot\text{lbs} + 151,800\text{ ft}\cdot\text{lbs})(3,281\text{ ft})}{77.25\text{ ft}^4}$$

$$\sigma = 6960.82\ \tfrac{\text{lbs}}{\text{ft}^2}\ \text{or } 48.34\text{ psi}$$

In conclusion, added weights and corresponding moments and stresses are unavoidable, but in order to achieve a successful design these factors must be minimized.

2.4 Forces Applied to Cable

Because the mechanism will be subjected to acceleration of 12 g-forces, a tip extension having a mass of 600 kg will produce a force of approximately 70,000 N or 15,736 lbs that must be matched by the mechanism to hold and move it. A pulley system, which is described in detail in **Section 3.4.2**, will be used for our design. This system uses ropes to enable the tip extension to move. To reduce the loads put on the cables and pulleys, mechanical advantages were implemented.

Observing **Figure 2.6**, System 3, the load required to move a 100 N weight is only 33.3 N. This is the amount of force applied to the rope. Using a similar design for the tip extension, a load reduction used, reducing the force applied to the rope to only 5,300 lbs. At the apex of the pulley, this force will be doubled because that force is held in opposite directions by the same rope. The concept that allows this to occur is called "force multiplication". This uses the concept that, three parallel ropes pulling up with 5,300 lb will result in a total force of 15,900 lb.

With a working load of 15,736 lbs and, for simplicity of design, a 1" diameter rope with a breaking strength of around 100,000 lb, Equation **(2.9)** was used to calculate the FOS for this section, which was highly critical.

$$\text{Factor of Safety} = \frac{\text{Minimum Breaking Strength}}{\text{Working Load}} \qquad \textbf{(2.9)}$$

$$\text{Factor of Safety} = \frac{100{,}000 \text{ lb}}{10{,}600 \text{ lb}} \approx 9$$

This equation shows that by using a 1" diameter rope, a FOS of 9 is achieved. This allows catastrophic over-spin of the turbine to occur and ensure that the rope will not break due to the loads subjected to it.

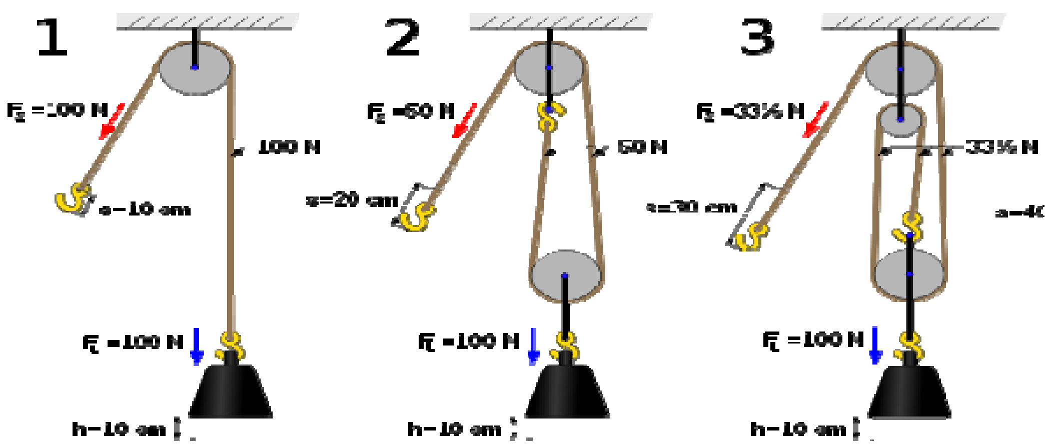


Figure 2.6 - Mechanical Advantage and Force Multiplication in Different Pulley Setups

2.5 Forces Exerted on Pulleys

The method of moving this tip extension, as mentioned in Section 2.4 employs a 3 times mechanical advantage. As can be observed on System 3 of **Figure 2.6**, each rope applies 1/3 of the total force applied, and therefore each pulley must carry 2/3 the total load, since each pulley has 2 effective forces applied to it. This is important to understand because although the pulling force exerted in the figure is 33.3 N, the attachment of the pulley must bear 66.6N. This means that when designing our retraction system, the pulleys will be subjected to greater loads than the rope itself.

3 Design Considerations

Due to the complexity of incorporating this mechanism into an existing design, the design was divided up into four distinct sections, each requiring consideration to incorporate the entire design. The four sections are broken up as follows:

- Tip extension blade
- Special structural modifications to main blade
- Rail attachment and retention structure
- Extension and retraction mechanism

Each of these design aspects is described in detail in the following section. As a whole, however, the design must add less than 50kg to the existing structure on our scale. Other design requirements are included in their respective sections. It is important to understand that the overall design for this project is comprised of these four distinct parts, and proving feasibility of the entire design would not be possible without considering each of these sections. A section on maintenance is also included to illustrate how this design would function on a full scale.

3.1 Tip Extension

Determining the design of the tip extension was crucial because it determines the increase in efficiency of the turbine. The tip extension consists of the airfoil-shaped blade that extends out of and retracts into the main turbine blade as well as forks attached to this blade (refer to **Figure 3.1**). These forks allow movement of the tip extension, give the airfoil structural integrity, and hold the airfoil steady when it experiences wind forces. Important aspects to consider when designing the tip extension were its size, weight, accessibility, and construction material.

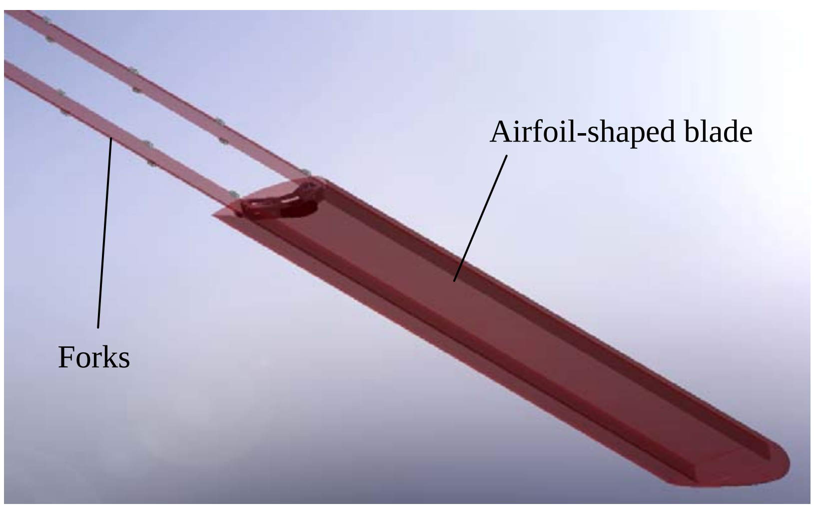


Figure 3.1 – Tip Extension Sub-Assembly with Forks

3.1.1 Size

The size of the tip extension airfoil is what creates an increase in efficiency. As explained in **Section 2.1**, the size of the blade including length, width, and height will determine the amount of lift produced. Making the largest and longest blade possible will produce the greatest amount of lift. Since higher lift increases power, this will increase efficiency. The initial design proposed to fit the tip extension within the box beam of the main turbine. However, fitting the tip extension within this box beam would dramatically reduce the height and width of the tip extension blade in comparison to the main blade, cutting down the amount of lift it can produce. To make up for this lack of width, a considerably longer tip extension blade would have to be used.

The current design plans to alter the interior structure of the main blade to allow the tip extension to have width and height dimensions comparable to the main blade. This means that to produce enough lift for a satisfactory increase in efficiency, the tip extension needs to extend 10 meters beyond the main blade.

3.1.2 Weight

The added weight of the tip extension was important because the project purpose explicitly requires keeping added weight to a minimum. Also, the tip extension is located at the end of the main blade, so any unnecessary weight near the end of the blade puts large bending moments on the main blade (see **Section 2.3**). Increasing the width of the tip extension allows its length and material thickness to be reduced, which lowers weight. An airfoil shell with the two supporting beams (see **Figure 3.1**) will be used for the tip extension. These beams, or forks, extend 10 meters beyond the length of the tip extension and double as compressive structural members and support arms. They provide good strength while minimizing the amount of extra weight added. This gives the tip extension enough strength to maintain its shape, even under high-speed wind loads.

3.1.3 Accessibility

The tip extension needs to be accessible for maintenance purposes. There are four ways to access the tip extension blade once it is installed inside the main blade: crawl through the length of the main blade to get to the tip extension, enter the tip of the main blade, retract the tip extension further inward towards the base of the main blade, or extend the tip extension outward from the base, until it is no longer inside the main blade.

The problem with the first two suggestions is the space available within the blade. Near the end of the main blade, the height of the blade tapers down to 14 inches. Adding the airfoil that takes up most of the remaining available space leaves no room for a human to crawl through to inspect the wing. The third suggestion of retraction would be plausible, but it would require additional track along the entire length of the blade for the tip extension to travel along, as well as resizing the tip extension to fit within the box beam. Both of these requirements would be undesirable since this would add weight and reduce lift, respectively. The final suggestion is the most reasonable one, since removing the tip extension from the main blade would allow inspection and repairs to be easily performed. Also, when maintenance is finished, the blade can simply be inserted back into the tip of the main blade and retracted inwards. Since this method for access is preferable, the design of the supporting tracks must allow for the tip extension to be easily inserted in or extended out of the end of the main blade.

3.1.4 Material

As previously stated, the tip extension weight is important because of the added stress it places on the main blade. Choosing a material for the tip extension that provides enough strength without being too heavy is critical. The main blade is fabricated from fiberglass, which has a good strength to weight ratio. Since Clipper has a good understanding of fiberglass, it was considered as a material for the tip-extension as well. Carbon fiber was also considered because of its high strength to weight ratio. Due to the amount of material required to make a 10 m long tip extension (with an extra 10 m long fork), carbon fiber would be far too expensive to use, giving good reason to choose fiberglass instead.

3.1.5 Final Proposed Tip Extension Design

The full-scale design that was agreed upon, as depicted in **Figure 3.1**, will utilize a 10 meter long tip extension. The shell of the airfoil will be supported by two 20 meter long beams (the fork). The forks will fit within the tip extension shell and extend 10 meters out of the base of the tip extension, toward the hub of the turbine. On the top and bottom edges of the forks, ball transfer rollers will be attached (see **Appendix 13**, **Figure C13.15**), which allows the forks to roll along the supporting track within the main blade. The tip extension shell will be supported by the fork and will have cross-sectional dimensions of 1.22 m chord length and 0.168 m height.

On our prototype scale of 1:2, the cross-sectional dimension of the tip extension shell will be halved to 61 cm chord length and 84 cm height. The total length of the tip extension (fork included) will be 3 meters. The two supporting beams (the fork) will be spaced 0.365 m apart, and both fork beams will have cross-sectional dimensions of 84 cm tall and 15 cm wide. The length was not kept to scale because a full length of 10 meters would be incredibly difficult to work with. Furthermore, truncating the length of the scaled prototype should not hinder proof of feasibility of this design, since performance is determined by the forces it can withstand, what speed the tip extension can move, and how much lift it will produce. The main design requirement that applies to the tip extension is that it must be structurally sound in order to bear a force of 6 g's applied by the retraction pulley.

3.2 Special Modifications to Blade

As mentioned in Tip Extension section, special modifications are needed to be made to the interior structure of the main turbine blade in order to allow enough space for a reasonably sized tip extension, as seen in **Figure 3.2**. This special modification called for removing the last 10 meters of each main turbine blade and replacing the box beam supporting structure with a rib and stringer structure. Adding these special modifications to the blade meant important design considerations needed to be addressed, including the change in weight, change in strength, change in space, and altering of the taper of the main blade height and chord length.

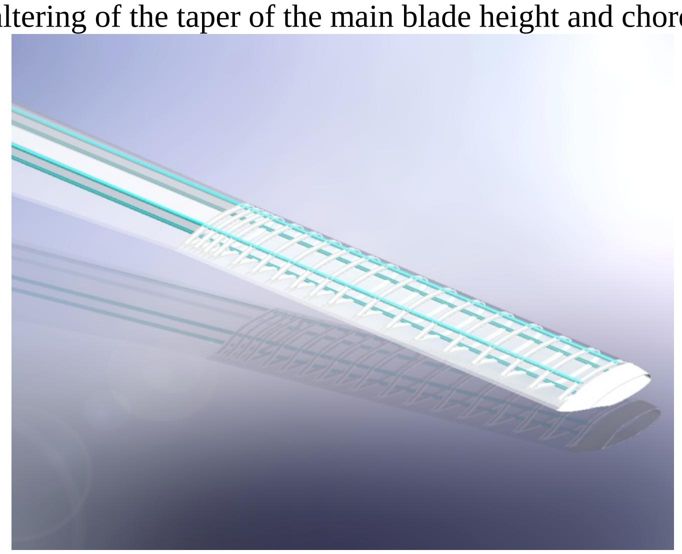


Figure 3.2 – Rib and stringer sub-assembly, pictured white with retention structure pictured turquoise.

3.2.1 Strength of Modifications

Clipper put a considerable amount of design research and effort into ensuring that their turbine blades would flex without breaking. This design proposes to remove a portion of the interior support structure and replace it with another structure, which should be as strong as or stronger than the original design. The design calls for ribs and stringers, a structure similar to the interior of an airplane wing. This is a complicated structure to analyze because the stringers receive support from one another through the ribs as well as through the outside airfoil shell. Simple hand calculations would not be accurate or simple, so analysis using computer Finite Element Analysis needs to be performed. Due to the complexity of modeling such a design in a computer, however, this analysis is still in progress. Using FEA analysis will help determine the strength of the new section and help determine if feasibility for implementation is possible.

3.2.2 Added Weight of Modifications

The increase in weight from this special modification had to be considered as well. Since the design calls for removing the last 10 meters of the box beam and replacing it with a ribs and stringers structure, the total added weight is any weight beyond the weight of the original section of blade that was removed. The existing airfoil shell was also increased in size, which also added weight. Because FEA analysis will be performed, weight analysis of this structure will be computed and compared with the original weight.

3.2.3 Added Space

One of the most important reasons this design calls for a change in the support structure is to open the interior of the main turbine blade so a tip extension of comparable size could fit inside it. The original cross sectional dimensions of the box beam were 0.7 m wide and 0.06 m high at the very tip of the blade. If these dimensions were not modified, the tip extension would need to be much smaller than is currently is proposed. By utilizing the ribs and stringers design rather than a box beam, 0.2 m in height and 0.51 m in width for the tip extension is added to the interior space, allowing more space for the tip extension. The final design increases wing surface area by 56%.

3.2.4 Change in taper and chord length

The chord length and width of the airfoil stop being tapered at this new section. This is to accommodate the tip extension. If the taper of the chord length and width of the main airfoil, the tip extension would be considerably smaller than it already is. It would also increase complexity of the design attempting fit within a tapered shape.

3.2.5 Retention Structure

Some of the space within this modified structure would have to be allocated to a device allowing the tip extension to move. As shown in **Appendix 13, Figure C13.6** below, the ribs will contain slots to allow a rail to be mounted across them, allowing the tip extension to travel through the main blade (see Appendix, Figure xx).

3.2.6 Final Proposed Special Modification Design

The full-scale design that was agreed upon, as depicted in **Figure 3.2**, will utilize the final 10 meters of the turbine blade to locate the ribs and stringers. There will be four stringers that run the 10 meter length of the turbine blade for lateral support and 20 ribs located strategically along the stringers to maintain the airfoil shape. The cross-sectional dimensions of the rib will be 1.6 m wide and 0.28 m tall, and of the stringer, 5.21 cm wide and 3.8 cm tall.

On the prototype scale of 1:2, the cross-sectional dimensions will be halved to 0.8 m wide by 0.14 m tall for the ribs and 5.21 cm wide and 3.8 cm tall for the stringers. Just like the tip-extension, the length will be truncated to an overall length of 3 meters. This modified structure will use half this length, allowing 1.5 meters to the ribs and stringers section and 1.5 meters to the box beam section. The box beam exterior dimensions will be 0.7 m wide by 0.3 m tall, allowing the forks of the tip extension to straddle the box beam when in its retracted state.

3.3 Rail Attachment and Retention Structure

To allow the tip extension to extend and retract within the main blade structure, a rail system needed to be designed. This rail design includes the track that the tip extension forks travel on, the rollers attached to this fork that actually roll along the track, and a fail-safe mechanism to prevent the tip extension from exiting the main blade by accident. Important design considerations included the added weight, placement of the structure, maintenance, and safety. These considerations helped form the final design.

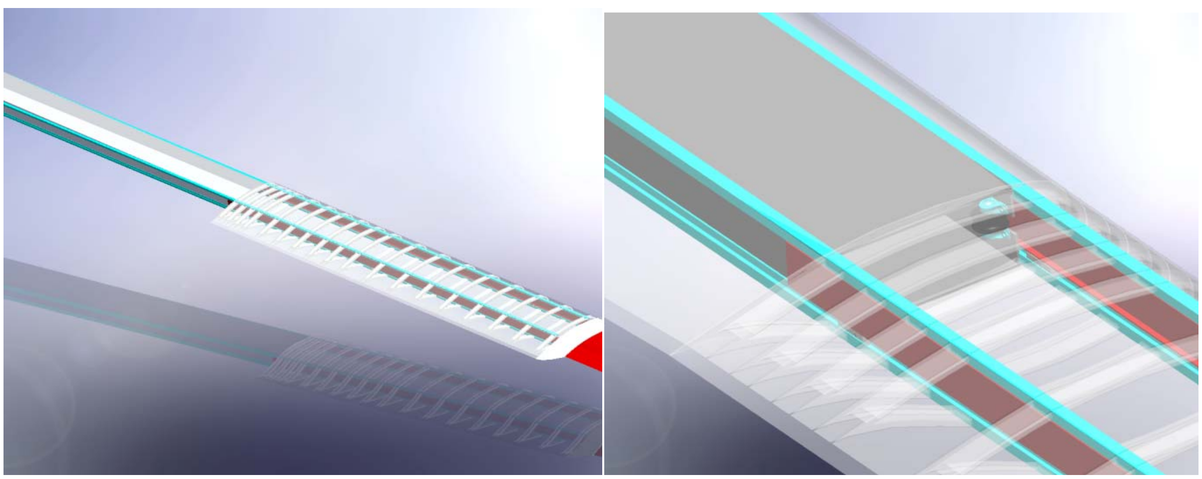


Figure 3.3L – Rail attachment structure attached to outside of box beam.
Figure 3.3R – Close up of rail attachment transitioning from box beam to ribs and stringers.

3.3.1 Added Weight

It was important to keep the weight of this part of the structure as minimal as possible. Since this design is located near the end of the main turbine blade, any substantial increase in weight can have a detrimental effect. The added weight of this rail structure, pictured **Figure 3.3** includes 20 meters of track as well as the safety brakes. These safety brakes, described in detail in **Appendix 9**, are used to ensure the blades cannot move in the unlikely event that the tip extension mechanism fails. The total added weight of this rail system was determined using FEA as approximately 231 lb.

3.3.2 Structure Placement

Deciding where to install this rail structure was critical because its location influences the final size of the tip extension. The initial design called for the rails to be attached to the inside of the box beam, unlike the design in **Figure 3.3L**, where the rails are attached to the outside of the box beam. Implementing this design would be impossible because when the rails extend into the final 10 meter space where the tip extension retracts into, the support for the rails would cause interference with the tip extension blade.

The next suggestion was to mount the rails on the inside of the leading and trailing edges of the main blade. This would be a feasible design, but the space taken up by the rails would force a reduction of the dimensions of the tip extension.

It was then suggested that the tracks be mounted on the outside of the box beam, like in **Figure 3.3L**, with the tip extension contacting the rails near the top and bottom of the airfoil. This suggestion proved to take up the least amount of space, leaving more space for the tip extension. The rails will also be attached to the ribs throughout this section, meaning these rails will also act as stringers to provide more strength within this section.

The rails would be simple C channels (see **Appendix 13, Figure C13.3**) that allow special rollers to travel along them. These rollers, which would be attached to the tip extension, achieve smooth motion through the use of ball bearings. The rollers will have three different planes of contact (see **Appendix 13, Figure C13.6**). This will allow each roller to support the tip extension vertically and in two sideways directions. By mounting roller in two orientations, (see **Appendix 13, Figure C13.7**), movement of the tip extension is restricted to only fore and aft along the track. There are 12 rollers mounted on each fork, allowing for load distribution along the track.

3.3.3 Maintenance

As mentioned in the tip extension section of this article, getting access to the tip extension from within the main turbine blade is difficult. An easier way to access and maintain the rollers would be to stop the turbine and extend the tip extensions out further from the main turbine blades than they normally travel. Details of this maintenance procedure can be found in **Section 3.5**. The tracks the tip extension rolls along will not have any stops on them, allowing the tip extension to roll all the way out. This will allow for easy installation and access, but this poses a safety risk, since a failsafe needs to be present in the event the main mechanism fails.

3.3.4 Safety

As was mentioned, failure of the mechanism holding the tip-extension during operation would be a large safety concern. Although the mechanism that holds the turbine blades will have a FOS of 6, three 600 kg blades flying out of a wind turbine would be incredibly dangerous in the event of failure. To mitigate this problem, a failsafe mechanism must be present.

To address this problem, rod locks, in conjunction with brake pads pushing against the tip extension fork were suggested. Rod locks, which are described in greater detail in **Appendix 9**, are linear actuators that, by default, extend a cylinder outward and lock in place (see **Appendix 9, Figure A9.1**). This means that forces must be applied to disengage the cylinder. The rod locks actuate in an incredibly short amount of time and with a large amount of force. By combining this cylinder with a brake pad, an emergency brake to stop the tip extension can be used.

In this way, if the mechanism holding the tip extensions breaks, this rod lock and brake pad system could be used to safely hold the tip extension within the turbine blade until the turbine can bring itself to a full stop. This breaking system must be mounted in a position where it can apply the brake pads to the wide area of the forks, pushing the forks inward toward the box beam and holding the tip extension in place.

This would require sensors and actuators, as well as a micro-processor to operate. The turbine already incorporates these to measure wind speed and wind loads, as well as actuate the turbine to swing the blades into the wind, feather the blades, and apply the turbine rotor brake. Incorporating this sensing mechanism into the system should not be a big problem, but it does increase the level of complexity.

This failsafe device is incredibly important when actually incorporating this design on a real turbine. Because this project's size, scope, and duration do not allow for considerable research into this failsafe, this part of the design will need to be further researched to ensure operation and functionality as a fail-safe. Although this failsafe will not be fully designed, it will not compromise feasibility for this project's overall design.

3.3.5 Final Proposed Rail Attachment and Retention Structure

The full-scale design that was agreed upon, as depicted in **Figure 3.3L**, will have rails that stretch 20 meters, starting from the tip of the main turbine blade and extending inward. A C-channel rail, mounted on the upper and lower section of the main interior airfoil will be used to allow the rollers to move smoothly. The ribs will have special notches in them to accommodate these tracks avoiding interference with the tip extension blade. The C-channel will have outside dimensions 5.21 cm wide and 3.8 cm tall. The tracks through the rib and stringer section will be 1.52 m long. The rollers attached to the tip extension will have outside dimensions 4.28 cm tall by 3 cm wide by 15.24 cm long.

On the scaled design of 1:2, the dimensions of each component will be halved, except for the length of the rails, which will be truncated to 3 meters in total length. The dimensions of the rollers will be modified based on availability of sizes from McMaster.

3.4 Extension/Retraction Mechanism

The extension and retraction mechanisms allow the tip extension to move in and out of the main blade at a controllable rate, and secure the tip extension in place at any position. This part of the design was most complicated and addressed many design issues. Important design considerations included the ropes, pulleys, winch, attachment methods, and safety features. The design implemented a double pulley system, one for extension and one for retraction of the tip extension. The extension mechanism used a mechanical advantage of 1 because of centripetal force constantly attempting to force the tip extension outward. The retraction mechanism used a mechanical advantage of 3 to reduce the high loads due to this centripetal force.

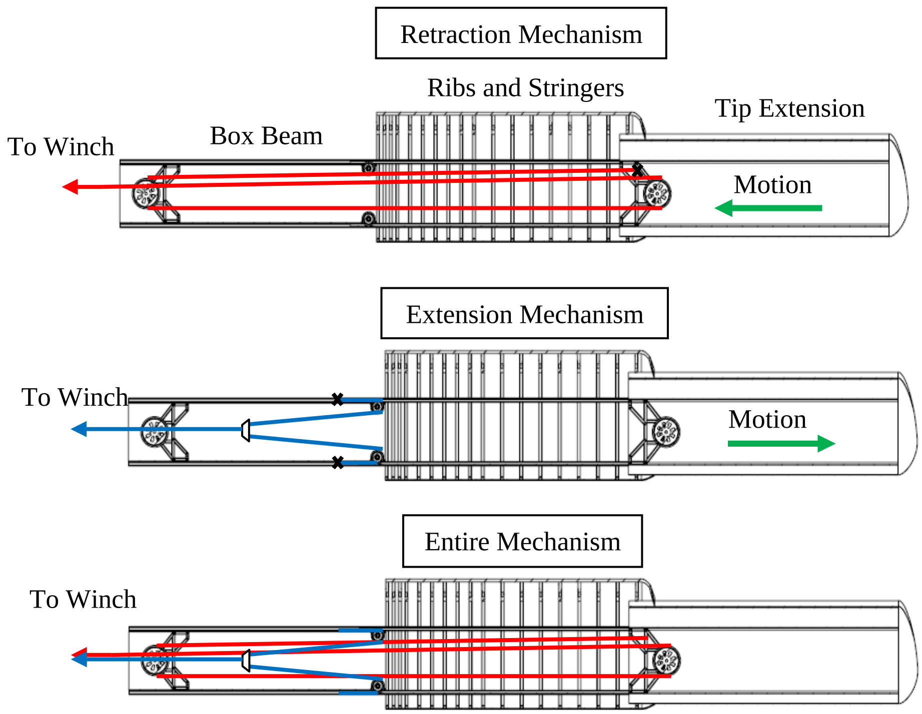


Figure 3.4T – Top View, Retraction Mechanism Pulley System
Figure 3.4M – Top View, Extension Mechanism Pulley System
Figure 3.4B – Top View, Extension and Retraction Mechanism Pulley Systems, combined

3.4.1 Ropes

As discussed in **Section 2.4** of this report, the loads that the rope must bear are critical to the successful operation of this design. The design must ensure that the chance of the ropes failing is minimal. Although the rope is designed to have a FOS of approximately 6 by reducing the load carrying capacity to a third of the total load, other design considerations regarding using ropes were addressed.

3.4.1.1 Fray Analysis

Failure due to fraying is a concern when using synthetic rope instead of steel cable in the pulley system of the tip-extension mechanism. While the strength to weight ratio of the synthetic rope is far higher than that of steel cable, the rope is not as resistant to abrasion (or fraying) as steel cable. It is important to address the abrasion resistance of the synthetic rope as it will determine the viability, lifespan, and maintenance schedule of our design.

Three approaches were considered to study the abrasion resistance of synthetic rope. The first approach was to search manufacturer's data sheets, websites, and independent tests that have been performed on synthetic ropes. The second approach was to find existing applications for synthetic rope that closely correlate to the project needs. This will help validate the decision to use synthetic rope in the design, as well as provide another set of supporting data regarding the use of synthetic rope. The final approach in determining the abrasion resistance of the synthetic rope will be to perform abrasion testing for the rope, compare the strength to steel cable, and then analyze the results to determine whether or not the rope is a viable solution for the design. These three approaches are addressed in much greater detail in **Appendix 8**, but the key concepts of each are expressed below.

Approach 1

DSM Dyneema was selected as the synthetic rope manufacturer to study. The manufacturer states that "In contrast to steel wire, which can fray and leave sharp edges, slings with Dyneema have a very smooth surface."[3] This comparison to steel wire is important in validating the claim that synthetic rope is a viable solution for the design. Frayed steel wire leaves sharp edges that could damage the pulley system, the track system, or the structure of the wind turbine blade itself. In contrast, synthetic rope would not leave sharp edges if it frayed, thus reducing the chances of overall system failure.

Another source is an independent study focused on the rope behavior in both the field and in laboratory simulations. The study examined samples of various rope diameters that were actively used in the field aboard tugboats in vessel escort service, and the samples were tested at certified, independent testing facilities. According to the study, most of the ropes that were tested were used in the field as a replacement for steel wire. The study concluded the following:

> *Abrasion and cutting damage as averaged may account for a strength loss of 5-10%. It has now been determined that compression from the drum accounts for a strength loss of 10-12%. Several lines that were tested had a moderate to severe twist, up to 1.5 turns per foot, which resulted in a 15-20% strength reduction. Abrasion and compression can account for 15-20% strength loss, and if the line has also been twisted, the combination of these three factors could account for up to 40% strength reduction.*

The synthetic rope considered has a factor of safety for strength of 9, so a 40% reduction in strength would not approach a level where failure would occur. The study also concluded:

> *The testing performed by both DSM-HPF and Samson Rope Technologies indicates that Dyneema fiber has excellent resistance to cyclic fatigue, even when tests are performed well in excess of the OCIMF TCLL cycle times. The resistance of Dyneema to both high magnitude loads and an extensive number of load cycles has been proven in laboratory testing.*

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[3] http://www.dsm.com/en_US/html/hpf/industrial.htm?source=search

The conclusions from this study validate both the abrasion resistance of Dyneema's synthetic rope and its strength.

Approach 2

The next approach was to study existing uses of the synthetic rope. It was found that Dyneema's synthetic rope is currently being used in winches attached to off-road vehicles, tow cables that connect gliders to powered aircraft, fishing nets, and yachting lines. The off-road vehicle winch application is similar to the extension and retraction mechanism, and Dyneema rope is used to replace steel wire cable in a very hostile environment.

In the glider aircraft function, Dyneema ropes are used to pull the glider down the runway behind a powered aircraft. According to stratfordgliding.com: "[Dyneema synthetic rope] lasts much longer, and is highly resistant to abrasion…we've seen a rope that's done 3000 launches on grass, and it looks like new, so it's quite possible that it will last five times as long as steel [wire]."[4] Dyneema synthetic rope is also used in fishing nets and yacht lines. This is pertinent to our design as both fishing nets and yacht lines run through a pulley system. In addition to the abrasion the synthetic rope will experience through the pulley systems, both fishing nets and yacht lines are used in a harsh and corrosive marine environment. Since wind turbines are installed both on land and in the ocean, it is important that the materials we specify in our design be resistant to the elements found in those environments, which Dyneema has proven to be.

The above applications were chosen because the uses related closely to our project. These uses impart a high level of confidence in the performance of the synthetic rope because not only are they replacing steel cables, but each use listed above involved personal safety considerations as well as design applications.

Approach 3

The final approach in validation of the decision to use synthetic rope instead of steel cable for this design is to conduct fray testing of synthetic rope and compare the results to those of steel cable. The test has been designed, and will be conducted at the beginning of next quarter. A diagram and detailed discussion of the test procedure can be found in the **Appendix 6.2**.

Conclusion

Pending the results of our fray testing, the three approaches to study the abrasion resistance of synthetic rope should provide our group the full faith and confidence in the decision of selecting synthetic rope for our design. It is evident that the synthetic rope will perform up to and beyond the design requirements. If a minimum factor of safety for strength of 9 is employed, the synthetic rope will not approach conditions leading to failure based on the results of the independent study. The current uses for synthetic rope also corroborate the decision to use synthetic rope in the design. All of these sources create a high degree of confidence in the abrasion resistant properties of synthetic rope.

3.4.1.2 Benefits

There are many benefits of choosing synthetic fiber ropes rather than steel cables which support our decision to use synthetic rope for the extension and retraction mechanisms. High performance synthetics like Spectra and Dyneema are up to 10 times stronger than similar

[4] http://www.stratfordgliding.co.uk/WR200507.HTM

diameter steel cable, and they are also much lighter than steel. Most synthetic ropes have a specific gravity of less than 1 allowing them to float in water. The weight is a significant factor, and having a rope that is much lighter than steel cable while maintaining very high strength allows our design to keep its weight reduction to a minimum.

Synthetic ropes have very good anti-corrosive properties, and are commonly used in marine environments. This would not be an issue for most wind turbines, but some wind farms have been installed in the ocean, where it is an important consideration. Polymer ropes offer exceptional bending and tensile fatigue resistance, high abrasion resistance, extremely low stretch, and low or no creep. They also are very resistant to kinking, unlike steel cable. These properties are important for this mechanism because they provide a high degree of confidence in the long term performance of the rope.

3.4.1.3 Aircraft Cables

During the design process, aircraft cables were considered as an alternative for synthetic ropes. An “aircraft cable” is a cable made of special-strength wire designed primarily for the use in aircraft for civilian and military applications. Aircraft cables have two standard strand constructions, 7x7 and 7x19, and can commonly be found in diameters ranging from 1/32” up to 3/8”. The reasoning for only having two standard strand constructions is due to the requirement that the cable manufacturers must meet certain military specifications in order to be placed on the Qualified Producer List (QPL). These specs include the following:

- MIL-DTL 83420 - Wire Rope. Flexible. For Aircraft Control
- MIL-DTL 87161 - Wire Strand. Nonflexible, For Aircraft Applications
- MIL-DTL 18375 - Cable. Steel Non-Magnetic

The materials that make up the cable vary by use, but the material most commonly used by Boeing in their commercial airliners is the tin over zinc variety of carbon steel, where they require up to ½ mile of aircraft cable at 1/8” diameter.

Since aircraft cable has been used thoroughly for many decades and is trusted by manufacturers of commercial airliners such as Boeing, applying these highly developed cables to the design of a wind turbine would not require extra reliability validation, but the strength of common sizes of these cables is much lower than what is required by the extension and retraction mechanism. QPL aircraft cable manufacturers commonly make cables up to 3/8” diameter with a maximum breaking strength that approaches 12,000 lb. The tip of the turbine blade will feel a maximum force of approximately 15,000 lb at full speed. Using a minimum factor of safety of 6 for the retraction cable to insure that the cable outlasts the lifespan of the windmill, a cable with break strength of greater than 90,000 lbs is necessary. In order for this breaking strength to be reached, a much larger diameter aircraft cable would be required resulting in much heavier cables of up to 1000 pounds per 1000 feet.

A direct comparison of aircraft cable versus Dyneema polymer fiber rope can be found in **Table 3.1**. This addition in weight would not be viable because it is too large a percentage of the allowable increase in weight of the entire design, and would not allow for the addition of other necessary components such as the tip extension itself. The additional weight would also significantly increase the bending moment, increasing stresses felt throughout the turbine blade. In order to counteract the increase in stress, the supporting box beam structure and the airfoil shell would need to be increased in thickness, once again increasing the weight of the turbine blade. This comparison is helpful in establishing that a polymer rope with a very high strength to weight ratio is preferred.

Type of rope	Aircraft Cable	Polymer Fiber Rope (Dyneema)
Diameter	¾ in	¾ in
Breaking Strength	46,000 lbs	68,000 lbs
Weight per 1000 feet	970 lbs	124 lbs
Pros	Highly developed and tested Used in high-value applications Reliable Good abrasion resistance Corrosion resistance	Extremely high strength-to-weight ratio Becomes approx. 10% stronger as it stretches resulting in less backlash if failure occurs Good abrasion resistance Low wear High corrosion resistance
Cons	High wear Relatively heavy Destructive backlash at failure Weaken over time	Not as high abrasion resistance as steel cable More prone to twisting Weaken over time

Table 3.1 – Comparison of Aircraft Cable and Dyneema Polymer Fiber Rope

3.4.1.4 Other Rope Considerations

Some other important aspects of ropes that were considered were mechanisms of rope attachment, types of rope, and temperature effects on rope. Detailed descriptions of each are given in **Appendix 7**, but short explanations of each are given here.

Ropes can be attached to each other or external points by splicing, knots, and composite termination fixtures. Splicing retains a large amount of the strength of the rope, and is semi-permanent. Knots are much weaker than splices, but require much less time and specialized labor for implementation. Composite termination provides a fixture on the end of the rope that can be specified based on the different types of attachment hardware that the rope connects to. They are strong and reliable, but are not always available.

The four types of rope considered were Dyneema, Technora, Spectra, and Vectran, which all possess the qualities necessary for the design. Spectra and Vectran ropes were tested in the mechanical testing lab, but the other two were not available in lengths of less than 1200 ft. For the full-scale design implementation, we recommend using either Dyneema or Technora ropes because they have optimal combinations of properties needed for this design. However, the prototype will use Spectra rope because it is easily obtained in small lengths, and can still support heavy loads.

As the temperature increases, the strength of ropes drops below the manufacturer's stated value. However, the ropes will not usually undergo extreme temperatures, and in cases where high temperatures will be encountered, Technora rope will be used because it is designed for temperature resistance.

3.4.2 Pulleys

The pulleys are also an integral part of this design, and must withstand maximum operational loads for the mechanism to operate properly. As discussed in **Section 2.5** of this report, the pulleys must withstand 2 times as much force as the ropes will, which is 2/3 of the total applied load. Using a setup of pulleys to provide a mechanical advantage of 3, the total loads that the pulleys must bear is approximately 10,000 lb.

In selecting parts for the prototype, high density nylon pulleys were the initial choice because of the large weight savings compared to metal. However, after consulting with Kirk Fields the decision was made to use metal pulleys, because the synthetic ropes slide along them much more easily than they would on nylon pulleys. This will avoid the problem if high frictional forces are applied creating a significant melting hazard to the pulleys. Metal pulleys are also much stronger, which is a significant issue, as they must support significant loads.

FEA analysis was performed on the pulleys to determine the strength under normal loading conditions. Using this analysis, we determined that Aluminum 6661-T6 pulleys will not fail under the given loading conditions. The results of this FEA analysis can be found in **Appendix 11**.

3.4.3 Winches

Two winches are needed for this design, one for both the extension and retraction mechanism. The extension winch doesn't have to supply a large amount of force because centrifugal force helps extend the tip. Most of the time, centripetal acceleration is such that the tip extension would move outward if it were unrestrained. The only situation where extra power is needed is if the turbine blades are stationary or moving very slowly. In that situation the maximum acceleration to overcome is 1g.

The retraction winch on the other hand, must be able to overcome a force of 12 g's when the turbine blades are turning at 15 rpm, so the winch pulling capabilities have to be increased accordingly. When a factor of safety of 1.5 is included, the winch must be able to support a tensile force of 15,000 lb through the rope.

This setup of winches in a full scale implementation of this design needs to be driven by a computer controller in order to work automatically. The computer must spool the two winches in unison because both ropes need to move any time the tip extension moves in or out. Gear reductions from the motors, difference in size of the two winches, and the difference in mechanical advantage cause the winches to spool at different rates, so a computer controller is needed to ensure cooperation between the two systems. The winches also must detect and correct for slack in the lines due to stretching of the ropes under load.

In order to minimize the bending moment on the blade, the winches are placed at the very base of each blade. This allows the winches to be easily accessed for service and maintenance because the base of the blade is open for access from inside the hub of the turbine.

3.4.4 Safety

3.4.4.1 FOS

Several different factors of safety have been implemented in the components of the project depending on the criticality of each one. A FOS of 1.5 was chosen for the motor. This number was arrived at through a compromise between performance and weight. A high factor of safety would result in a motor which is much more powerful than necessary, which would also dramatically increase the weight added to the structure. It is important for the motor to be robust and to have enough power to operate smoothly while overcoming the centripetal force, so a FOS of 1.5 is a good middle ground.

A FOS of 1.5 was also employed for the structure itself with a similar justification balancing the strength of the structure and its weight. Again, a large FOS causes dramatic increases in weight, creating a structure that is strong but too heavy. Using a FOS of 1.5 for structural members allows some room for approximation, but keeps the weight down to an acceptable level.

Finally, a FOS of 9 is applied to the ropes used in our extension and retraction mechanisms. This high factor of safety is necessary for two important reasons. First, the rope will be required to hold loads up to about 15% of its breaking strength. Second, the ropes are the only radial support the tip extension relies on. It is very important for the ropes to be able to withstand a much greater force than could ever be felt by the mechanism. The use of synthetic ropes means that an increase in weight due to a large FOS is not as significant as what that same increase would be for a structural member. This is justified because the weight of the full length of rope is only around 2% of the total allowable weight of for this project.

3.4.4.2 Worker Safety/Blade Structure Protection

Synthetic ropes are much safer for workers than steel cable because of the low weight and high flexibility. According to HiMFR.com,

> *[Stainless steel]Wire cables on your winch will fray, kink and get impossibly tangled on your winch drum… Wire cables retain a great deal of energy and a cable that snaps under load becomes a high speed knife with an unpredictable path. Synthetic winch rope, however, will not kink or tangle and retains very little energy. If you cut our synthetic winch rope while it is under load it simply falls to the ground.*[5]

This critical failure characteristic of steel would be devastating to a windmill. Since synthetic rope would not exhibit the same failure mode in a tensile break, the rope would cause minimal damage, if any, to a windmill.

3.4.5 Fiberglass Attachment Methods

The structures and mechanisms in our project rely heavily on the assumption that we will be able to securely attach our equipment to the fiberglass box beam. Finding a suitable fiberglass attachment method to withstand the high forces and stresses is essential to the success of our design. On a full scale implementation of this tip extension design, forces of up to 15,000 lb would need to be supported at a single attachment point. Where possible, a combination of bolting and adhesive bonding will be employed, with special care taken to support the critical attachments. The testing of different attachment methods supporting this joint type is discussed in detail in **Appendix 6.3**.

[5] http://www.himfr.com/d-p113208739860144600-Dyneema_Electric_Winch_Rope-The_Wire_Rope_Best_Replacement/

3.4.6 Final Proposed Extension/Retraction Mechanism Design

The final proposed design for the extension and retraction mechanisms can be seen in **Figure 3.4B**. It is comprised of two separate mechanisms, an extension and retraction mechanism.

The extension mechanism consists of Dyneema rope running down the center of the box beam that attaches to a three-way turnbuckle. The output ropes from the turnbuckle run around 5” diameter pulleys attached to the box beam and attach to the ends of each fork (**Figure 3.4M**). When the extension winch spools in, the tip extension will be pulled out.

The retraction mechanism consists of a rope attached to the base of the tip extension which runs through a 14” diameter pulley attached to the box beam, and then back through another 14” pulley on the tip extension, before running back through the box beam to the winch (**Figure 3.4T**). The winch spools in and retracts the tip extension with a 3X mechanical advantage due to the pulley system. The two winches are contained by a bracket that attaches to the base of the box beam near the hub.

On our prototype 1:2 scale, Spectra rope will be used (as explained in **Section 3.4.1.4**). The size of the extension pulleys will be x in diameter. The size of the retraction pulleys will be y in diameter. Because the length does not allow us to attach a winch to our prototype, when performing our testing, winches and our tip extension must be secured separately. These must be secured so neither winch nor prototype can influence the secured distance from one another. Using these two pulley mechanism in conjunction with one another, we will determine if we will meet our design requirements of overcoming a 6 g force on the tip extension, and both systems allowing the full 1.5m of travel in less than 20 seconds.

3.5 Maintenance

According to Clipper Windpower’s Liberty wind turbine brochure, the Liberty wind turbine is “Germanischer Lloyd certified to 20 and 30 year fatigue lives.”[6] Clipper also states in the brochure that scheduled maintenance is to occur after the first 700 hours and then the turbine should be serviced every six months from that point on. One of the design specifications given to us for our design was that it be a robust and low maintenance design because unscheduled maintenance is a very costly consequence of design failure.

We have created a maintenance plan for our project that will extend the fatigue life (or useful life) of our design to the stated fatigue life of the rest of the turbine (20-30 years). Each component of our design was analyzed to determine which material and which component design would best extend the fatigue life for that component. The next level of fatigue life analysis studied how the assembly systems would wear over time using the specified components. The final task was to create a maintenance plan for each assembly that would extend the fatigue life of the project to the necessary time frame. The maintenance plan we created was designed to fit in with Clipper Windpower’s existing maintenance schedule.

3.5.1 Component Specification

The components most likely to fatigue in our design are assemblies that contain moving parts. There are three major assemblies in our design that contain moving parts. Those assemblies are the winch motor, pulley system, and track.

[6] Liberty brochure [1], Clipper Windpower, www.clipperwind.com

The winch motors will be located in the root of the base blade near the hub of the turbine, so they will be easily accessed through the existing opening in the nacelle of the turbine. The winch motors will need to be lubricated and have their electrical components checked. If prevailing wind conditions allow, the winch motors should be tested through a complete tip extension and retraction cycle. The winch motor mounts should be inspected as well as the winch braking system.

The pulley system of our design consists of smooth stainless steel pulleys, pulley bearings, pulley mounting brackets, and synthetic rope. Since the pulleys, pulley bearings, and pulley mounting brackets are located toward the end of the base blade, access to these components is extremely limited. Access is such an issue that it was deemed unfeasible to create a maintenance plan for those components. The decision to forgo regularly scheduled maintenance for those three components meant that our group had to select design materials that would last for the specified fatigue life of the turbine.

The first component of the pulley system that we considered was the pulley. Stainless steel pulleys were selected for the design. Stainless steel pulleys "are stronger than Delrin and nylon pulleys for handling small diameter rope" and "supply superior corrosion resistance"[7] according to McMaster-Carr. The pulleys our group selected include bearings, which will also increase the life of the pulley over nonbearing pulleys.

The best bearing option for the pulleys are plain bronze bearings which are also "ideal for long life."[8] In addition to the material that was selected for the bearings, the construction of the plain bronze bearings is to be double sealed which will "block out dirt, preserve lubricants and reduce noise."[9] The pulley brackets will be made from titanium and are designed to withstand the forces that will act on them. Titanium has a high fatigue life, and has great strength to weight ratio, meaning it can have a higher FOS to increase lifespan to beyond 30 years.

These three components of the pulley system have been designed to meet or exceed the fatigue life of the rest of the turbine and will not require a maintenance plan. In the unlikely event that one of the above components should fail before the expected fatigue life of the turbine, the entire blade of the turbine will need to be removed and serviced onsite or replaced. The old turbine blade may need to go to Clipper's manufacturing facility where Clipper will determine whether or not to refurbish the blade, or to recycle it. Our group realizes the enormous expense associated with replacing an entire blade on a turbine. However, any tip extension design with the mechanisms located toward the end of the base blade will need to address the issue of inaccessibility. It might help to consider offsetting the increased maintenance costs of the tip extension design with the increased value of generating power at lower wind speeds.

The final component of the pulley system to consider is the synthetic rope. We have developed a maintenance plan for the synthetic rope that should be implemented with Clipper Windpower's existing maintenance schedule. The maintenance plan is to check the synthetic rope for any abrasion or other signs of wear. This can be done by inspecting the spool of the winches located at the root of the base blade. As the winches extend and retract the tip extension, the inspector should study the synthetic rope as it spools around the drum of the winch.

There are two pulley systems, one for extending the tip extension and another for retracting it. It will only be feasible to replace the rope in the retracting pulley system. In the

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[7] http://www.mcmaster.com/#stainless-steel-pulleys/=xqwsb
[8] http://www.mcmaster.com/#9466t63/=x7ce8
[9] http://www.mcmaster.com/#9466t63/=x7ce8

event that the retracting rope should be replaced, the main turbine blade which contains the worn rope will need to be locked in the downward facing position. The tip extension will then be lowered beyond the tip of the base blade so that the base of the tip extension is beyond the end of the base blade by about one meter. The rod lock failsafe system will then be engaged to lock the tip extension in place.

This will reveal the pulley system attached to the tip extension, as well as the termination points of the rope to the tip extension. With the rod locks in place, the rope can be detached from the tip extension without the tip extension falling out. A technician will rappel down the face of the turbine to the gap between the base blade and the tip extension where they can perform the rope replacement procedure. After studying Clipper Windpower's existing maintenance plan, it was discovered that part of the maintenance technique includes a technician harnessed to a line that hangs from the nacelle of the turbine and rappelling down the face of the turbine to conduct the scheduled maintenance, so the rope maintenance is a reasonable addition. The new rope will be temporarily spliced to the old rope so that the old rope can guide the new rope through the pulley system. Once the new rope reaches the winch, the old rope will be cut from the new rope and a permanent end connector will be installed onto the new rope. The new rope will then be assembled to the winch, and the tip extension will be retracted.

It is not feasible to replace the extension pulley system rope as the rope termination points are well within the base blade which creates an accessibility issue. This means that the extending pulley system rope will need to meet or exceed the turbine's anticipated life expectancy. That is why our group has designed the extension rope with a factor of safety of at least 9 for strength. In addition to this safety factor, it should be noted that there will be minimal forces applied to the rope in the extending pulley system as centripetal forces will provide most of the force needed to extend the tip extension. What this means is that the rope in the extending pulley system will have no problems meeting the longevity requirement.

The track system of our design consists of bearings and a composite track. As with the pulley system, the track is inaccessible because it is located inside the end of the main blade. The bearings specified in the track design are double sealed in the same manner as the bearings selected for the pulley system. The track is going to be made from a similar material as the base blade. The design and material selection of these track system components considered the life expectancy of the turbine and they should have a similar lifespan.

3.5.2 Maintenance Summary

The components that are inaccessible have been designed with materials and a construction that have been proven to be robust in the past. The same or similar materials as the base blade will be used in our design in an effort to match the life expectancy of the turbine. For the components that are of concern, namely the winch motors and the synthetic rope used in the pulley system, a maintenance plan has been designed that will integrate with Clipper Windpower's existing scheduled maintenance plan.

4 Results

The technical and design considerations for this report has helped this project progress a considerable amount. Aside from these considerations, prototyping, testing, modeling, and analysis have been performed to help prove the feasibility of our design. At this point in time, proof of concept for our design has been achieved. The proof of concept model aided our selection of the pulley mechanism and proved that the design concept we chose is a valid one. Our team is close to proving design feasibility, pending the results from our planned testing and FEA analysis, which is to be completed within the first two weeks of spring quarter.

The prototype, as shown in **Figure 4.1**, will demonstrate design functionality. The scope of this project only requires the design of a mechanism that will move a tip extension into and out of a turbine blade under 6 g's of acceleration. The final design will consist of the four proposed designs described in **Sections 3.1.5, 3.2.6, 3.3.5, and 3.4.6**: an interior support structure of a main blade, a tip extension to provide extra lift, a rail system for the tip extension to travel upon, and a dual independent pulley system used for extension and retraction. All will be built on a 1:2 scale, with a truncated overall length of 3 meters. Explained in **Section 3.1.5**, this truncation in length shouldn't affect our prototype from meeting the design goals. Performance of this prototype will be tested, which will validate feasibility and check if our design goals are met. Our design and analytical efforts will go further than our scope by performing a preliminary analysis to determine the feasibility of implementing this mechanism on a turbine blade.

To understand more succinctly what results have been achieved, the following actions are described in more detail.

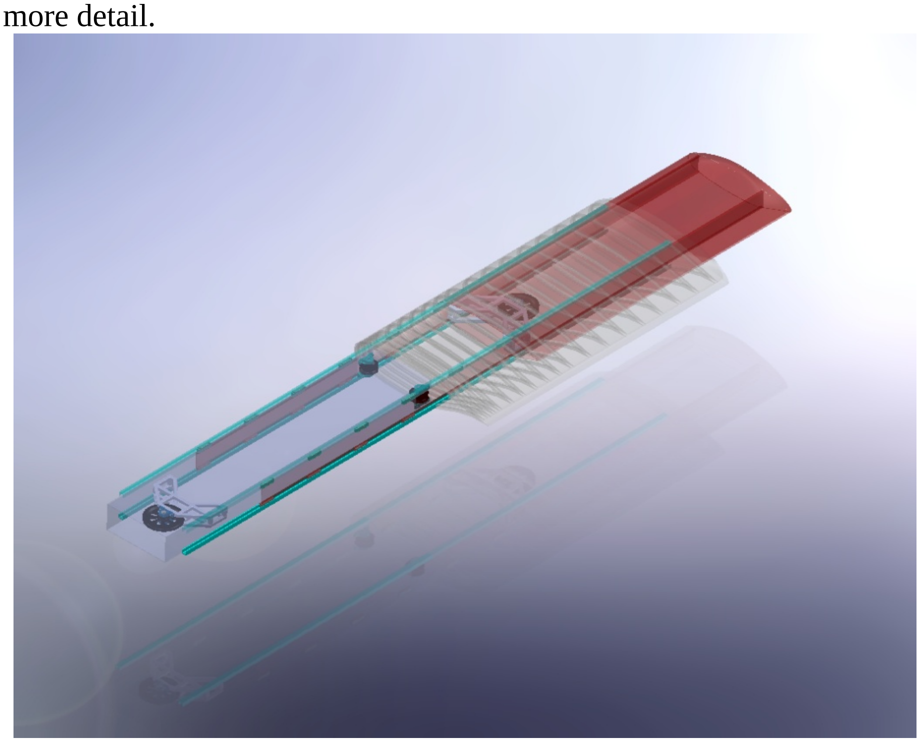


Figure 4.1 – Proposed Prototype Model

4.1 Prototyping

To date, a design has been agreed upon, and a scale prototype has been drawn up (see **Appendix 12)**, with the intention of being completely constructed by the end of spring quarter. This prototype will be used to understand if the design will meet planned performance characteristics, and should validate design feasibility.

To understand if the proposed design would work on a smaller scale, our team constructed a test-bed prototype this quarter, found in **Appendix 12**, which was used to understand and prove our design would function. Being a test-bed prototype, it also accommodated an alternatively proposed “lead screw” mechanism to understand and prove if it operated (the alternative proposed designs are described in **Appendix 3.3)**. This prototype demonstrated proof-of-concept of our design.

4.2 Testing

Some components of our design required physical testing to help prove feasibility and make correct design decisions. Because our team chose to use ropes instead of steel cable for our design, physical testing of the proposed ropes needed to be performed. Unlike steel cable, ropes can stretch which results in slack –an unwanted aspect in our design. Our team needed to determine how much slack would result from certain levels of loading, and which type of rope provided a good weight reduction and low level of slack. Our test results determined that Dyneema was the rope of choice to use for this design, but due to its large order size, Spectra, a similar type rope will be substituted for our prototype.

Another aspect of ropes that differs from cables is fray. Testing to determine how rope would handle under frayed conditions needed to be performed. Because of the level of complexity involved with setting up this test, it has not yet been completed. Plans and drawings of this test can be found in **Appendix 6.2**.

To understand how to attach fiberglass components to one another, our team tested different types of attachment methods. This is incredibly important for our design, because without it, our design may be feasible but its implementation may not be. Our testing results found that using a bonded lap joint used in conjunction with a bolt attachment will be sufficient to hold the loads we have determined our mechanism will be subjected to. All testing details, planned and performed, can be found in the **Appendix 6.3**.

4.3 Modeling

Extensive modeling has been performed, in order to understand how our design will interact with existing turbine blades. A drawing packet has been supplied in the **Appendix 14**, which includes models and drawings of the proposed scale prototype as well as models, drawings, and an assembly package for the full-scale design. Extensive, detailed work has gone into modeling this design, which all can be observed in the included drawing package.

4.4 Analysis

Extensive analysis has been conducted this quarter. At the start of this quarter, a decision matrix was created to analyze which type of approach to increasing a wind turbine’s efficiency should be picked. Following this decision and after extensive analysis into different types of mechanisms used for tip extension deployment, a cost vs. weight analysis was performed on the two final mechanisms to determine the final mechanism design which is located in **Appendix 3.3**.

Analysis of the increase in efficiency of the turbine, the force required to hold and move the tip extension, the result of adding weights and moments, the forces applied to ropes, and the forces applied to pulleys was performed. The details of these analyses can be found in **Section 2** of this report. Analysis of our test results was performed to determine what types of rope and what types of fiberglass attachment methods should be used. There was thorough analysis of the maintenance factors in order to understand how to design our mechanism around the lifetime of the product we're intending to enhance.

The final things that need analysis for this project that are scheduled to be completed next quarter are the FEA analysis of the rib and stringer section of the turbine blade and the rope fray testing results. The analysis of the rib and stringer section is necessary because understanding how this proposed structure will support itself is critical to determining the feasibility of the project. Understanding if the ropes proposed to be used in this project will fail or not is critical. If we learn that these ropes can either easily fray or eventually start to fray, analysis needs to be performed to understand if or when the ropes proposed to be use would need replacing.

4.5 Action Items

As assigned in our Preliminary Design Review, the following action items have been addressed.

4.5.1 Research airplane wing cables and functionality

As explained in detail in **Section 3.1.4.6**, airplane wing cables will not be feasible to use in our design. Aircraft cable is too heavy, has internal friction, and does not come in sizes large enough to support the weights our design intends to hold. Also, it is impossible to inspect the center of steel cables, where heavy wear can occur. Further, if failure of steel cable would occur, it fails catastrophically, which could result in the destruction of the main turbine blade. By comparison, rope doesn't fail catastrophically, which would leave the turbine blade unharmed. This led us to the conclusion that aircraft cable should not be used.

4.5.2 Research long term wear issues – Cables versus Ropes

As mentioned in the previous section, cables break violently whereas ropes do not. Cables are about 10 times heavier per length than rope and have far less strength than rope. According to research performed, the wear for cables and ropes occurs, and is unavoidable by each. Replacement would be easier with rope, since new rope can be spliced with old rope and fed through a pulley system to be replaced, whereas steel cables cannot. Steel cables of large thickness are difficult to wrap around a small pulleys, whereas rope of the same thickness can easily bend. These findings, paraphrased from **Section 3.4.1**, lead us to the conclusion that ropes are a better choice for our design.

4.5.3 Identify patents on mechanism

Our patent review, found in **Appendix 5**, resulted in a few patents that implemented the same tip-extension design concept, but they did not supply details of the mechanism that would deploy the tip extension. Another patent did, and implemented a pulley system. The design outlined in the patent used a single pulley system, and only one pulley for that system. The concern of a patent infringement is relevant, since similar patents do exist, but our research has not found anything that matches the details of our design with any accuracy.

5 Conclusion and Recommendations

Our team has performed in-depth technical and design efforts for this project. The last two quarters were spent performing mostly conceptual design work and research. Several key metrics from the PDS and Project Plan have been updated as well. The minimum wind speed for maximum power output at the full scale was revised from 10 m/s to 12 m/s after analysis showed that the tip extension would be required to be as long as the base blade to reach the original estimate. As discussed earlier, the scale of the prototype has been adjusted from a 1:43 scale to a truncated 1:2 scale. As more detailed design work was done, we came to the conclusion that a 1 m base blade would cause the tip extension to be so small and thin that it would be impossible to find the necessary parts without doing extensive custom machining. The decision was made to alter the scale because it allows for a much more accurate view of what the design would look like at full scale. The budget and milestone calendar have also been amended to reflect the changes the design has undergone.

At this point in time, we propose to spend a majority of the rest of our time allocated for this project to build a prototype of the design. A prototype scale has been picked out and performance metrics have been calculated for the design. Spending our time building this prototype will show proof of concept. Testing the prototype to achieve actual performance results will prove feasibility.

As addressed in the preliminary design review, our team lacked physical proof regarding the feasibility of incorporating rope instead of steel cable into part of the design mechanism. Fray testing is planned and has begun being set up (**Appendix 6.2)**. However, the complexity of the setup for this test means that our team could not design, prepare, order, and build a testing apparatus in two weeks. We will use the start of next quarter to complete this test and draw conclusions regarding whether rope is a suitable choice for the design.

To fully prove feasibility of our design, FEA analysis also needs to be performed. The strength of the rib and stringer section needs to be analyzed using FEA due to its structural complexity. However, because of the complexity of the modeling of this section, which has already been completed, more powerful computers are needed, and will be obtained next quarter to complete our analysis.

Upon completion of these tasks, we will finish the project by scaling the performance characteristics of the scaled prototype to the expected performance for a full-scale prototype. We will also determine if our design will meet weight requirements, and double-check that our tip extension will supply enough lift to increase efficiency. With these results, we can provide Clipper Windpower with a recommendation as to the feasibility and potential economic viability of our design.

Appendices

1 Project Team and Acknowledgements

Member	Responsibility
Ryan Sass	Team Leader, project planning, communications, project presentations
Adam Donald	Test sample construction, component testing, report and presentation editing
Rustin Harrington	Scale prototyping and proof of concept construction, project budget
Michael Ringen	Solidworks modeling, FEA analysis, project binder organization
Adam Sutton	Results analysis, scaling calculations, decision matrix forming

Special thanks to:

Stephen Laguette	Course Advisor
Robert McMeeking	Faculty Advisor
Kirk Fields	Test Lab Supervisor
Zach Markey	Clipper Windpower
Tony Chobot	Clipper Windpower

2 Project Budget and Expenses

Fall Budget	
$ 75.00	Stick and Bubblegum Prototype
$ 75.00	**TOTAL**

Winter Budget	
$ 125.00	Proof of concept mockup
$ 125.00	Testing materials
$ 350.00	Motor
$ 100.00	Track/Pocket Bearings
$ 100.00	Pulley system/bearings
$ 100.00	Composite adhesive
$ 40.00	Tip extension bracket
$ 700.00	Blade shell material & Resin
$ 50.00	Box beam
$ 200.00	Cable/Rope
$ 40.00	Fasteners
$ 250.00	*Contingency*
$2,180.00	**TOTAL**

Spring Budget	
$ 200.00	Tip extension shell material
$ 100.00	Tip extension support beam
$ 100.00	Composite adhesive
$ 50.00	Tip extension rollers
$ 60.00	Fasteners
$ 200.00	Poster/Printing
$ 20.00	Electric Cabling
$ 20.00	Batteries
$ 200.00	(Rod Locks)
$ 40.00	Paint
$ 250.00	*Contingency*
$1,240.00	**TOTAL**

TOTAL PROJECT COST	**$ 3,495.00**

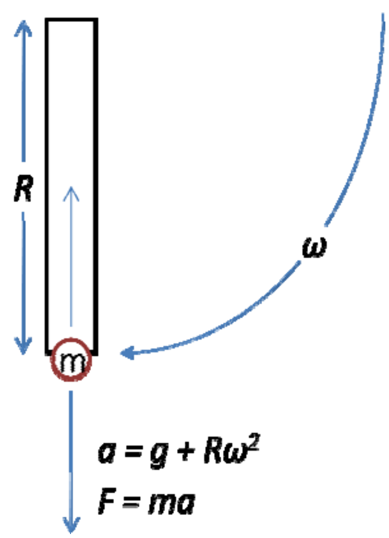
3 Extension Mechanism Report

3.1 Objective

To determine the mechanism for a tip extension that is has a light weight, minimizes the added moment, and is cost effective.

3.2 Design Restrictions

The extension mechanism must be able to withstand the max acceleration and corresponding force that is produced when the blade is moving at a rotational velocity of 15 RPMs (1.57 radians/second) at the very bottom of the windmills rotation while the blade is being retracted. In order to make this calculation, a blade length (R) of 43 meters was used with a rotational velocity of 1.57 radians per second.



The resulting acceleration came out to be approximately 115.79 m/s^2 or 11.81 g-forces.

The mass that the extension mechanism along with the tip extension itself cannot exceed a mass of more than $1/10^{th}$ of the mass of the turbine blade which provides us with a weight envelope of approximately 1200 kg.

3.3 Design Considerations

3.3.1 Double Acting Pneumatic Cylinder

The pneumatic cylinders work by compressed air entering into one end of the tube where it produces force on a piston and the piston becomes displaced due to the air trying to achieve atmospheric pressure. To calculate the maximum pressure case equation **(1)** was used. The pressure calculated from this case is the pressure required for the in-stroke with a max acceleration case of 11.81 g's with an approximate tip mass of 600 kg.

$$F = P\frac{\pi}{4}({d_1}^2 - {d_2}^2) \quad \textbf{(1)}$$

$$F = Force$$
$$P = Pressure$$
$$d_2 = piston\ rod\ diameter$$
$$d_1 = full\ bore\ piston\ diameter$$

The results of multiple tests can be seen in the table below. For the calculations, a tip mass of approximately 600 kg was used.

Force required (N)	111205.5556
Length (m)	10
Tip Mass	600 kg

Piston Diameter (m)	Rod Diameter(m)	Pressure Required (Pa)	Pressure Required (kPa)	Pressure Required (psi)	Weight (kg)
1	0.2	147490.9489	147.4909489	21.39208723	3699.22535
0.95	0.19	163424.8741	163.4248741	23.70314374	3338.550878
0.9	0.18	182087.5912	182.0875912	26.40998423	2996.372533
0.85	0.17	204139.7216	204.1397216	29.60842523	2672.690315
0.8	0.16	230454.6076	230.4546076	33.42513629	2367.504224
0.75	0.15	262206.1313	262.2061313	38.03037729	2080.814259
0.7	0.14	301001.9365	301.0019365	43.65732087	1812.620421
0.65	0.13	349091.0033	349.0910033	50.63215911	1562.92271
0.6	0.12	409697.0802	409.6970802	59.42246452	1331.721126
0.55	0.11	487573.3847	487.5733847	70.71764372	1119.015668
0.5	0.1	589963.7955	589.9637955	85.5683489	924.8063374
0.45	0.09	728350.3649	728.3503649	105.6399369	749.0931333
0.4	0.08	921818.4305	921.8184305	133.7005452	591.8760559
0.35	0.07	1204007.746	1204.007746	174.6292835	453.1551053

Table A3.3.1- Results that show the weight of the pneumatic cylinder and the pressures required to move the system under an 111,205 N force with various piston and rod diameters.

The main problem with the pneumatic cylinder is the amount of mass that it would add to the turbine blade. Some other problems include the consistency of the displacement under quickly changing loads, which may cause balance issues on the windmill, and the mass of the air compressor that would be needed to provide the air pressure. This design is not possible due to the mass restrictions given to us by Clipper.

3.3.2 Lead Screw

Lead screws are very efficient in translating rotational motion into linear motion. Some of the advantages to using a lead screw would be that it has a large load carrying capacity, simplicity in design, precise linear motion, minimal number of parts, and the fact that many are self locking. One of the large disadvantages is that they have a high degree for friction on the threads which causes the threads to wear out quickly. They are also not very efficient and as a result, it is not recommended that they be used in continuous power transmission applications.

To further look in to the possibility of using a lead screw for the tip extension mechanism, added weight, cost and moment were all looked at. In order for this mechanism to be able to pull in the tip extension in the max acceleration case (the lead screw having a working load of ~ 25000 lbs) a minimum lead screw diameter of 2 ¼ inches is required. Some of the other main components that are required for the lead screw mechanism include a motor, a nut, and a gear box. The corresponding weight, costs, and added moments can be found in **Table A3.3.2** below.

Component	Part Number	Quantity	Length (ft)	Weight Per 100 Foot	Weight (lbs)	Cost Per 100 ft	Total	Moment (lb*ft)
Lead Screw	11104	1	33	1000	330	400	4400	35145
ACME flange nut	70260	1			0.6	40	40	54
DC Motor	CD2010P-2	1			267	4500	4500	24030
Gear box		1			68.2	400	400	6138
				Totals	665.8		9340	65367

Table A3.3.2- Shows the added weight, cost, and moment of various components of the lead screw mechanism.

The main concern with the lead screw design is not the weight, but the location of the weight in respect to the hub. Due to the large moments produced by the 10 meter long 2 ¼ inch lead screw and the motor required to turn it, this design would not be feasible without the introduction of very large stresses throughout the turbine blade.

3.3.3 Pulley System:

Pulley systems are commonly used in systems where a mechanical advantage is needed in a linear system of motion. Some advantages of using a pulley over other systems of linear extension are that it is relatively lightweight, the pulleys can be used at a distance from each other, and they have been used extensively in the past in heavy lifting applications.

To look into the possibility of using a pulley system as our mechanism of extension, added weight, costs, and moments for the main components of a pulley system were looked in to. The table below (**Table A3.3.3**) sums up the results found from various sources.

Component	Part Number	Quantity	Length (ft)	Weight Per 100 Foot	Weight (lbs)	Cost Per 100 ft	Total	Moment (lb*ft)
Pulley		6			42	10	60	4830
Vectran Rope		1	400	19.2	76.8	9.25	3700	6912
Spectra Rope		1	400	13.3	53.2	9.5	3800	4788
Spool\winch		1			183.3333333	5000	5000	0
				Totals	302.1333333		8760	11742

Table A3.3.3- Shows the added weight, cost, and moment of various components of the pulley mechanism.

Based on the minimization of the three important factors of added weight, cost, and moment, the pulley system comes out on top. The difference in added moment alone between the pulley and lead screw mechanisms is approximately 53,625 ft*lbs. This leads to the conclusion that the pulley system will be the most practical and efficient way of conducting our extension mechanism.

3.4 Conclusion

The three design considerations include a double acting pneumatic cylinder, a lead screw, and a pulley system. The double acting pneumatic cylinder would introduce many weight problems that would arise due to the weight of the piston and the air compressor. As a result of the weight issues, this concept was disregarded. The lead screw faced a similar problem, although besides the added weight causing the issue, it was the location of that weight relative to the hub of the windmill. Because of the large moments caused by the lead screw design, this concept was disregarded as well. The system that will be used for our extension mechanism is the pulley design due to its low weight and the fact that is has been used through the past on many heavy loading applications.

3.5 References

"Product Detail Balador 10 HP DC Motor." Applied.com Industrial Bearings, Material Handling, Power Transmission, Fluid Power Products | Applied Industrial Technologies | Applied.com | Applied.com. 16 Feb. 2009 <http://www.applied.com/apps/commerce/catalog/catalog.do?e=10&s=615741&r=0&type=a&mp=CD2010P-2>.

"Product Detail 2 1/4 in Acme Lead Screw." Applied.com Industrial Bearings, Material Handling, Power Transmission, Fluid Power Products | Applied Industrial Technologies | Applied.com | Applied.com. 16 Feb. 2009 <http://www.applied.com/apps/commerce/catalog/catalog.do?e=10&s=8580040&r=0&type=a&mp=11222>.

Winches and Winch Accessories by Winch Depot. 16 Feb. 2009 <http://www.winchdepot.com/>.

4 Decision Matrix

		Tip Extension			Hinged			Flaps/Slats		
Weighting Factor	Categories	Magnitude	Rating	Score	Magnitude	Rating	Score	Magnitude	Rating	Score
25	Weight Increase	818 kg/blade	5	1.25	3181 kg/blade	1	0.25	1000 kg/blade	3	0.75
20	Cost Increase	$20k / blade	10	2	$53k / blade	4	0.8	$24 k / blade	8	1.6
15	Power Increase (at 12 m/s wind speed)	12%	5.5	0.825	20%	7	1.05	30%	10	1.5
20	Added Moment	231 kN/m / blade	5	1	1213 kN/m / blade	1	0.2	215 kN/m / blade	6	1.2
10	Complexity	-	5	0.5	-	3	0.3	-	5	0.5
20	Reliability	-	5	1	-	2	0.4	-	7	1.4
5	Maintainance	-	6	0.3	-	1	0.05	-	5	0.25
5	Scalability	-	7	0.35	-	3	0.15	-	8	0.4
	Total Score			**7.225**			**3.2**			**7.6**

Table A4.1 – Decision matrix

4.1 Summary

As shown in this decision matrix, the tip extension design came in a close second to the flaps/slats design based on our decision matrix from the end of the fall quarter. When we talked about the details of both possible designs with Mr. Laguette, he suggested that the flaps/slats design would present significant problems over the course of the rest of the project, and that within the scope of this class, the tip extension might be a better way to go. Taking this advice into consideration, we made a decision as a team to go forward with the tip extension design. Although the tip extension scored lower on our decision matrix, the difference in scores was minimal enough that we feel confident that it is a very plausible design.

5 Patent Review

We have found several patents relating to retractable wind turbine blades that should be considered for review. Although there were several patents and patent applications that should be considered, we have condensed the list in order to touch upon the most relevant for this particular project. The patent number and a brief description of these patents are found below. The remaining patents that we have identified are listed in the Appendix.

US 6726439 – Retractable Rotor Blades

This patent is owned by Clipper Windpower. It utilizes concepts of retractable blades. This has positive impact on our design by facilitating the further development of our retractable rotor blade design concept.

US 6972498 – Variable Diameter Wind Turbine Rotor Blades

This patent is owned by GE, Clipper's competition. It also utilizes concepts of different types of retractable blades. This impacts our design because of the various designs patented for retractable blade deployment. If we realistically considering copyright infringement, this patent would be given very careful consideration.

US 6902370 – Telescoping Wind Turbine Blade

This patent utilizes concepts of telescoping turbine blades. This patent includes lead screw and pulley system tip extending mechanisms. This relates closely to Clipper's patent (6726439), and should be closely considered when pursuing our extending tip design.

US 5642982 – Retraction/Extension Mechanism for Variable Diameter Rotors

This patent utilizes concepts of retraction and extension of variable diameter rotors for helicopter blades. This patent relates to our project when considering the deployment of wind turbine rotor blades, since helicopter blades are of similar fashion to turbine blades.

US 7071578 – Wind Turbine Provided with a Controller

This patent is owned by Mitsubishi Heavy, Clipper's competition. It utilizes concepts for rotor blade extension and retraction. This has impacts on our design because Mitsubishi has a patent regarding how to extend and retract blades. Again, careful consideration should be made if we aim to not infringe any patents.

Conclusion

Although the above patents relate closely to our project, there did not seem to be any patent that matched our project exactly, though it is beyond our group's capacity to say that our design does not infringe on any of the claims stated in the above patents. Further, it is our groups understanding that we were to merely identify the patents that relate to our project. It would not be appropriate for our group to try and mitigate the claims found in these patents. We suggest that a qualified patent specialist from Clipper Windpower review the patents listed here and in the Appendix to verify whether or not any of them have been infringed upon with our design.

6 Testing Summary – Winter 2009

The testing for this stage of the project is divided into three parts, two of which have been completed, with tooling for the third part in the process of fabrication. The important design considerations to test are the ropes and the fiberglass attachment points. The ropes need to be tested in tension as well as tested for abrasion resistance. The fiberglass attachment methods are investigated through lap shear tests.

6.1 Rope Stretch Testing Results

The first set of tests involved loading a short rope sample in tension in order to determine the initial and post-load spring constants. Spectra (pink) and Vectran (tan) ropes were tested, and while Dyneema and Technora ropes are both under consideration, we were unable to obtain testing samples. The ropes were 1/4" diameter, which is scaled down from the full scale diameter of 1" for feasibility of tying knots and stretching the rope with the available equipment.

For 1/4" diameter, Spectra has a working load limit of 1700 lb and Vectran has a working load limit of 1500 lb. So as to not approach the work load limit they were loaded in tension to approximately 800 lb. The ropes were loaded and unloaded three separate times in order to have a comparison and establish whether the spring constant values changed after multiple cycles.

The first loading cycle was performed slowly to tighten the knots and take up slack in the rope and the test machine. This first load cycle stretched the ropes three times more than the second and third cycles did, indicating that after an initial loading, the rope would perform the same way each time.

The deflection of the rope was measuring by shining a laser at two pieces of reflective tape attached to the rope, which recorded the distance between the two pieces as the load was applied. The load vs. displacement data were used to calculate the initial spring constant, the post-load spring constant, and the stretch of the rope as a percentage of total length under a load equal to the work limit for that rope. These values are compiled in **Table A6.1.0** below, and the source graphs can be seen in **Figures A6.1.1 and A6.1.2** on the next page.

Rope Type	Initial Spring Constant (k_0)	Post-Load Spring Constant (k)	Percent Stretch at Work Load Limit
Vectran	$k_0 = 33{,}378 \frac{lb}{in}$	$k = 59{,}388 \frac{lb}{in}$	2.53% of total length
Spectra	$k_0 = 18{,}498 \frac{lb}{in}$	$k = 96{,}674 \frac{lb}{in}$	1.76% of total length

Table A6.1.0- Rope Stretch Test Results

Conclusions to be drawn from these data are that although Spectra has a lower initial spring constant, the post-load stiffness is much higher than Vectran. Of these two ropes, Spectra performs better, with a lower percent stretch. This is important because stretching of the rope under tension requires the winch mechanism to adjust its behavior to take up the slack. Given the choice of these two ropes, the tests support using pre-tensioned Spectra for the pulley system.

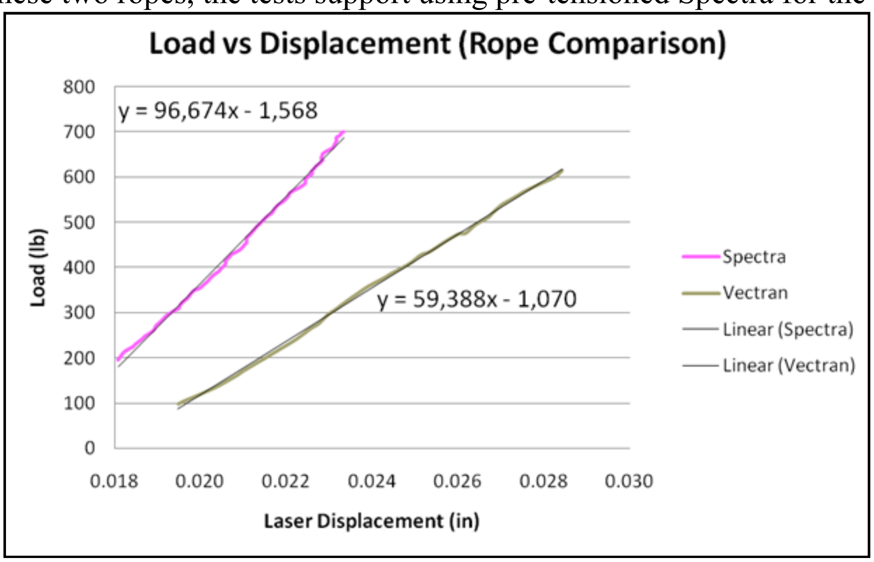


Figure A6.1.1- Rope Displacement vs. Applied Load

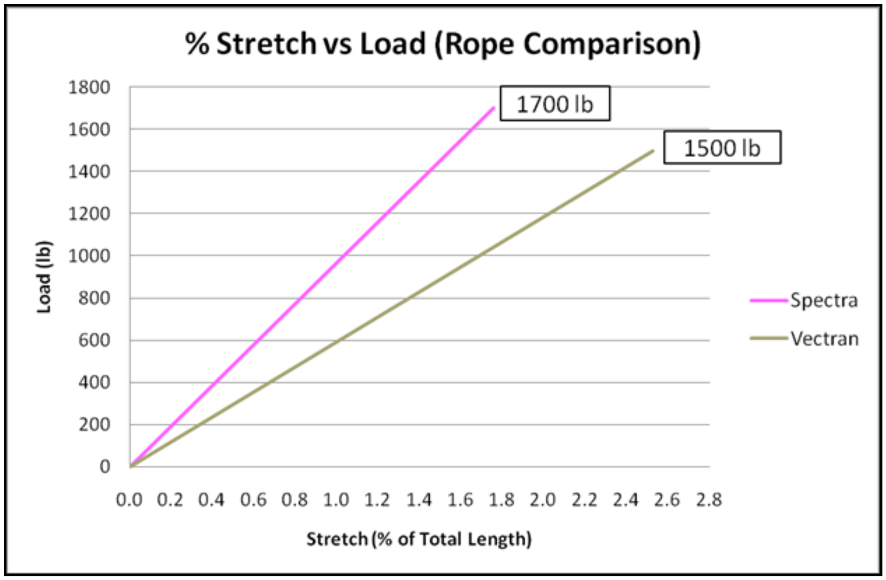


Figure A6.1.2- Rope Stretch vs. Applied Load

Figure A6.2.1- Rope Stretch Testing Apparatus

6.2 Rope Abrasion Testing (Future)

The second property of the rope to be tested is the abrasion or fray resistance. This is important to know because the lifespan of the rope depends on how much wear and tear it can stand up to before its strength is reduced. Rope/rope contact and rope/winch contact are low stress situations, but a situation which would cause a significant effect is having one of the pulley bearings freeze. If that occurs, the rope will slide around the pulley rather than roll, increasing the friction coefficient and causing a higher rate of wear.

There is a good deal of literature on rope manufacturer and marine applications websites concerning the durability of synthetic ropes compared with similar use of steel cables as discussed in the body of the report, but a test is in order to confirm the information found there.

A specialized fray testing tool will be fabricated for this test. The tool will be attached to a tension/compression test machine located in the testing lab with the assistance of Kirk Fields. The fray testing tool is a C-clamp with modifications for attachment of the rope. Holes will be drilled in the top of the clamp and the threaded shaft of the clamp, and the rope will be attached using those holes. The rope will be pre-twisted so as to minimize the twist in the rope when it is tensioned by the rotation of the threaded part of the clamp. Using a torque wrench and a modified socket that will accept the lever of the C-clamp, 500 ft-lb of torque will be applied to rope. The back of the C-clamp will be modified so that it can attach to the tension/compression machine. This tooling can be seen in the drawings in **Appendix 6.3.1**. A pulley will be attached to the moving shaft of the tension/compression machine, which will push into the rope to create a tension of 500lbs. The test machine will perform a set number of cycles of movement up and down, simulating movement of the rope along the pulley. The rope will be inspected after equally spaced numbers of cycles throughout the test, and any abrasion will be noted by visual inspection. Following the abrasion test, the results will be compared to the properties of steel cable and documented.

6.2.1 Fray Testing Setup

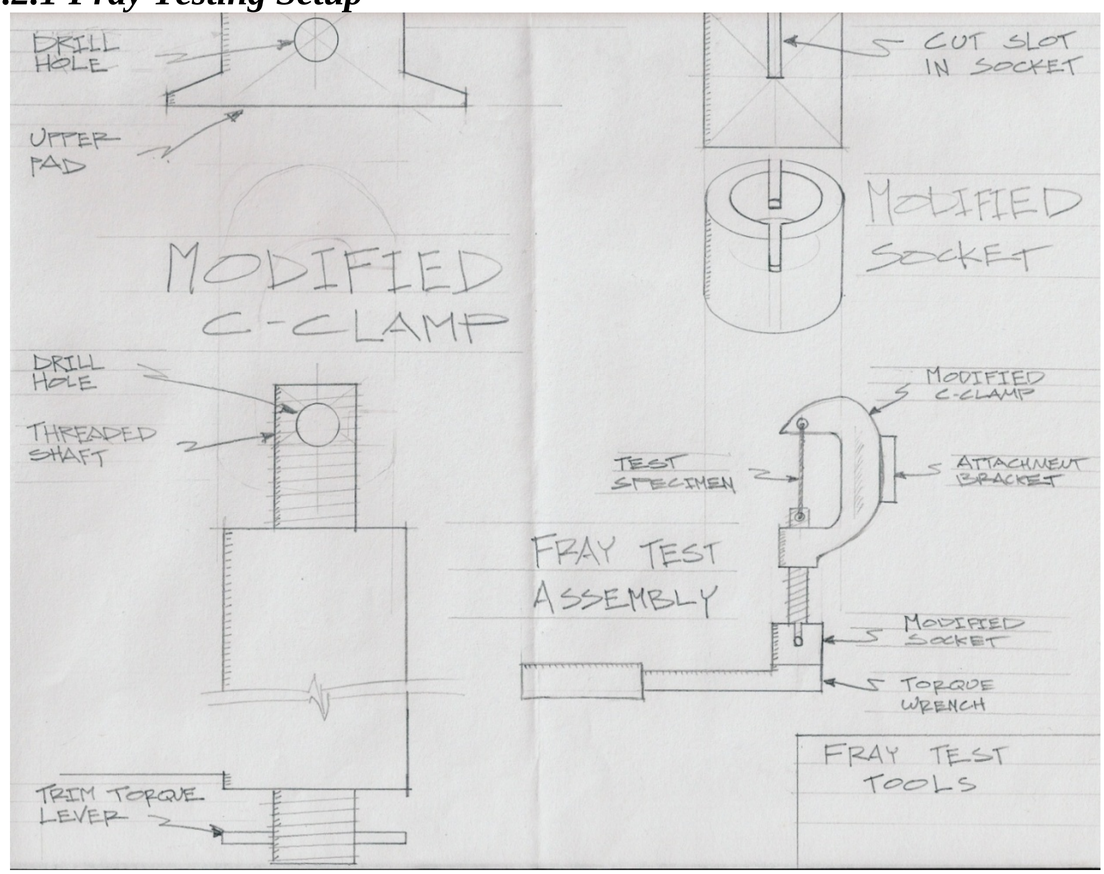


Table A6.3.1- Shows Fixture to be used for Fray Testing

6.3 Lap Shear Testing Results

Lap shear testing was performed to find out which of three bonding configurations would provide the strongest and most reliable joint between two pieces of fiberglass. Some parts of the prototype will have the bond type determined by physical space constraints, but others will be joined in the manner supported by this testing.

Three types of joints between fiberglass FRP samples were tested in single lap tension. The samples used a lap bond length of 1 in, and the adhesive used was Super Glue Plastic Fusion, rated to 4000 psi. The three joint types tested were adhesive bonding, bolting, and combined adhesive and bolting. Three samples were tested in each of these configurations.

The results for the adhesive bonding are shown in **Figure A6.4.1** below. These joints failed at relatively similar loads, with the adhesive peeling off of one or both samples. The important conclusion to draw from this set of tests is that the same failure mode was observed in all the samples, which means the weakness of the bond can be addressed. The results pointed to the adhesive itself not being as strong as had been hoped for, and in the construction of the prototype a more suitable adhesive will be used. Another suggestion for future adhesive bonding is to improve the bond surface by cleaning it more thoroughly and roughing the bond surface with sandpaper to create a more durable bond.

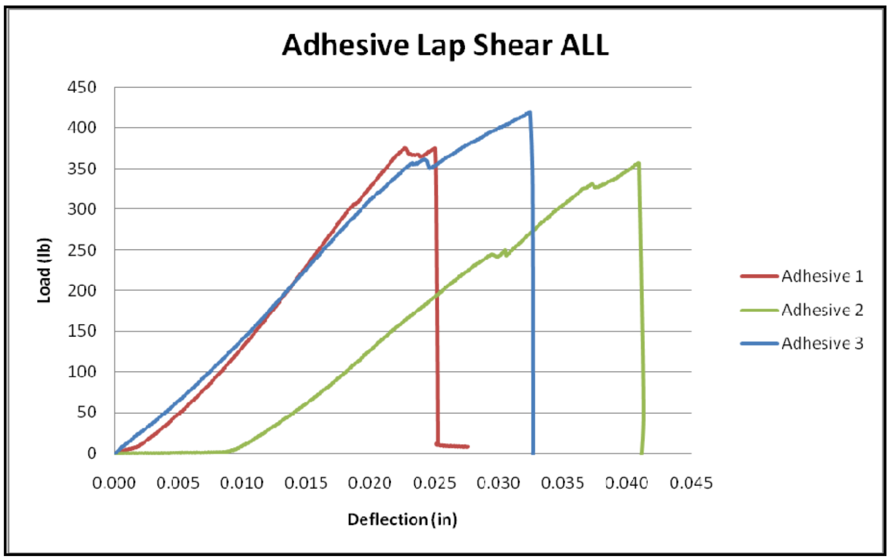


Figure A6.4.1- Lap Shear Testing – Adhesive Bonding

The next set of tests involved the bolted lap samples, with the results shown in **Figure A6.4.2** below. This method of attachment provided the most reliably high failure load, but unwanted damage to the samples at loads lower than the yield load. The first two samples caused compressive delamination failure in one side of the fiberglass. These samples cracked and split, which was the expected outcome of the test. The third test sample showed a different failure mode as the bolt broke in shear. It was unexpected that the two failure modes would have such similar values of maximum load. An improvement for this type of attachment is to use bolts which are smooth rather than threaded over the bearing surface, which greatly cuts down on stress concentrations in both the fiberglass and the bolt. Further gains would be made with higher precision in the drilling of bolt holes, since keeping the bolts level would reduce the stress on the outside edges of the fiberglass pieces.

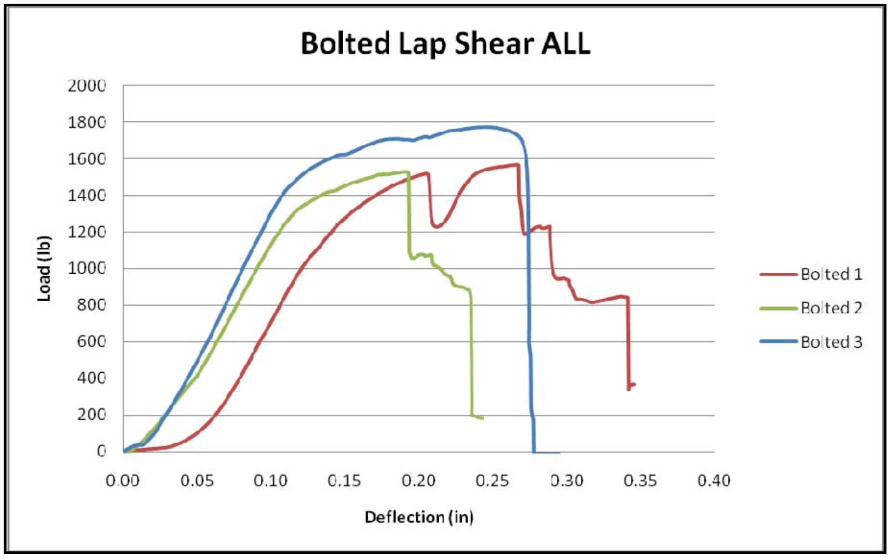


Figure A6.4.2- Lap Shear Testing – Bolting

The last set of test samples were both adhesively bonded and attached with bolts. This method provided much more stability for the bolts, but since the adhesive was much weaker than the bolts, the adhesive broke long before the joint became unstable. This can be seen as the sudden drop in deflection at around 500lb in **Figure A6.4.3** below. Having the bolts in place actually raised the breaking strength of the adhesive, but the bond strength was still far below the bolt break strength. The first two samples failed when the bolts broke in shear. Using adhesive as well as bolts apparently eliminated the possibility of compressive failure in the fiberglass, probably due to the lack of wiggle room around the bolts. The third sample unexpectedly failed at a much lower load when the fiberglass simply cracked around the bolt hole. Upon inspection, this was attributed to cutting that sample in the wrong direction, so the fibers were lined up perpendicular to the load. This provides a reminder that care must be taken in construction of the prototype to align the fiberglass properly in order to take advantage of its full strength.

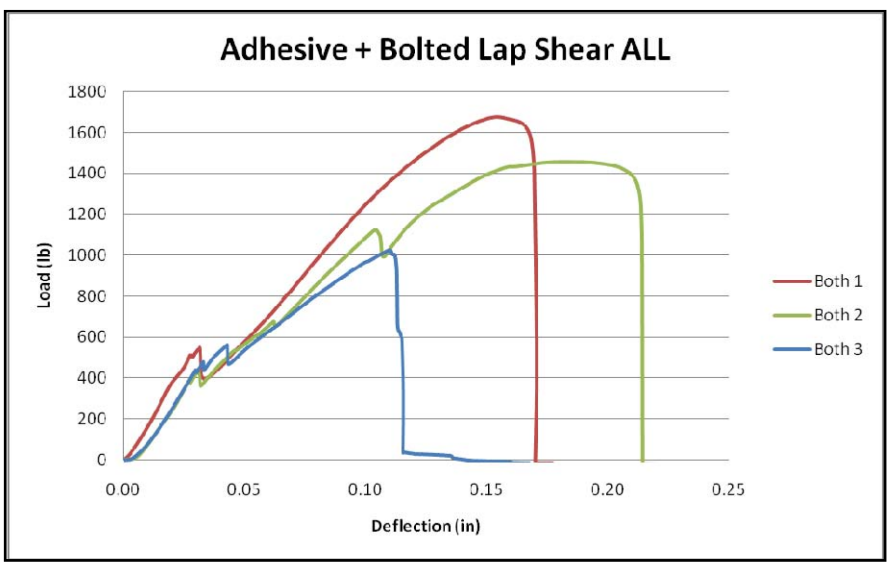


Figure A6.4.3- Lap Shear Testing – Adhesive Bonding and Bolting

The recommended joint type is a combination of bonding and bolting where possible. A much stronger adhesive is necessary, and more care will be taken in preparing the bond surface. Where bolting is used, the bearing surface of the bolts will be smooth rather than threaded. Although the testing results support bolts alone being the strongest joint, they do not provide stability and stiffness, which are important factors for the prototype. Combined bolting and bonding will create a high strength bond that maintains maximum stiffness.

6.4 Lap Shear Testing Pictures

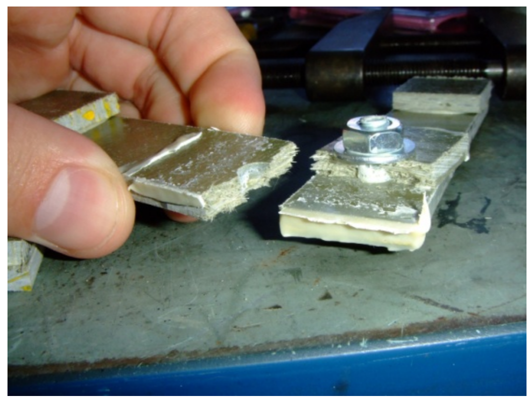


Figure A6.5.1- Lap Shear Testing Setup and Broken Test Sample

7 Fiber Rope Considerations

7.1 Introduction

The diameter rope for each case is based off of the max loading case that can be seen in the Design Restrictions section of the Extension Mechanism Report. The max loading case is located at the bottom of the windmills rotation of 15 RPM while the blade is being retracted. The force required to overcome an acceleration of 12 g-forces with a tip extension mass of 600 kg is approximately 70,000 N or 15,736 lbs (4.4482 $\frac{N}{lb}$). With a factor of safety of 4 and a required max working load of 15,736 lbs, equation **(1)** was used to calculate the required breaking strength of the fiber rope.

$$Working\ Load = \frac{Minimum\ Breaking\ Strength}{Factor\ of\ Safety} \quad \textbf{(1)}$$

This provided us with a minimum breaking strength of approximately 63,000 lbs.

7.2 Effect of Temperature on fiber Ropes

All of the values for strength shown on the fiber rope charts in **Appendix 7.5** are valid for room temperature. As the temperature is increased, the tensile strength of the fiber rope decreases. This decreases the load capabilities of that rope creating safety concerns.

The temperature inside the blade is not much higher than the ambient temperature. Even for wind turbine installations in hot areas, the loss of tensile strength due to temperature effects is not significant enough to reduce the working load enough to surpass the FOS, but in situations where high temperature conditions are a concern, Technora ropes would be used. Since the critical temperature for Technora is 400 °F, there would have to be a large amount of unforeseen heating before any loss of strength would occur.

7.3 Mechanisms of rope attachment

Rope attachment is necessary to understand and consider because both ends of the rope must be securely terminated in order for our design to fail. A simple failure by a lack of attachment could easily occur if not carefully considered.

7.3.1 Rope Splicing

Rope splicing is the forming of a semi-permanent joint between two ropes or two parts of the same rope by partly untwisting them and then interweaving the loose strands. Some applications for rope splicing include forming a stopper at the end of a line, forming a loop or eye in a rope, or joining two ropes together. Splices are preferred to knots since knots can reduce the strength of the rope up to 40% while some splices can retain up to 95% of the strength of the line. An example of a splice is shown in **Figure 3.5** below. The places in the mechanism where a splice might be appropriate are at the ends, where the ropes need to be attached to fixed points on the tip extension.

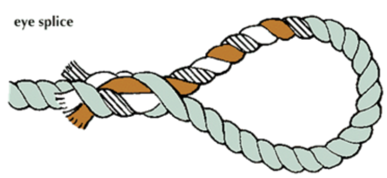


Figure A7.2.1 - Eye splice is the most commonly used rope splicing technique

7.3.2 Knots

A knot is a method for fastening or securing linear material such as rope by tying or interweaving. Knots have a wide range of possible applications but some problems do exist. Knots invariably weaken the rope they are made in, and when knotted rope is strained to its breaking point, the most common failure point is in or near the knot unless it is defective or damaged elsewhere. Knot slippage occurs when tension is present, which causes the rope to work back in the direction of the load. One possible reason for using knots rather than splicing is that they require less specialized skilled labor and can be completed more quickly, however the ropes in the extension and retraction mechanism are meant to last a long time, so the extra effort of splicing rather than knotting is worth the time expenditure in the long run.

7.3.3 Composite Termination

Composite termination is a fairly recent development in cable termination for tightly controlled cable assemblies. Composite termination works by using a terminal with a hollow, expanding internal cavity being placed around the end of the rope and filled with a high-performance resin. The terminal and rope are shown in **Figure 3.6** below. This type of termination outperforms traditional fittings in breaking strength, fatigue resistance, length stability, and performance repeatability. Connective hardware can be attached to the terminal for various applications. It would be reasonable to use this type of connector to attach the extension and retraction ropes to the tip extension. Some rope manufacturers have an option to purchase lengths of rope with this terminal already in place, and the connector hardware could be varied depending on the point of attachment.

Figure A7.3.1 - An example of a composite termination of a fiber rope

7.4 Benefits

There are many benefits of choosing synthetic fiber ropes rather than steel cables which support our decision to use synthetic rope for the extension and retraction mechanisms. High performance synthetics like Spectra and Dyneema are up to 10 times stronger than similar diameter steel cable, and they are also much lighter than steel. Most synthetic ropes have a specific gravity of less than 1 allowing them to float in water. The weight is a significant factor, and having a rope that is much lighter than steel cable while maintaining very high strength allows our design to keep its weight reduction to a minimum.

Synthetic ropes have very good anti-corrosive properties, and are commonly used in marine environments. This would not be an issue for most wind turbines, but some wind farms have been installed in the ocean, where it is an important consideration. Polymer ropes offer exceptional bending and tensile fatigue resistance, high abrasion resistance, extremely low stretch, and low or no creep. They also are very resistant to kinking, unlike steel cable. These properties are important for this mechanism because they provide a high degree of confidence in the long term performance of the rope.

7.5 Polymer Rope Considerations:

In investigating the possible ropes for this design, four types of synthetic fiber were found that have properties which would be suitable for the extension and retraction mechanisms. Spectra and Vectran ropes were tested in the mechanical testing lab, and the other two were not available in lengths of less than 1200 ft. Tables of the weight and strength of each type of rope are shown below.

For the full-scale design implementation, we recommend using either Dyneema or Technora ropes because they have optimal combinations of properties needed for this design. However, the prototype will use Spectra rope because it is easily obtained in small lengths, and can still support heavy loads.

7.5.1 Spectra 12-Strand

Spectra is an extremely strong fiber rope that has a higher breaking strength than steel wire rope of similar diameter at one tenth of the weight. Spectra fiber rope has good UV, seawater, abrasion, cutting, fatigue, and stretch resistance, and is used in marine, industrial, utility, rescue, and commercial fishing applications.

Size			Approx Weight		Min Tensile Strength		Approx. Avg. Tensile Strength	
Dia. Inches	*Dia. MM*	*Circ. Inches*	*Lbs./100 ft*	*KG/100 ft*	*Lbs.*	*KG*	*Lbs.*	*KG*
1/8"	3	3/8"	0.53	0.8	1620	735	1800	817
3/16"	5	9/16"	1.04	1.5	3420	1550	3800	1725
1/4"	6	3/4"	1.7	2.5	5900	2690	6600	2996
5/16"	8	1"	2.9	4.3	8800	4000	9800	4449
3/8"	10	1-1/8"	3.8	5.7	12690	5760	14100	6401
7/16"	11	1-1/4"	4.3	6.4	14850	6740	16500	7491
1/2"	12	1-1/2"	6.4	9.5	22500	10200	25000	11350
9/16"	14	1-3/4"	7.5	11.2	27720	15600	30800	13983
5/8"	16	2"	10.6	15.8	36600	16630	40700	18478
3/4"	18	2-1/4"	13.2	19.7	43200	19600	48000	21792
7/8"	22	2-3/4"	19.5	29.0	61000	27700	67800	30781
1"	24	3"	23.3	34.7	72000	32700	80000	36320
1-1/8"	28	3-1/2"	32.0	47.7	91800	41700	102000	46308
1-1/4"	30	3-3/4"	36.1	53.8	102600	46580	114000	51756
1-5/16"	32	4"	41.8	62.3	114300	51890	127000	57658
1-1/2"	36	4-1/2"	51.7	77.0	141300	64150	157000	71278
1-5/8"	40	5"	65.8	98.0	167400	76000	186000	84444
1-3/4"	44	5-1/2"	78.3	116.6	198000	89800	220000	99880
2"	48	6"	91.4	136.1	225000	102150	250000	113500

Table A7.5.1- Table of properties for Spectra 12-strand fiber rope.

7.5.2 Vectran 12-Strand

Vectran braided 12-strand is a strong fiber rope that is well known for its high strength, low stretch, virtually no creep, soft and easy handling, and its ability to be easily spliced. Vectran rope is commonly used for steamer cables, mooring line, tow line, and winch line applications.

PART NUMBER	BREAK STRENGTH LBS	kN	DIAMETER IN	mm	WEIGHT LBS/1000 FT	kg/km
BR 9500V	9,500	42	1/4	6.4	21.5	32
BR 13000V	13,000	58	5/16	7.9	30.8	46
BR 18500V	18,500	82	3/8	9.5	43.1	64
BR 24000V	24,000	107	7/16	11.1	58.5	87
BR 31000V	31,000	138	1/2	12.7	77	115
BR 38000V	38,000	169	9/16	14.3	99	147
BR 47000V	47,000	209	5/8	15.9	120	179
BR 68000V	68,000	302	3/4	19.1	166.2	247
BR 10500V	105,000	467	1	25.4	301.6	449

Table A7.5.2- Table of properties for braided Vectran 12-strand fiber rope.

7.5.3 Dyneema

Dyneema is an extremely strong polyethylene fiber that provides maximum strength and low weight. It is up to 15 times stronger than steel and 40% stronger than other aramid fibers, and has very low creep and stretch properties. Dyneema is an important component of ropes, cables, and nets in the fishing, shipping, and offshore industries due to its high resistance to moisture, UV lights, and other chemicals. It is also used commonly in bullet resistant armor for both military and police personnel.

PART NUMBER	BREAK STRENGTH LBS	kN	DIAMETER IN	mm	WEIGHT LBS/1000 FT	kg/km
BR 9500PE	9,500	42	1/4	6.4	14.6	22
BR 13000PE	13,000	58	5/16	7.9	22.6	34
BR 18500PE	18,500	82	3/8	9.5	34.7	52
BR 24000PE	24,000	107	7/16	11.1	40	60
BR 31000PE	31,000	138	1/2	12.7	53	79
BR 38000PE	38,000	169	9/16	14.3	67	100
BR 47000PE	47,000	209	5/8	15.9	84	125
BR 68000PE	68,000	302	3/4	19.1	124	185
BR 105000PE	105,000	467	1	25.4	215	320

Table A7.5.3- Table of properties for Dyneema braided 12-strand fiber rope.

7.5.4 Technora

Technora is a para-aramid fiber that has a high tensile strength, high stiffness, high abrasion resistance, and excellent resistance to heat and chemicals. Technora works very well around sheaves and turning blocks and is commonly used in rubber reinforcement, ropes, protective goods, cement and plastic reinforcement, and many other industrial applications.

PART NUMBER	BREAK STRENGTH LBS	kN	DIAMETER IN	mm	WEIGHT LBS/1000 FT	kg/km
BR 10000T	10,000	45	1/4	6.4	21.5	32
BR 14000T	14,000	62	5/16	7.9	30.8	46
BR 20000T	20,000	89	3/8	9.5	43.1	64
BR 26000T	26,000	116	7/16	11.1	58.5	87
BR 35000T	35,000	156	1/2	12.7	77	115
BR 42000T	42,000	187	9/16	14.3	99	147
BR 52000T	52,000	231	5/8	15.9	120	179
BR 77000T	77,000	343	3/4	19.1	166.2	247
BR 125000T	125,000	556	1	25.4	301.6	449

Table A7.5.4- Table of properties for Technora (Aramid) braided 12-strand.

8 Rope Fray Issues

Failure due to fraying is a concern with using synthetic rope as opposed to steel cable in the pulley system of the tip-extension mechanism design. While the strength to weight ratio for the synthetic rope is far superior to the steel cable, the rope is not as resistant to abrasion (or fraying) when compared to steel cable. It is important to address the abrasion resistance of the synthetic rope as it will determine the viability of our design.

Three approaches were considered to study the abrasion resistance of synthetic rope. The first approach was to search the manufacturer's data sheets, website and independent tests that have been performed regarding synthetic ropes. The second approach is to find existing applications for synthetic rope that correlate closely to the project needs. This will validate the decision to use synthetic rope in the design as well as provide another set of supporting data regarding the use of synthetic rope over steel cable in those applications. The final approach in determining the abrasion resistance in the synthetic rope will be to perform abrasion testing for the rope, compare the strength to steel cable and then analyze the results to determine whether or not the rope is a viable solution for the design.

8.1 Approach 1

DSM Dyneema was selected as the synthetic rope manufacturer to study. DSM Dyneema is the leading synthetic rope manufacturer who claims to have the strongest synthetic rope in the world. While there was abundant information about the tensile strength of their rope, information about the abrasion resistance was limited. The manufacturer stated that their synthetic rope "has good abrasion and cutting resistance"[1] and "Dyneema fiber demonstrates 10 times higher abrasion resistance than traditional net materials"[2].

[1] http://www.dsm.com/en_US/html/hpf/industrial.htm?source=search
[2] http://www.dsm.com/en_US/html/hpf/industrial.htm?source=search

Also posted on Dyneema's website was "And in contrast to steel wire, which can fray and leave sharp edges, slings with Dyneema have a very smooth surface."[1] This comparison to steel wire is important in validating the claim that synthetic rope is a viable solution for the design. Since steel wire would leave sharp edges (referred to as "fish hooks" in the industry) once it frays, the chances of critical failure increases. This is due to the sharp edges of the frayed steel wire that could damage the composite pulley system, the track system, or the structure of the blade of the wind turbine itself. In contrast, synthetic rope would not leave sharp edges if it frayed, thus reducing the chances of critical failure.

The medical industry has also incorporated the use of Dyneema into many of its medical instruments. According to DSM Dyneema's website, "Dyneema does not break even if bent and folded thousands of times, and is highly resistant to abrasion."[2] The use of Dyneema's synthetic material in medical devices is yet another validation for the robustness of the product.

In a DSM Dyneema press release found on their website, it states that "[cut resistant gloves] can be up to 25 times more abrasion resistant than gloves made with aramids."[3] Along with the given abrasion resistant statistic of Dyneema being 25 times more abrasion resistant than aramid (aromatic polyamide) fibers, it is important to note that the product utilizing the synthetic material is a cut resistant glove. The use of this material in a cut resistant glove supports the decision to use synthetic rope in the pulley system design because the necessary design considerations of making a cut resistant glove are directly in line with the pulley system design considerations. The material used for both designs (cut resistant gloves and our pulley system) must be a very robust, abrasion resistant material.

The abrasion resistance of Dyneema was compared to all other fibers. According to DSM Dyneema's website, Dyneema's fibers were found to experience "2.5 to 8 times lower dry abrasion and 1.5 to 40 times lower wet abrasion than all other fibers."[4]

An independent study called "Residual strength testing of Dyneema, Fibre Tunglines"[5] studied AmSteel – Blue rope made from Dyneema SK75 fiber. The study focused on the rope behavior in both the field and in laboratory simulations. The study examined 40 separate brake samples of the material of various rope diameters that were actively used in the field aboard tugboats in vessel escort service. The samples were tested at certified, independent testing facilities.

The part of the study that is of interest to the project is the portion which covers abrasion and cutting damage. According to the study, most of the ropes that were tested were used in the field as a replacement for steel wire. As such, the synthetic ropes were measured against that datum. In the study, it was stated that "The obvious fact is that strength of any rope will degrade from external factors…The best compromise is to assure maximum strength over the longest possible period. This is best accomplished through proper application, due care and protection." The study concluded the following:

[1] http://www.dsm.com/en_US/html/hpf/industrial.htm?source=search
[2] http://www.dsm.com/en_US/html/hpf/applications_medical_devices.htm?source=search
[3] Dyneema: Bringing comfort to cut resistant gloves, DSM Dyneema Press Release, http://www.dsm.com/search/public/result.do?pagestart=4&branding=hpf&locale=en_US&entitlement=10&strongendorsed=true&docscount=0&sortby=score&stemming=false&within=this&language=en&querytext=abrasion
[4] http://www.dsm.com/en_US/html/hpf/dozens_of_reasons.htm?source=search
[5] Residual Strength testing of Dyneema, Fibre Tunglines, Phil Roberts and Danielle Stenvers, Samson Rope Technologies; Paul Smeets and Martin Vlasblom, DSM High Performance Fibers, 2002.

Figure A8.1- Synthetic Rope Tow Line

"Abrasion and cutting damage has averaged 5-10% wear (total internal and external abrasion), which may account for a strength loss of 5-10%. It has now been determined that compression from the drum accounts for a strength loss of 10-12%. Several lines that were tested had a moderate to severe twist, up to 1.5 turns per foot, which resulted in damage to about 10% of the total fiber in the rope. Line twist of 1 to 1.5 turns per foot equates to a 15-20% strength reduction. Abrasion and compression alone can account for 15-20% strength loss. If the line has also been twisted, the combination of these three factors could account for up to 40% strength reduction."

This part of the study's conclusion suggests that wear due to abrasion and compression could reduce the strength of the synthetic rope by as much as 20% and, in the event the rope is also twisted, the strength could be reduced by as much as 40%. The synthetic rope considered should have a factor of safety for strength of at least 6 (required to hold a force of ~16,000 lbs and contains a breaking strength of 105,000 lbs @ 1 inch diameter).

The study also concluded the following:

"Degradation of the fiber does not appear to be the contributing factor to the strength loss of the main lines (synthetic rope). Samples of used ropes have been sent to DSM-HPF for analysis and the strands and yarns taken from the tug lines show almost no abrasion damage."

"The testing performed by both DSM-HPF and Samson Rope Technologies indicates that Dyneema fiber has excellent resistance to cyclic fatigue, even when tests are performed well in excess of the OCIMF TCLL cycle times. The resistance of Dyneema to both high magnitude loads and an extensive number of load cycles has been proven in laboratory testing. The resistance of Dyneema to much higher strain rates has also been proven on yarns."

The conclusions found in this study validate not only the abrasion resistance of Dyneema's synthetic rope, but also validate its strength. The study began with stating that the best compromise to using synthetic rope in lieu of steel wire is to assure maximum strength of the synthetic rope over the longest possible period. This study concluded that Dyneema's synthetic rope accomplished that goal.

8.2 Approach 2

After studying the manufacturer's website and independent studies about the manufacturer's synthetic rope product, existing uses of the synthetic rope were studied. It was found that Dyneema's synthetic rope is currently being used in winches attached to off-road vehicles, tow cables that connect gliders to powered aircraft, fishing nets and yachting lines.

The existing use of synthetic rope that is perhaps most closely related to our project's use is in winches attached to off-road vehicles. According to HiMFR.com, "[Stainless steel] Wire cables on your winch will fray, kink and get impossibly tangled on your winch drum. These cables are extremely dangerous, and not just to your hands. Wire cables retain a great deal of energy and a cable that snaps under load becomes a high speed knife with an unpredictable path. Synthetic winch rope, however, will not kink or tangle and retains very little energy. If you cut our synthetic winch rope while it is under load it simply falls to the ground."[1] This quote brings up another advantage to Dyneema's synthetic rope over steel wire and that is the fact that steel wire contains a lot of energy when it is under load. If the steel wire breaks, the energy is

[1] http://www.himfr.com/d-p113208739860144600-Dyneema_Electric_Winch_Rope-The_Wire_Rope_Best_Replacement/

released as the cable flies uncontrollably through the air. This critical failure characteristic would be devastating to a windmill. Since Dyneema's synthetic rope would simply fall to the ground if it were to fail under load, the rope would cause minimal damage, if any, to a windmill. This is another validation for using Dyneema's synthetic rope in the design.

Another interesting current use for DSM Dyneema's synthetic rope is for glider tow cables that connect the glider to a powered aircraft. After the glider is connected to the powered aircraft, the powered aircraft pulls the glider down the runway and takes off with the glider in tow. Once the glider has been pulled to a specified altitude, it is released. According to stratfordgliding.com "[Dyneema synthetic rope] lasts much longer, and is highly resistant to abrasion…we've seen a rope that's done 3000 launches on grass, and it looks like new, so it's quite possible that it will last five times as long as steel [wire]."[1]

Figure A8.2- Winch with Synthetic Rope

Dyneema synthetic rope is also used in fishing nets and yacht lines. This is pertinent to our design as both fishing nets and yacht lines will run the synthetic rope through a pulley system. In addition to the abrasion the synthetic rope will experience through their respective pulley systems, both fish nets and yacht lines are used in the harshest and most corrosive environment on Earth, the ocean. As our design is for Clipper Windpower, who installs wind turbines both on land and on the ocean, it is important that the materials we specify in our design be resistant to the elements found in those environments.

The existing uses for Dyneema synthetic rope listed above are just a few of the uses available. The above applications were chosen because they related most closely to our project. In addition to the above uses being similar to our design, they also necessitate a high level of confidence in the performance of the synthetic rope. Each use listed above involved the safety of a human being. Even though the safety of a human being was a consideration, Dyneema synthetic rope was still preferred over and considered safer than steel cable.

8.3 Approach 3

The final approach in validation of the decision to use synthetic rope rather than steel cable in this design is to conduct fray testing of synthetic rope and compare the results to those of steel cable. The test has been designed, and will be conducted at the beginning of the 2009 spring quarter. A diagram of the test setup can be found in **Appendix 6.3.1**.

A specialized fray testing tool will be fabricated for this test. The tool will be attached to a tension/compression test machine located in the testing lab with the assistance of Kirk Fields. The fray testing tool is a C-clamp with modifications for attachment of the rope. Holes will be drilled in the top of the clamp and the threaded shaft of the clamp, and the rope will be attached using those holes. The rope will be pre-twisted so as to minimize the twist in the rope when it is tensioned by the rotation of the threaded part of the clamp. Using a torque wrench and a modified socket that will accept the lever of the C-clamp, 500 ft-lb of torque will be applied to rope. The back of the C-clamp will be modified so that it can attach to the tension/compression machine. This tooling can be seen in the drawings in **Appendix 6.3.1**. A pulley will be attached to the moving shaft of the tension/compression machine, which will push into the rope to create a tension of 500lbs. The test machine will perform a set number of cycles of movement up and down, simulating movement of the rope along the pulley. The rope will be inspected after

[1] http://www.stratfordgliding.co.uk/WR200507.HTM

equally spaced numbers of cycles throughout the test, and any abrasion will be noted by visual inspection. Following the abrasion test, the results will be compared to the properties of steel cable and documented.

8.4 Conclusion

After conducting the three approaches to studying the abrasion resistance of synthetic rope, our group feels, pending the results of our testing, confident in the decision of selecting synthetic rope for the design. Along with the vastly increased strength to weight ratio, it is evident that the synthetic rope will perform to the design requirements. While synthetic rope may not be as resistant as steel cable in the abrasion test, the mechanism will be designed around this weakness. It was found that if a minimum factor of safety for strength of 6 is employed, the synthetic rope should not break based on the results of the independent study. The current uses for synthetic rope also corroborate the decision to use synthetic rope in the design. Not only is the synthetic rope being applied to pulley systems in harsh environments, but human lives are also dependent the synthetic rope's performance. This is a huge indication of the viability of the synthetic rope's design applications. Also, through the study of existing uses, it was found that there are additional benefits of using synthetic rope over steel wire. The most significant benefit is the fact that synthetic rope will not store copious amounts of energy when it is under loading like steel cable would. This is important because should the cable fail inside the design, the synthetic rope will simply fall with gravity while the steel cable will lash out uncontrollably and cause a devastating amount of damage to the design. Pending the fray test results, our group feels confident in moving forward with synthetic rope in the design.

9 Rod Locks

Rod locks are commonly used as linear braking and locking devices. Some of the most common applications for rod locks include integration into inspection and transfer equipment, packaging machinery, and machine tools. Rod locks are high performance, spring-engaged, air-released units that are mostly used in emergency situations. An advantage is that they can be used in power-off situations with multiple springs providing redundancy in the system. Many of the models are also sealed to withstand harsh environments. This is vital since windmills, especially those located in the ocean, consistently face harsh environments. The locks also provide very high clamping forces and extended engagement capabilities with a very minimal amount of air-pressure required in order to release the locks. The most common rod locks are the cylinder mounted and stand-alone types. These two models can be seen in **Figure A10.1**.

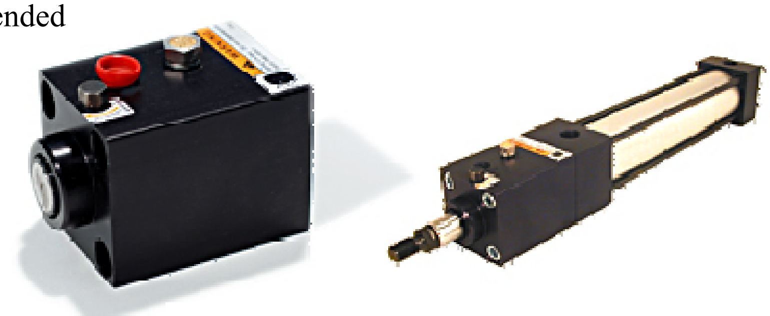


Figure A9.1- Pictures of a cylinder mounted rod-lock (left) and a stand-alone rod-lock (right)

For the extension/retraction mechanism, rod locks are a solid choice for an emergency braking system in the event of a critical failure in one of the ropes. Although this critical failure is very unlikely due to the very high factor-of-safety applied to the polymer fiber rope (approximately a FOS of 8 @ 1" polymer fiber rope), it is very important for an emergency locking mechanism to exist. Not only will this prevent the windmill from catastrophic failure if a rope fails, it will also allow for maintenance on parts of the tip extension and the mechanism. In order for these rod locks to be viable as an emergency locking mechanism on the tip extension design, there would need to be several of them in order to hold the tip extension at the maximum loading case, which is spinning at 15 RPM with the tip fully extended and in the downward position of its rotation (~15,000 lbs with no factor-of-safety). Having multiple rod-locks for the tip extension would also provide necessary redundancy, since a system failure could lead to a complete failure of the entire windmill, which would be both dangerous and expensive.

10 FEA Analysis

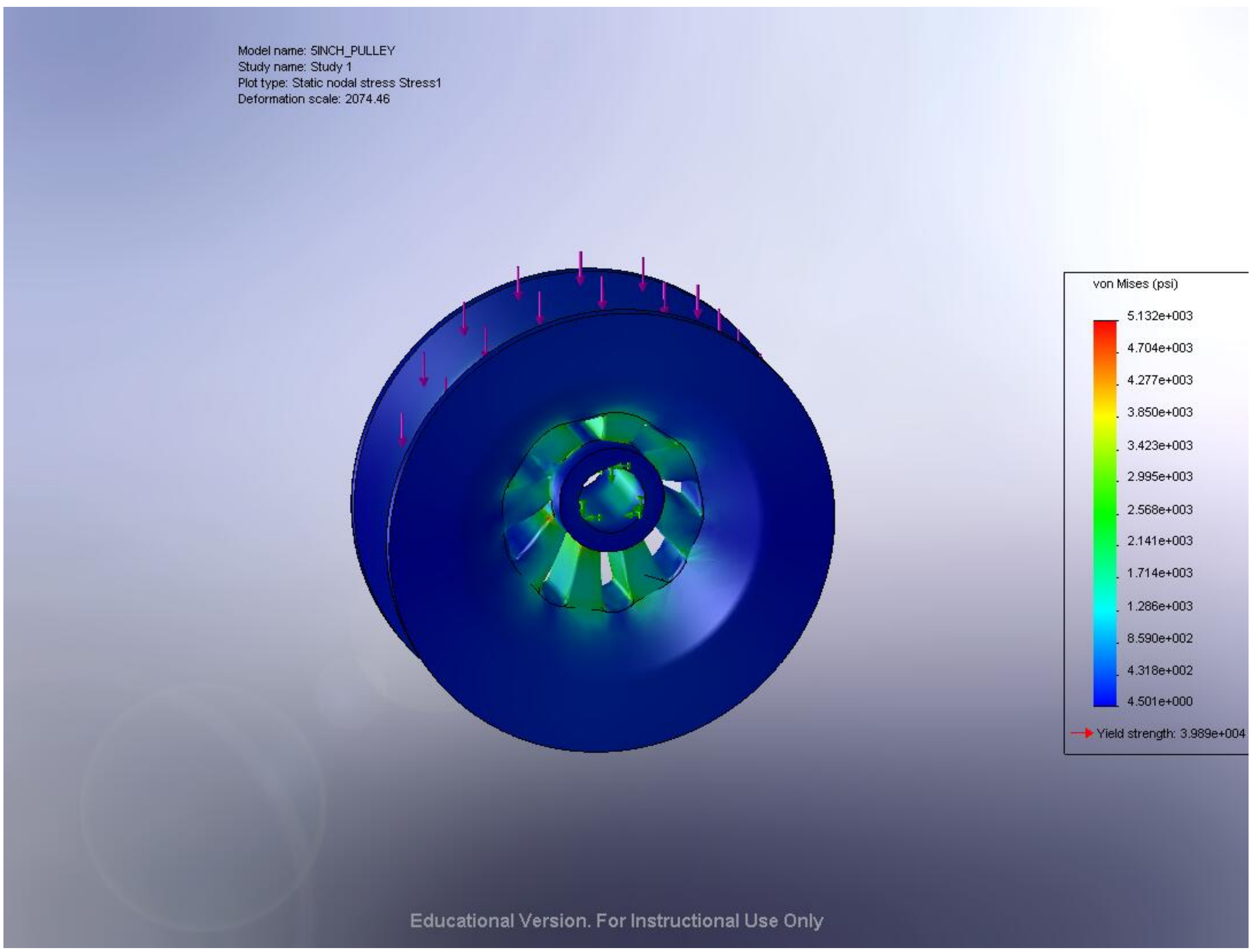


Figure A10.1- Shows 5" pulley stress distribution and scale (psi) under loading conditions

Conclusion

As shown above in the stress plot, the 5" pulley displayed a max stress of 5.132×10^3 psi and the yield strength is 3.989×10^4 psi. This results in a minimum FOS of 7.75 on the pulley. Due to this result it has been proven that the pulley will not fail under normal loading conditions.

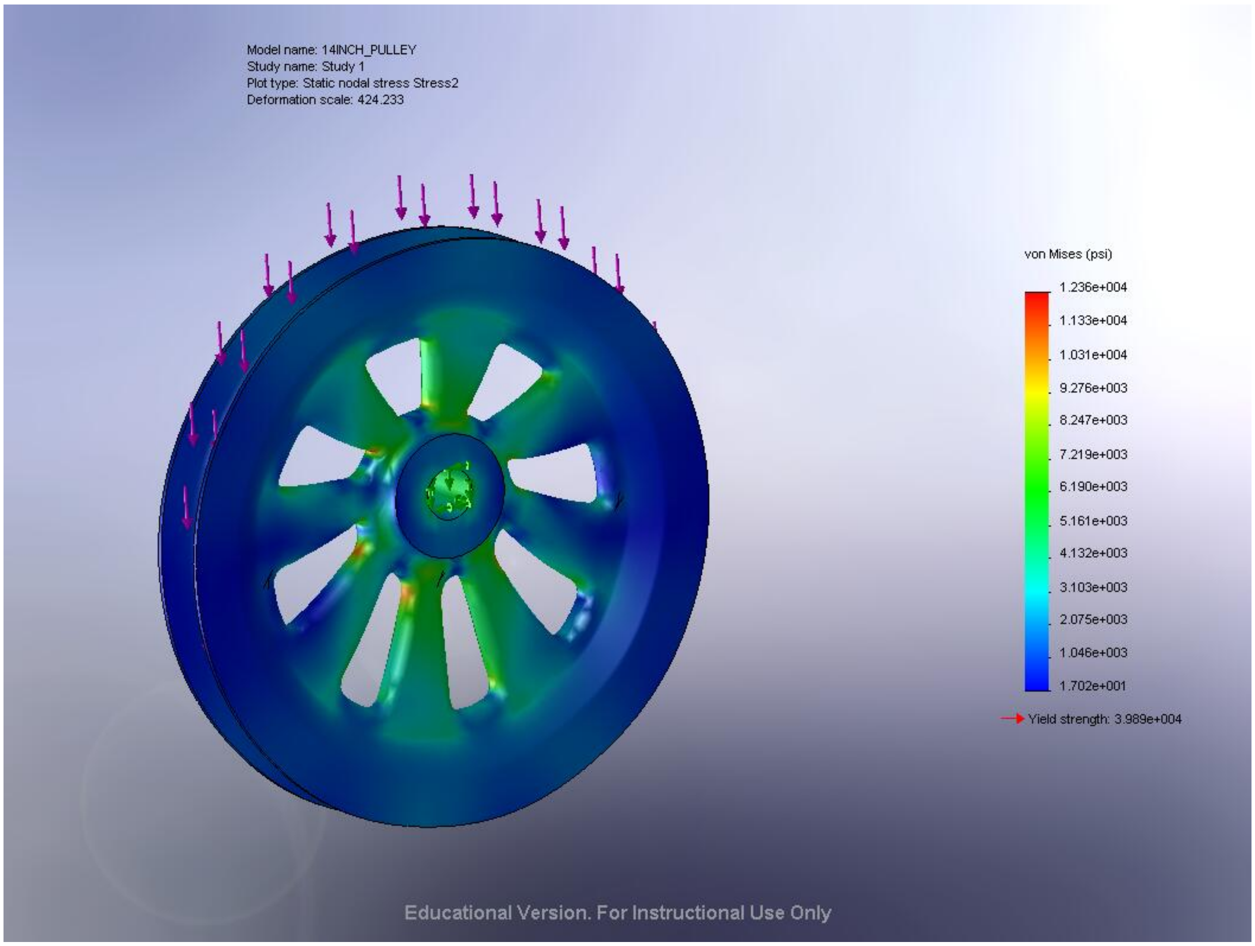


Figure A10.2- Shows 14" pulley stress distribution and scale (psi) under loading conditions

Conclusion

As shown above in the stress plot, the 14" pulley displayed a max stress of 1.236×10^4 psi and the yield strength is 3.989×10^4 psi. This results in a minimum FOS of 3.23 on the pulley. Due to this result it has been proven that the pulley will not fail under normal loading conditions.

11 Parts List for Prototype

11.1 Track System

Track – 4X – FRP fiberglass C-channel

11.2 Pulley system

Motors – 2X – 1 large (retraction), 1 small (extension)
Winches – 2X – 1 large (retraction), 1 small (extension)
Gearing – unknown quantity and type
Rope – 2x – ≈11m retraction, ≈8m extension
– Ropes are ¼" diameter 12 strand synthetic fiber ropes, probably Spectra brand
Pulleys – 4X
– 2X extension pulleys
– 1X extension pulley (perpendicular) – attached to box beam
– 1X extension pulley (parallel) – attached to tip extension
Triangle bracket – 1X – attachment for the "Y" of the extension system

11.3 Base Blade

Box beam – 1X – FRP fiberglass, 1.5m long, width and height TBD
Ribs – at least 4X – FRP fiberglass laid in the airfoil shape
Stringers – at least 8X – FRP fiberglass (C-channels?), 4 of which are concurrent with tracks

11.4 Extension Blade

Forks – 2X – FRP fiberglass structural members, 3m long
– C-channels or I-beams, width and height TBD
Base – 1X – FRP fiberglass plate for pulley attachment
Skin – 1X – FRP fiberglass laid in airfoil shape over the forks
Rollers – at least 8X – pocket bearings

11.5 Miscellaneous (unknown quantities and types)

Resin – for molding fiberglass to desired shapes
Epoxy/other adhesive – attachment for fiberglass to fiberglass or fiberglass to other material
Fasteners – chicken bolts (backup for epoxy), fasteners for all metal pieces
Rod locks – at least 4X – failsafe backup system (if employed)

12 Prototype Pictures

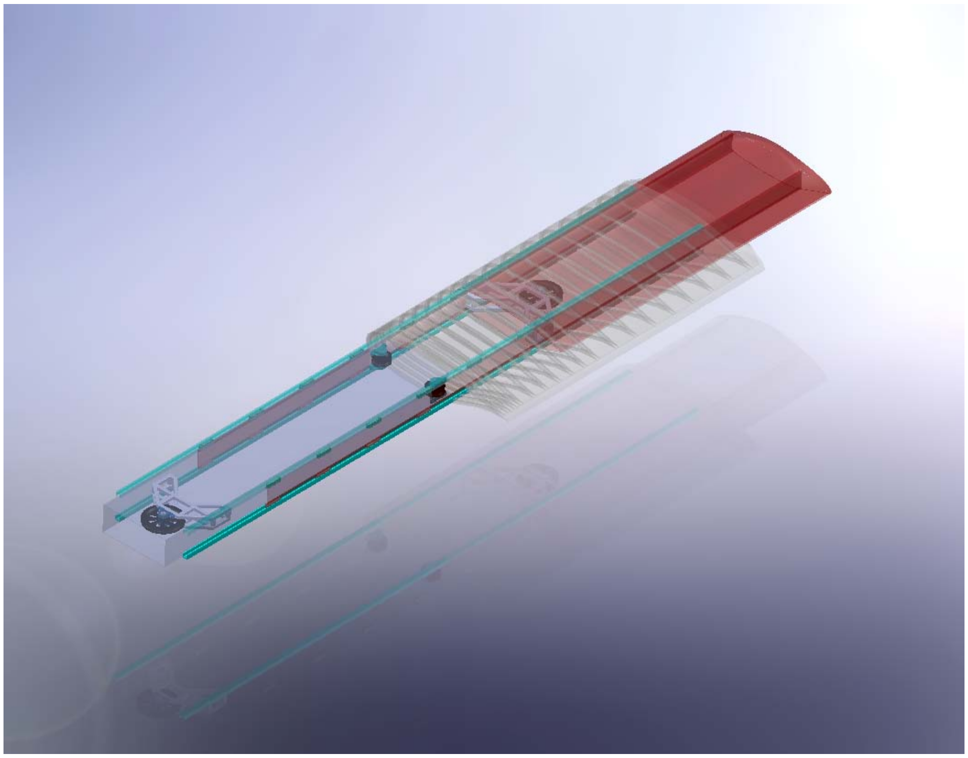


Figure B12.1 – Prototype Tip Extension Mechanism

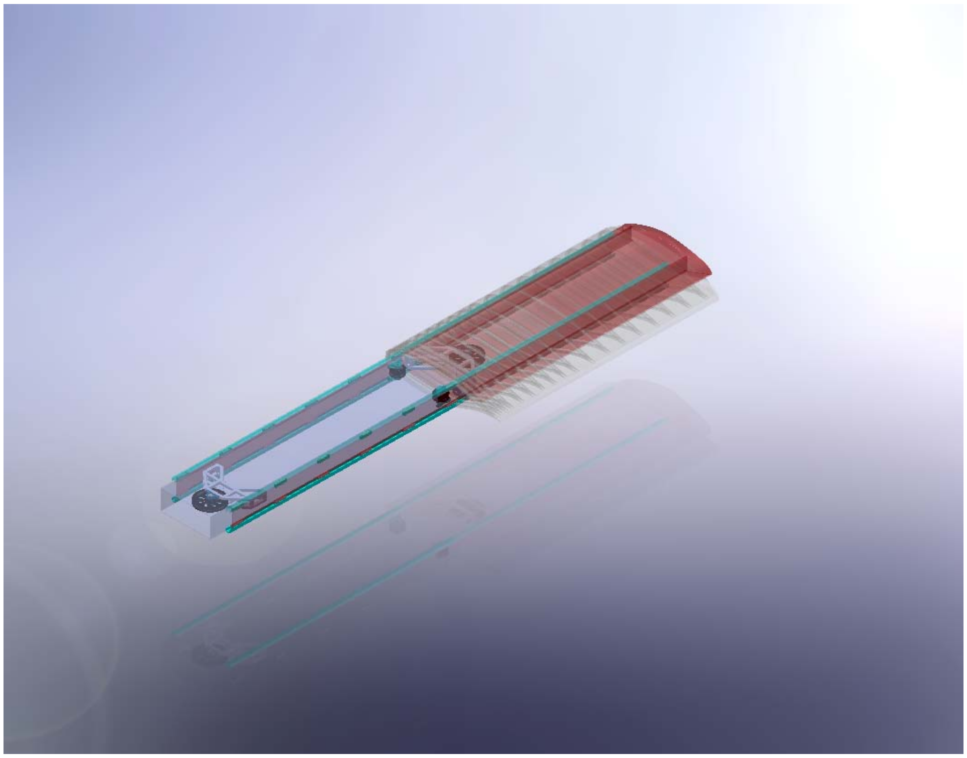


Figure B12.2 – Prototype Tip Extension Mechanism

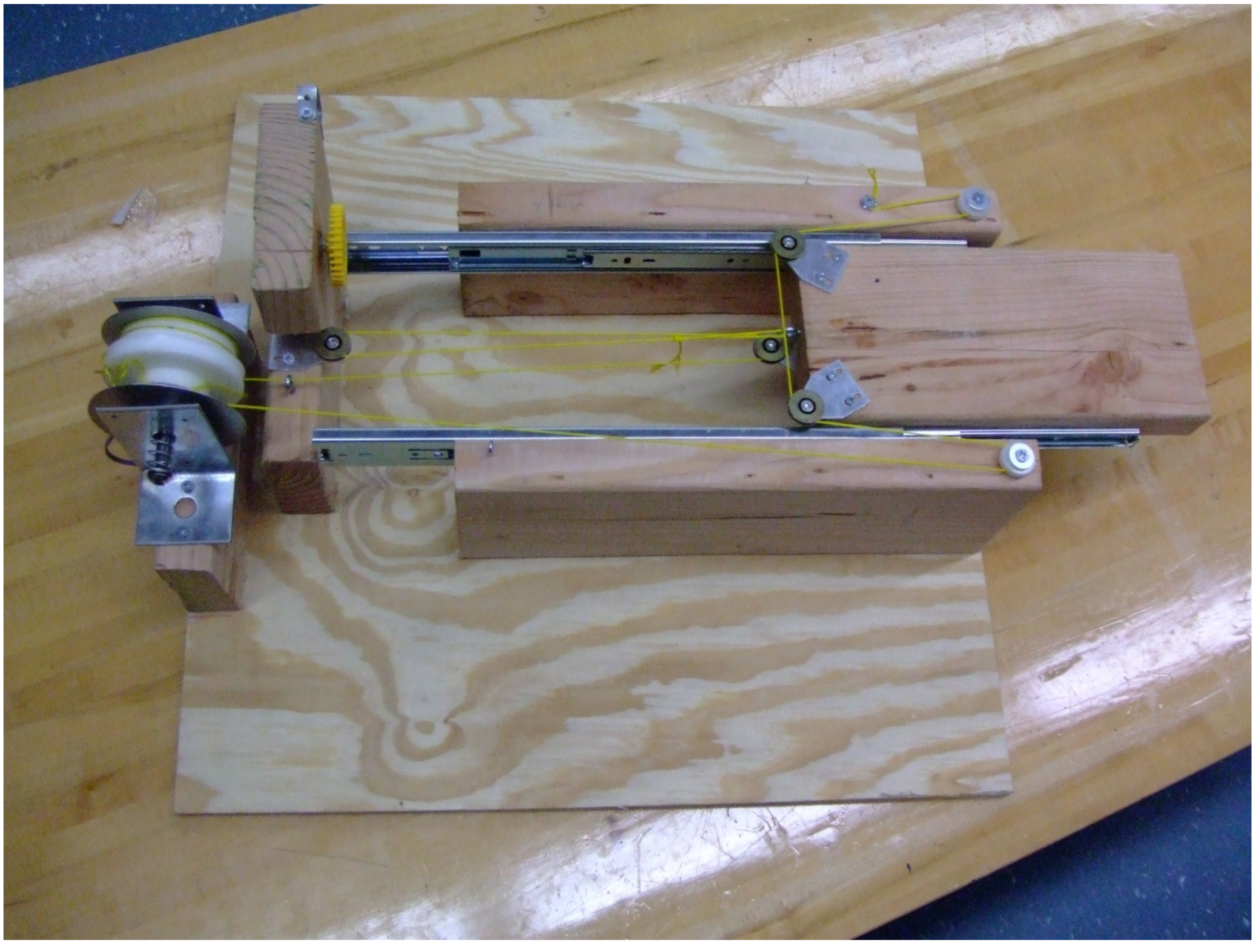


Figure B12.3 – Proof of Concept Tip Extension Mechanism

13 Full-Scale Pictures

Figure C13.1 – Turbine Blade Assembly with Tip Extension

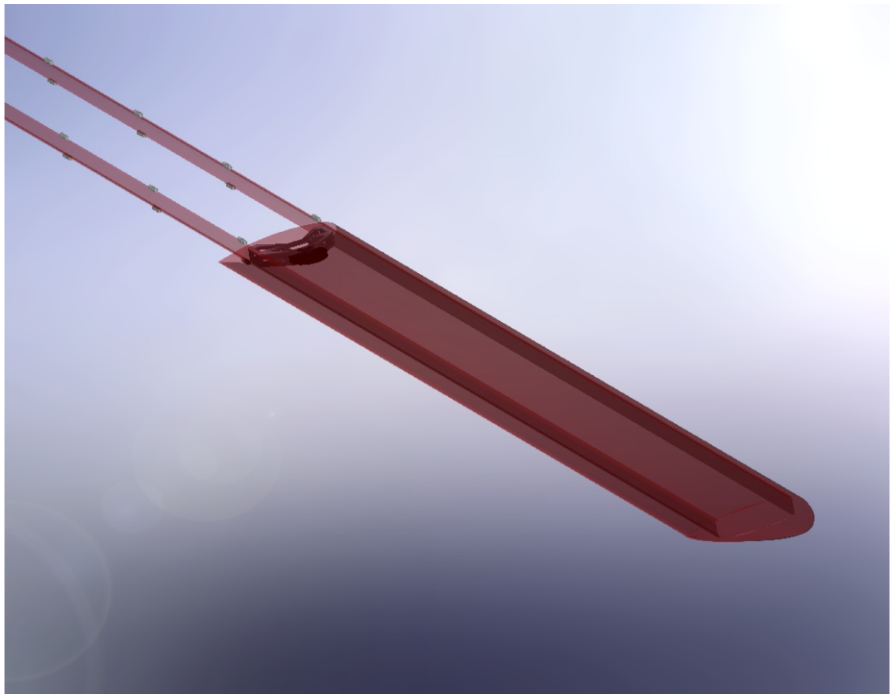


Figure C13.2 – Tip Extension Sub-Assembly

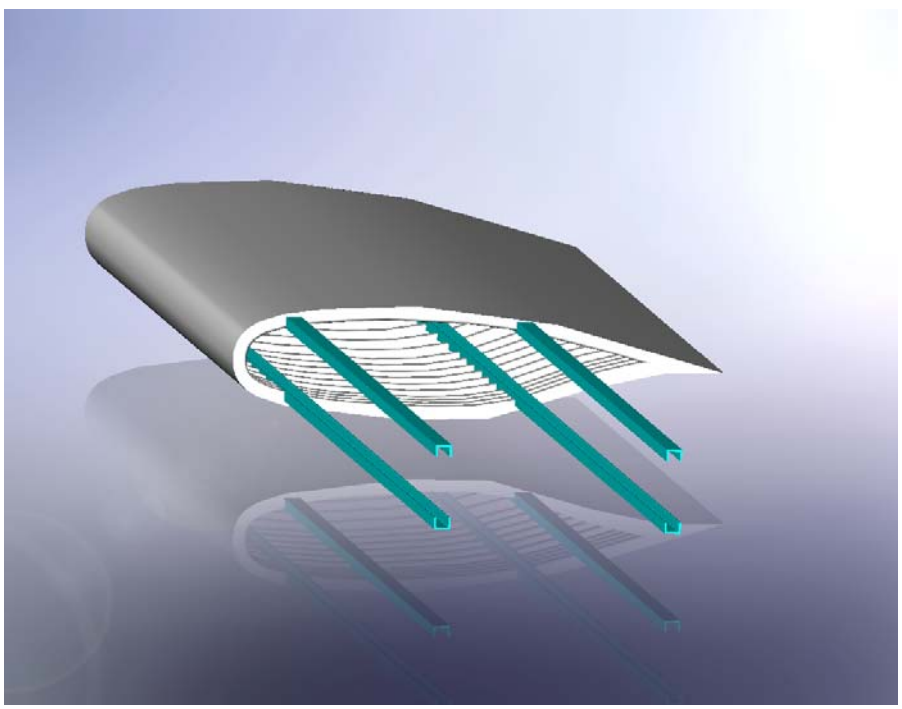


Figure C13.3 – Rail Structure

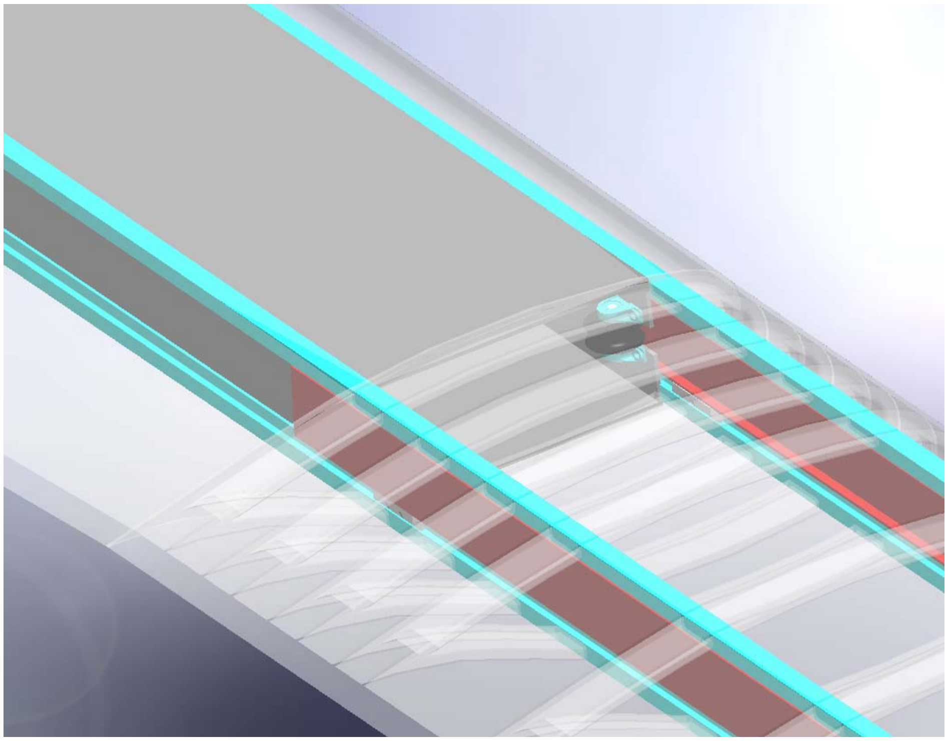


Figure C13.4 – Rail Structure Attached to Outside of Box

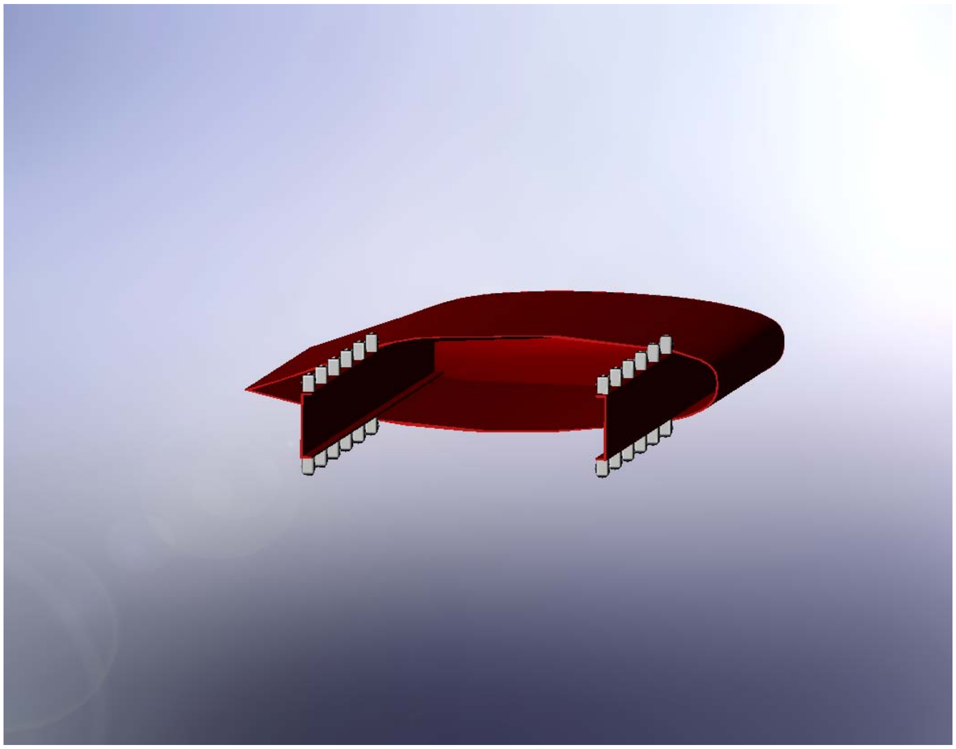


Figure C13.5 – Tip Extension C-Channel

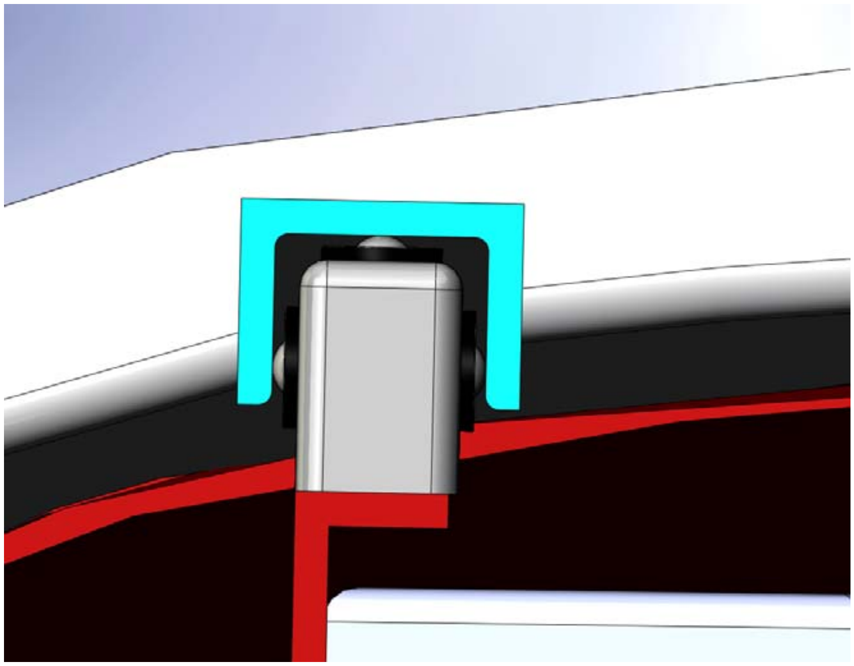


Figure C13.6 – Ball Bearings in Railing System

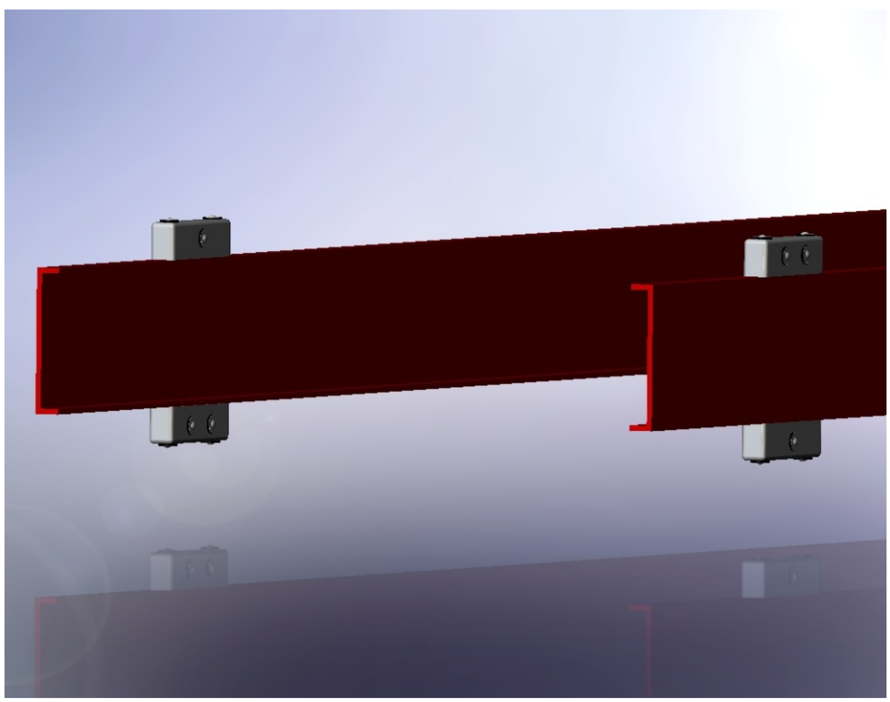


Figure C13.7 – Ball Bearings Mounted on C-Channel

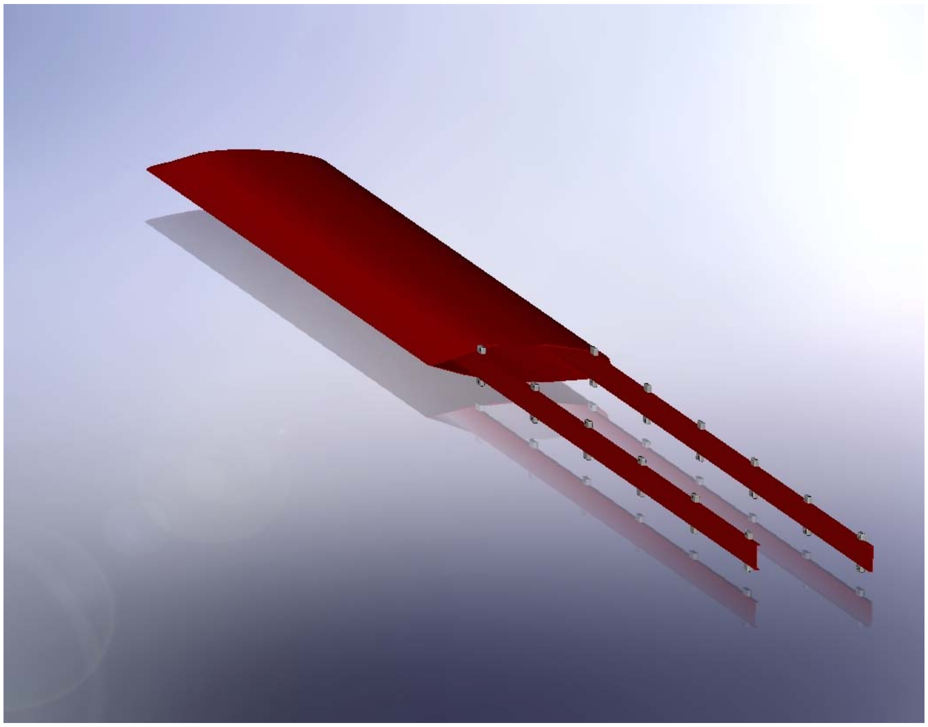


Figure C13.8 – All of the Ball Bearings Mounted on C-Channel of Tip

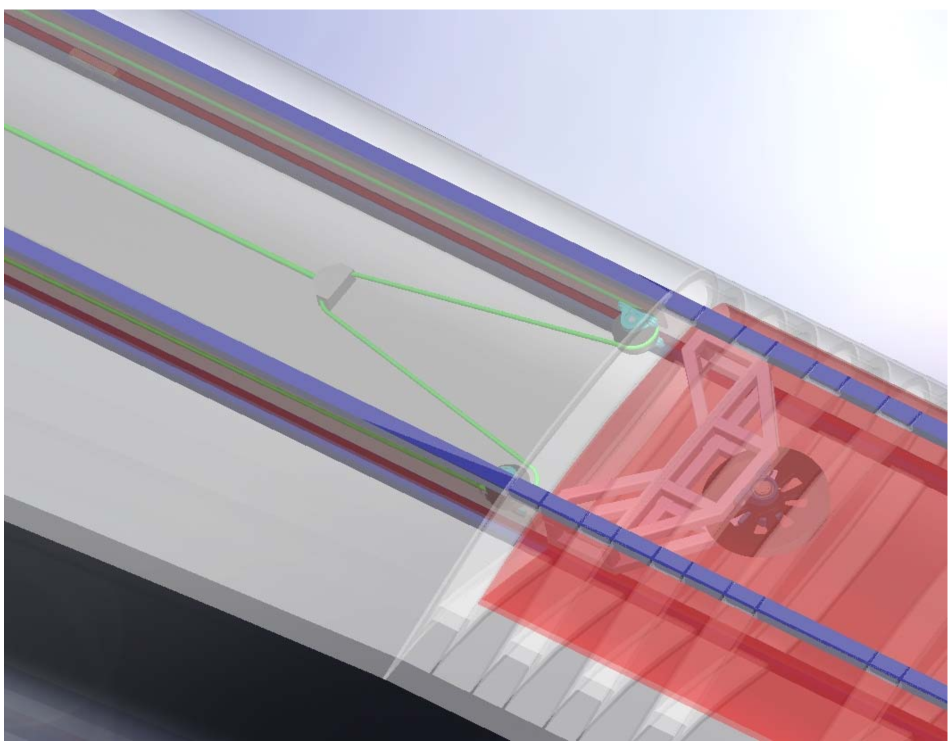


Figure C13.9 – All of the Ball Bearings Mounted on C-Channel of Tip Extension

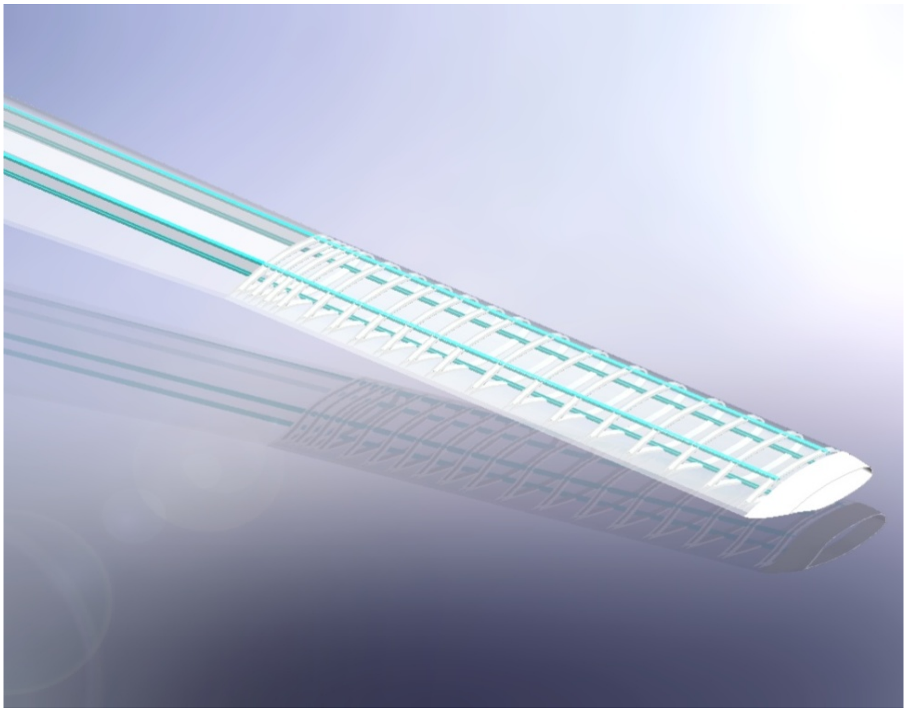


Figure C13.10 – Prototype Tip Extension Mechanism

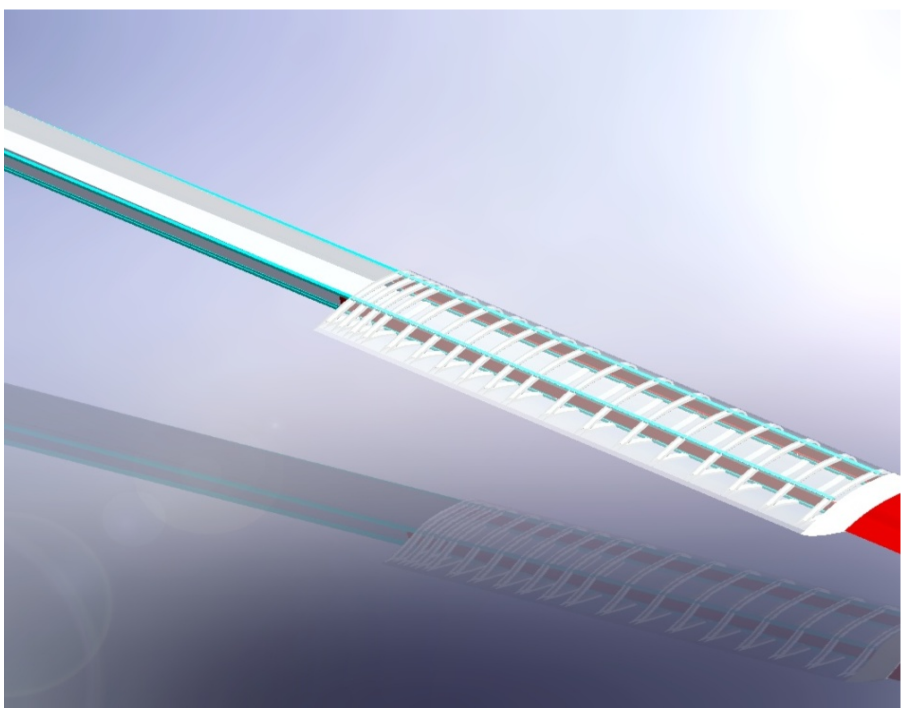


Figure C13.11 – Railing/Stringer Attachment to Box Beam

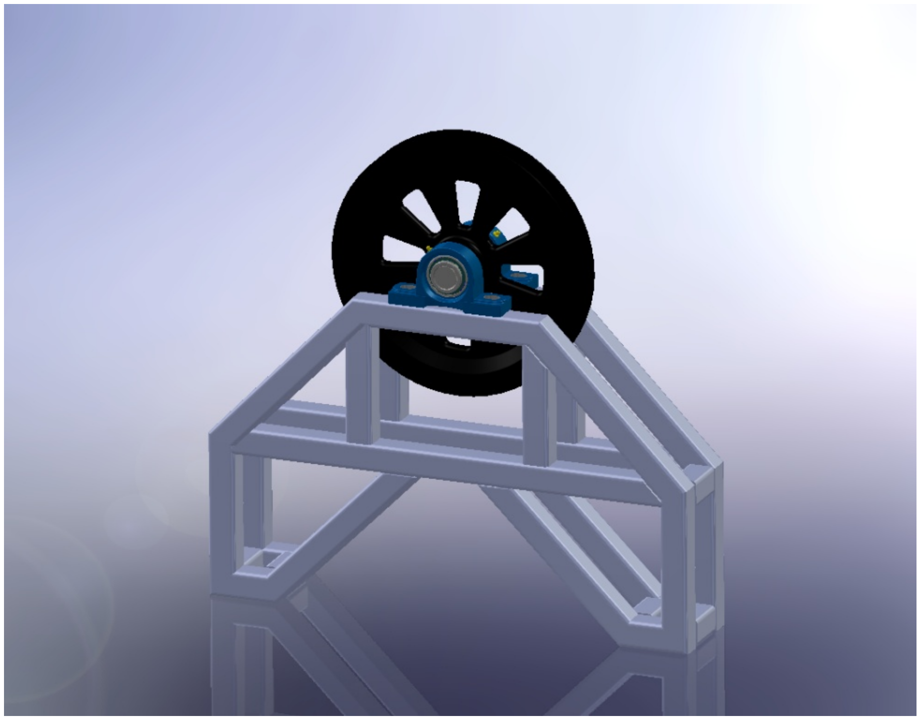


Figure C13.12 – 14 Inch Pulley Assembly

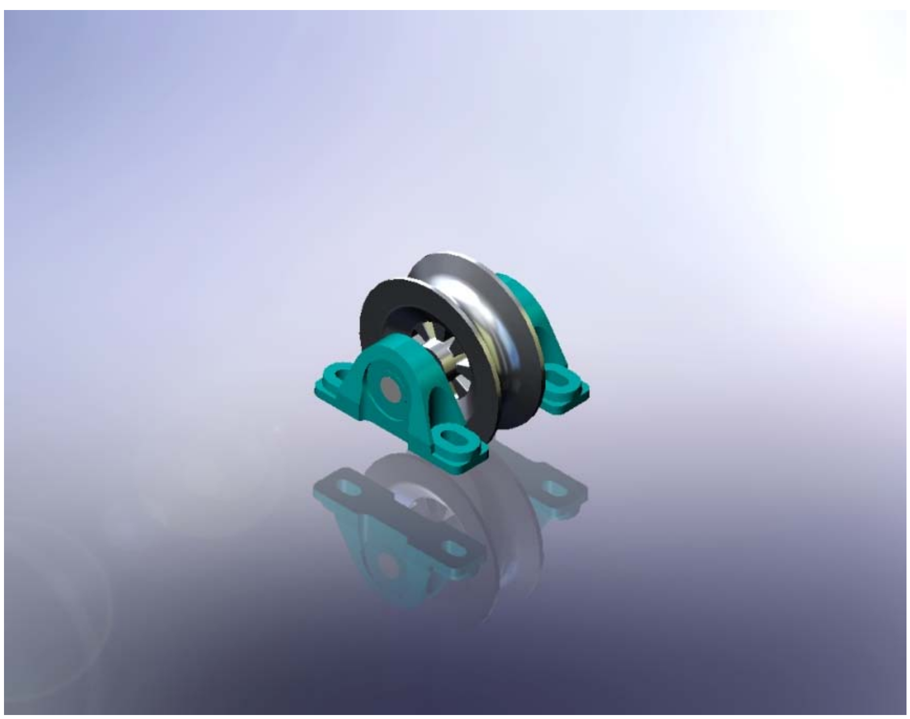


Figure C13.13 – 5 Inch Pulley Assembly

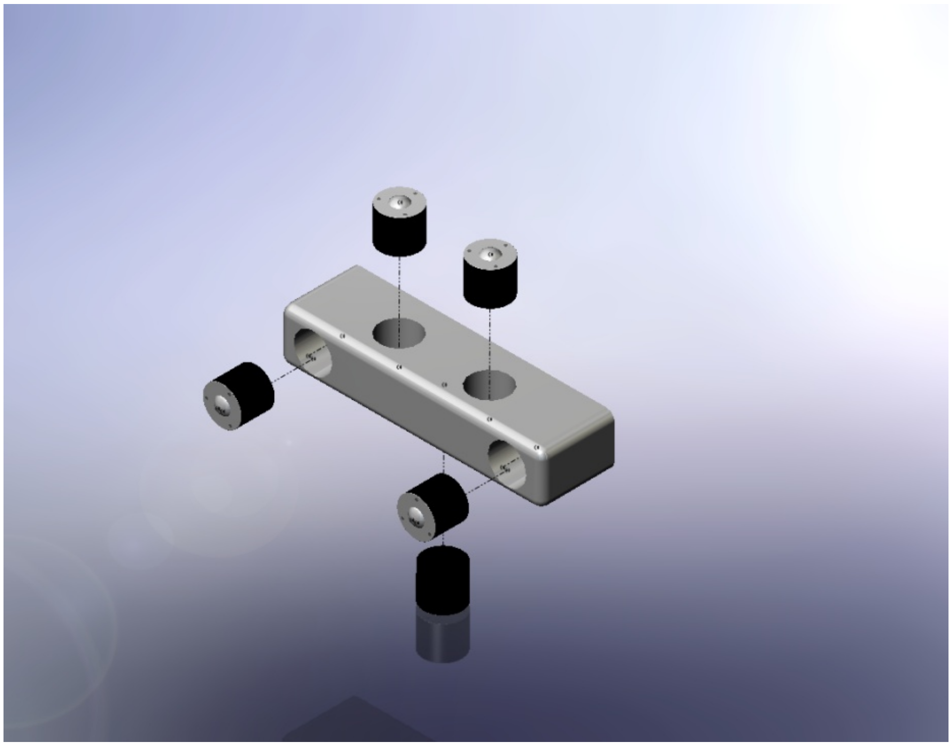


Figure C13.15 – Exploded Caster Assembly

Figure C13.16 – Caster Assembly

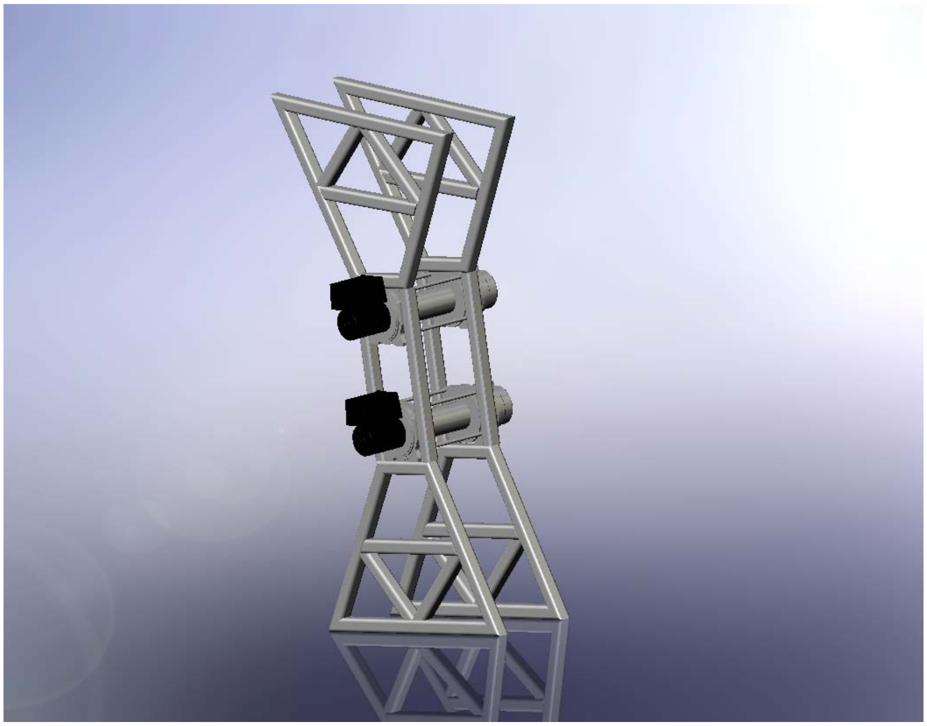


Figure C13.17 – Winch Assembly

Figure C13.18 – Rib

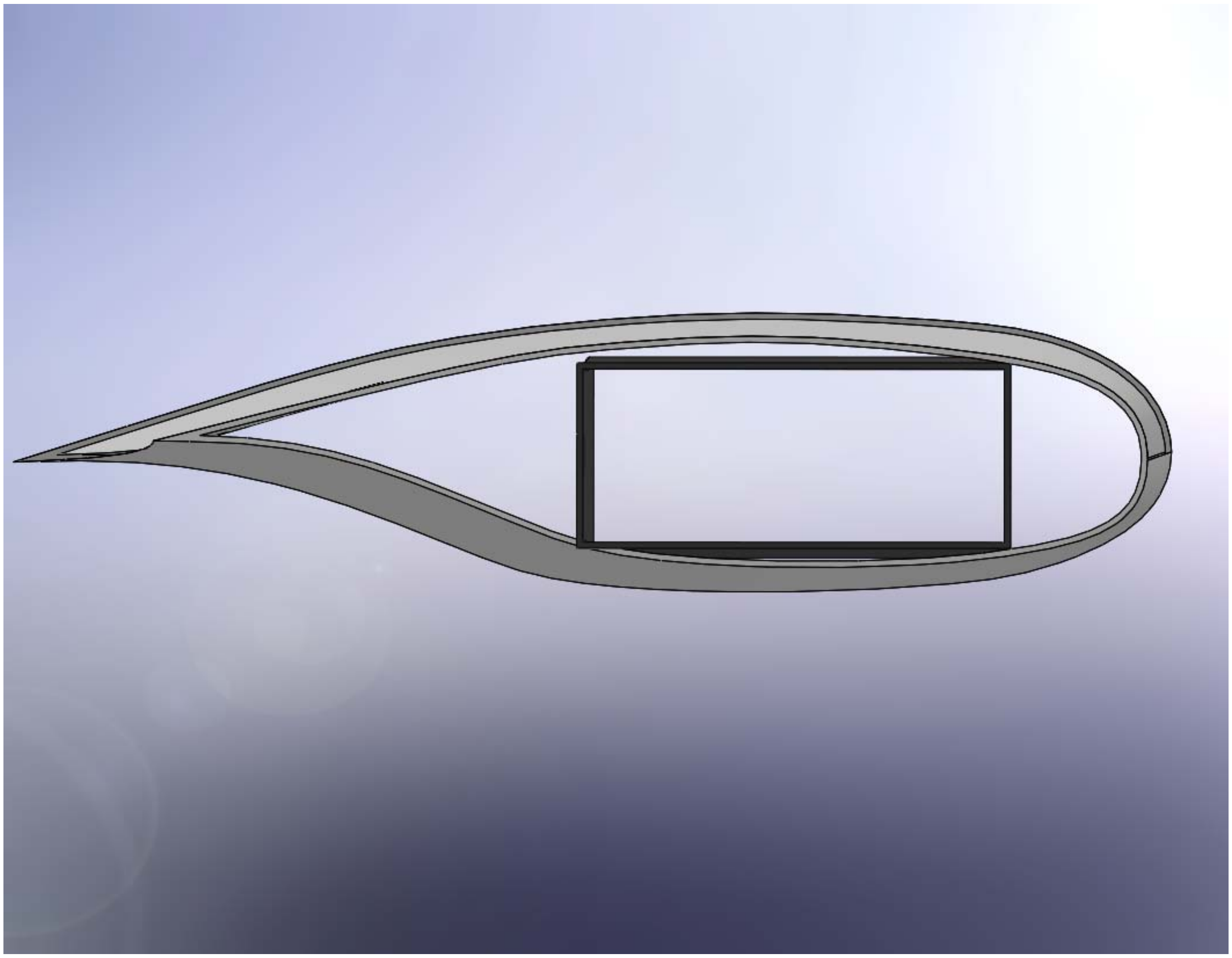
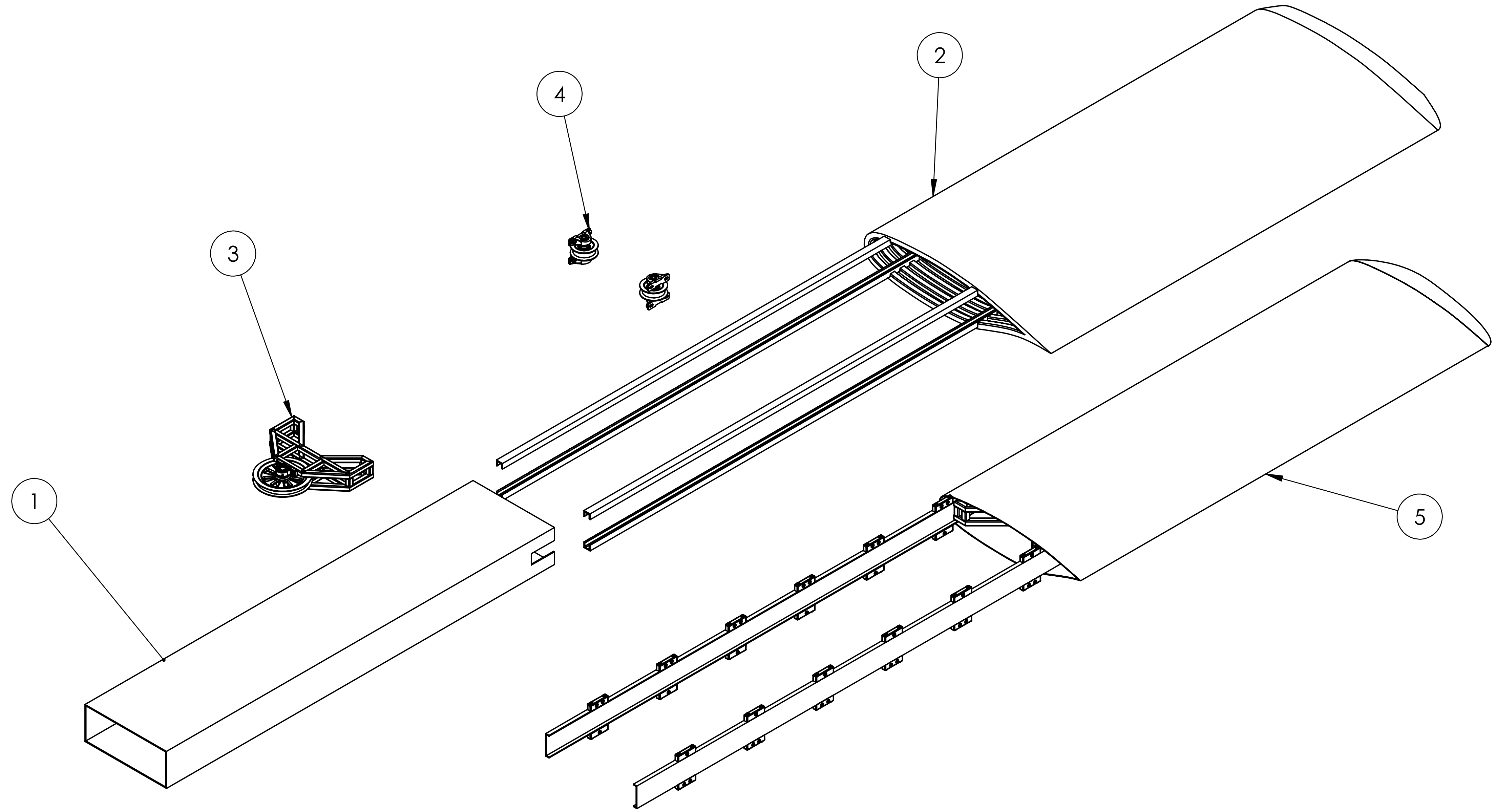


Figure C13.19 – Cross Section

14 Drawing Packet

Scaled Prototype Drawings

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	PROTO_BOXBEAM	PROTOTYPE BOXBEAM	1
2	PROTO_RECEIVER_TRACK_ASSEMBLY	RECEIVER AND TRACK	1
3	PROTO_14INCH_PULLEY_ASSEMBLY1	14" PULLEY AND BRACKET	1
4	PROTO_5INCH_PULLEY_ASSEMBLY	5" PULLEY AND BEARINGS	2
5	PROTO_TIP_SUB1	TIP EXTENSION SUBASSEMBLY	1



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MATERIAL

FINISH

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DRAWN	MLR	3/11/2009
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TEAM 9

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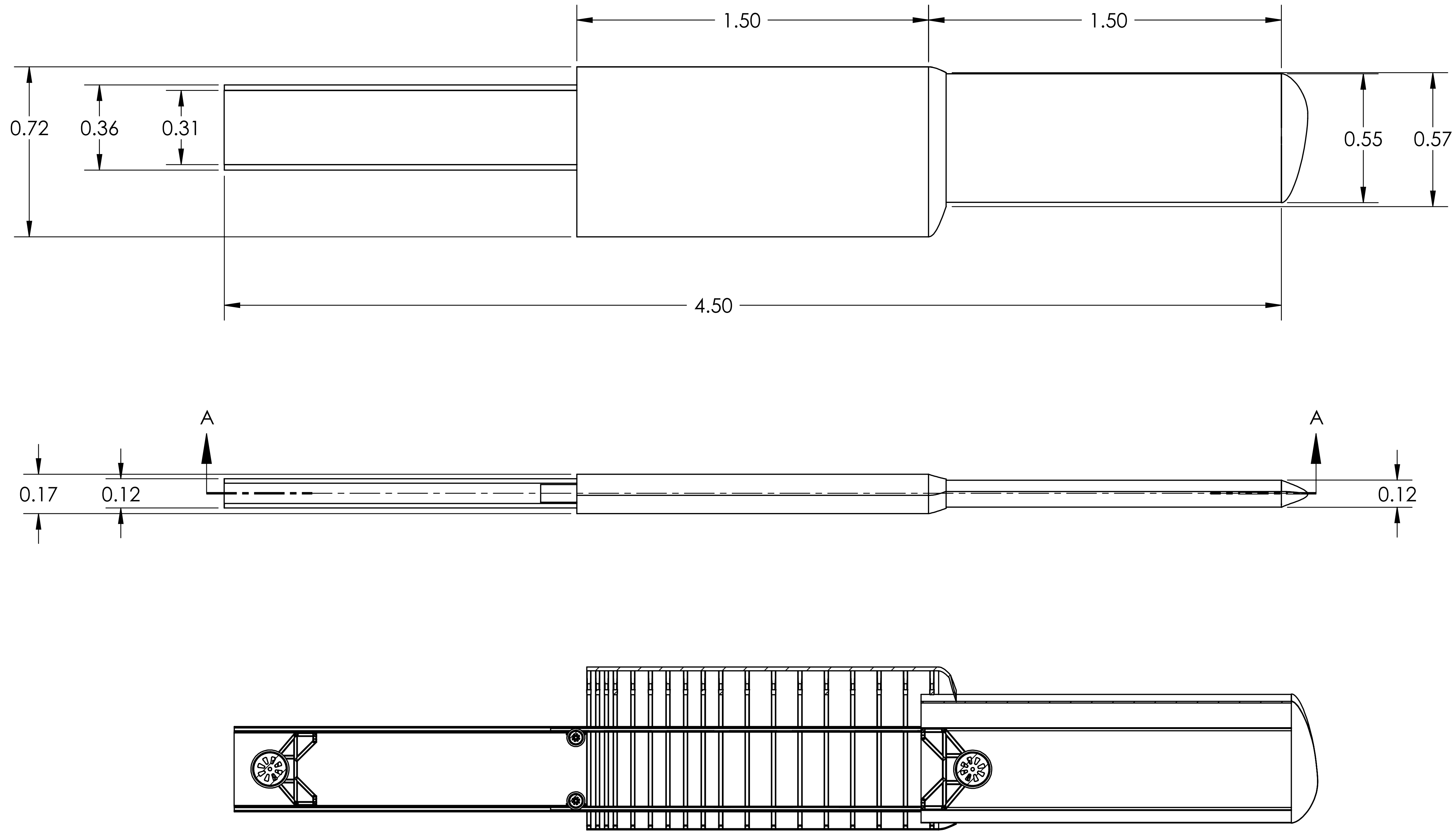

SECTION A-A

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ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TEAM 9

TITLE: PROTOTYPE ASSEMBLY

SIZE	DWG. NO.	REV
B	PROTO-1	

SCALE: 1:20 WEIGHT: SHEET 2 OF 2

14 Drawing Packet

Full-Scale Drawings

Bill of Materials

ITEM NO.	ASSEMBLY/SUBASSEMBLY	PART NUMBER	DESCRIPTION	QTY.	DESIGNERS NAME
0	**CLIPPERWIND_TURBINE_BLADE**		RETRACTABLE WIND TURBINE BLADE CONCEPT		MICHAEL RINGEN
1	**CLIPPERWIND_AIRFOIL_BOXBEAM**		MODIFIED AIRFOIL AND BOXBEAM OF CLIPPER WINDPOWERS LIBERTY	1	MICHAEL RINGEN
1-001		CLIPPERWIND_AIRFOIL	MODIFIED CLIPPER WINDPOWER LIBERTY BLADE AIRFOIL	1	MICHAEL RINGEN
1-002		CLIPPERWIND_BOXBEAM_MOD3	MODIFIED CLIPPER WINDPOWER LIBERTY BLADE BOXBEAM	1	MICHAEL RINGEN
1-2	**WINCHASSEMBLY**		DUAL WINCHES AND MOUNTING BRACKET	1	MICHAEL RINGEN
1-2-001		WINCHBRACKET	2" SQ. TUBING WELDMENT	2	MICHAEL RINGEN
1-2-002		SW3dPS-INDUSTRIAL_WINCH	ELECTRICWINCH(CONTENTCENTRAL)	2	CONTENTCENTRAL
1-3	**5INCH_PULLEY_ASSEMBLY**		5" PULLEY AND PILLOW BLOCKS	2	MICHAEL RINGEN
1-3-001		SW3dPS-BEARING O.75, PILLOW	BEARING Ø.75, PILLOW BLOCK(CONTENTCENTRAL)	2	CONTENTCENTRAL
1-3-002		5INCH_PULLEY	5" DIA. ROPE PULLEY	1	MICHAEL RINGEN
1-3-003		5INCH_PULLEYSHAFT	0.75" SHAFTING	1	MICHAEL RINGEN
1-4	**14INCH_PULLEY_ASSEMBLY2**		14" PULLEY, PILLOW BLOCKS AND BRACKETING	1	MICHAEL RINGEN
1-4-001		14INCH_PULLEY_BRACKET 2	1.5" SQ. TUBING WELDMENT	1	MICHAEL RINGEN
1-4-002		14INCH_PULLEY	14' DIA. PULLEY	1	MICHAEL RINGEN
1-4-003		1303	PILLOW BLOCK BEARING,UCP-206-20(CONTENTCENTRAL)	2	CONTENTCENTRAL
1-4-004		1.5INCH_DIA_PULLEYSHAFT	1.5" DIA. SHAFTING	1	MICHAEL RINGEN
2	**RECEIVER_TRACK_TIPEXT_ASSEMBLY**		BLADE MODIFICATIONS AND TIP EXTENSION	1	MICHAEL RINGEN
2-1		RECEIVER_REV3	RIBS AND STRINGERS EXTENSION HOUSING	1	MICHAEL RINGEN
2-2		TRACK	U-CHANNEL SUPPORTS TIP EXTENSION LOADS	4	MICHAEL RINGEN
2-3	**TIPEXT_SUB1**		TIP EXTENSION, STRUCTURE, CASTERS AND PULLEY	1	MICHAEL RINGEN
2-3-001		TIPEXT_AIRFOIL_STRUCTURE	EXTENSION AIRFOIL AND STRUCTURE	1	MICHAEL RINGEN
2-3-002	**14INCH_PULLEY_ASSEMBLY1**		14" PULLEY AND BRACKET	1	MICHAEL RINGEN
2-3-002-1		14INCH_PULLEY_BRACKET 1	1.5" SQ. TUBING WELDMENT(WIDER THAN BRACKET 2)	1	MICHAEL RINGEN
1-4-002		14INCH_PULLEY	14' DIA. PULLEY	1	MICHAEL RINGEN
1-4-003		1303	PILLOW BLOCK BEARING,UCP-206-20	2	CONTENTCENTRAL
1-4-004		1.5INCH_DIA_PULLEYSHAFT	1.5" DIA. SHAFTING	1	MICHAEL RINGEN
2-3-003	**CASTERASSEMBLY**		BEARINGS AND HOUSING	24	MICHAEL RINGEN
2-3-003-1		BALLTRANSFER_CASE	RECTANGULAR HOUSING	1	MICHAEL RINGEN
2-3-003-2		BALLTRANSFER_6460K53	BALL TRANSFER BEARING(MCMASTER)	5	MCMASTER

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
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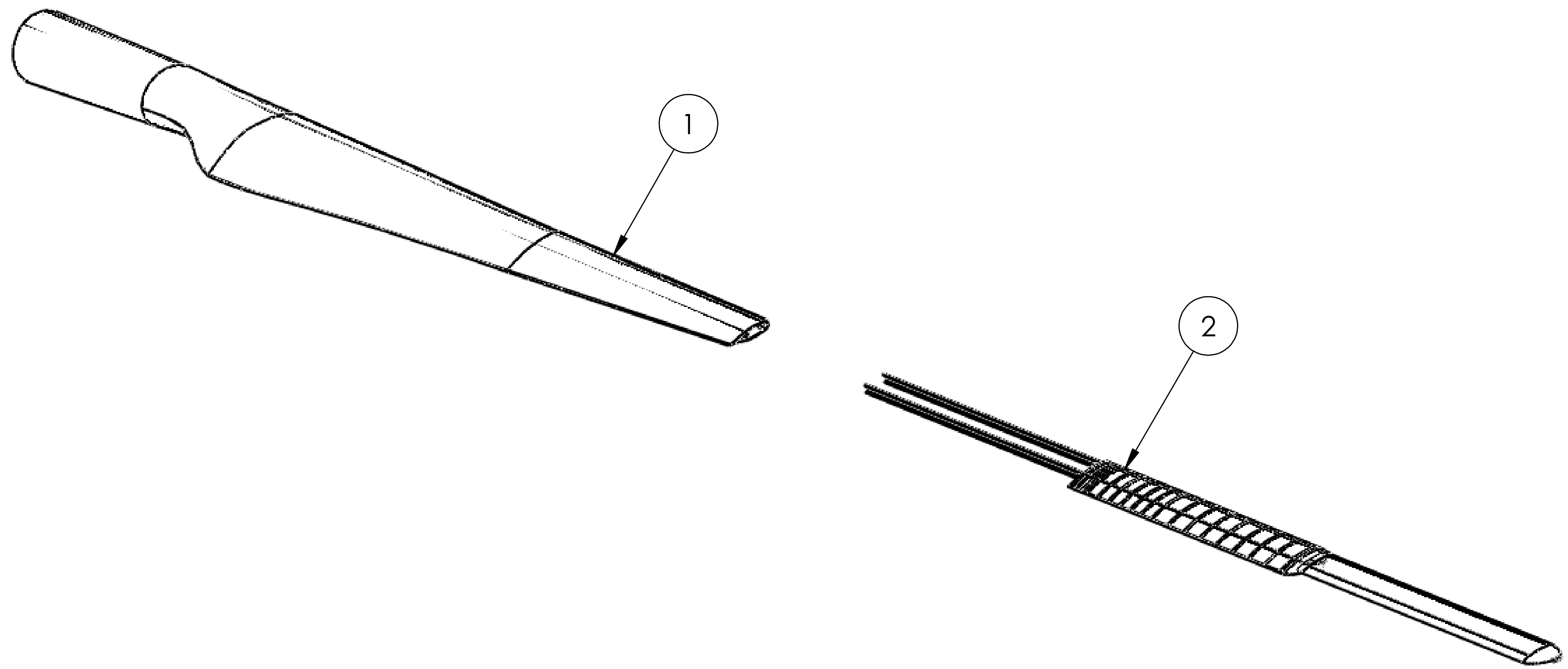

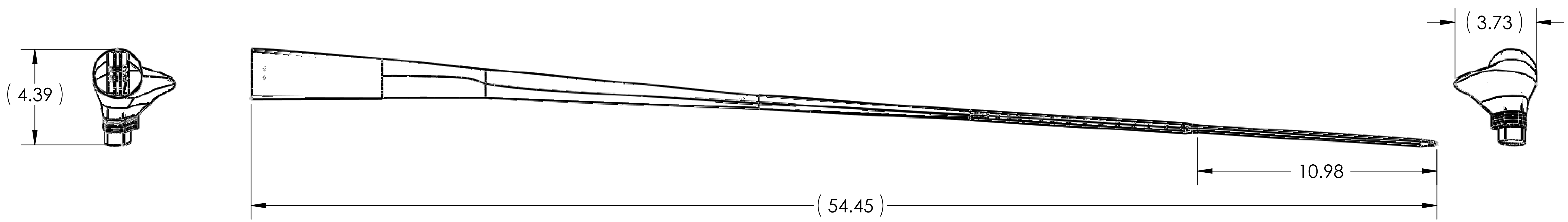

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MATERIAL
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MLR	3/8/2009
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

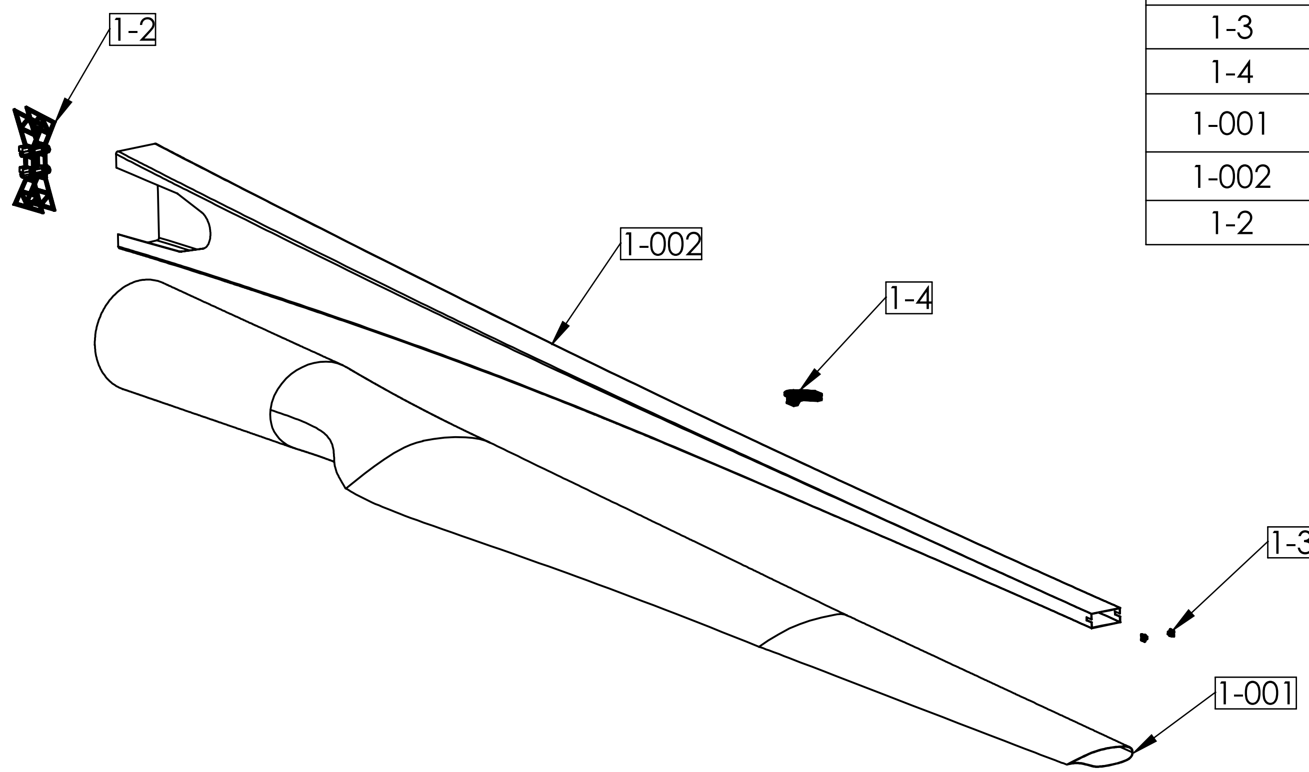
TEAM 9

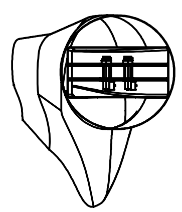
TITLE: CLIPPER WINDPOWER RETRACTABLE TURBINE BLADE

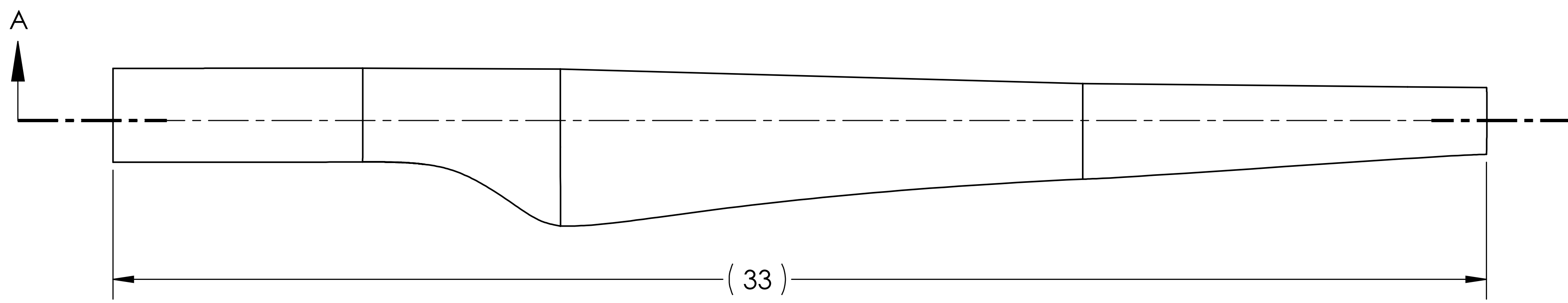
SIZE **B** DWG. NO. 0 REV

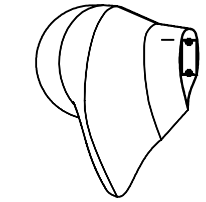
SCALE: 1:200 WEIGHT: SHEET 1 OF 1

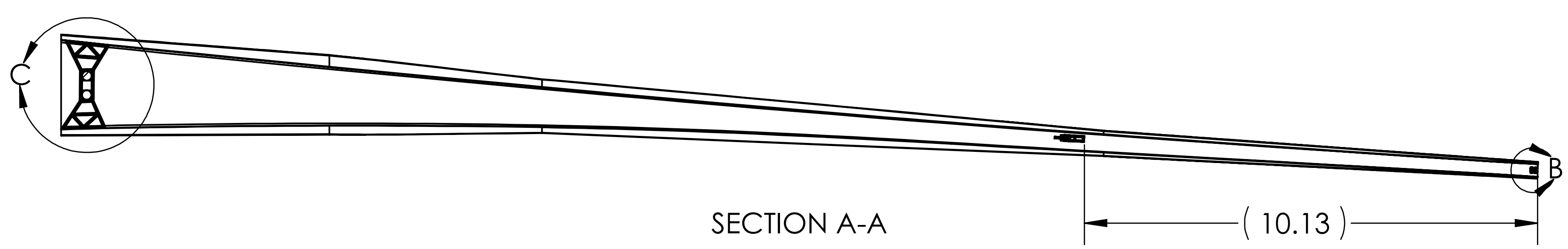
ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1-3	5INCH_PULLEY_ASSEMBLY	5" PULLEY AND PILLOW BLOCKS	2
1-4	14INCH_PULLEY_ASSEMBLY2	14" PULLEY, PILLOW BLOCKS AND BRACKETING	1
1-001	CLIPPERWIND_AIRFOIL	MODIFIED CLIPPER WINDPOWER LIBERTY BLADE AIRFOIL	1
1-002	CLIPPERWIND_BOXBEAM_MOD3	MODIFIED CLIPPER WINDPOWER LIBERTY BLADE BOXBEAM	1
1-2	WINCHASSEMBLY	DUAL WINCHES AND MOUNTING BRACKET	1



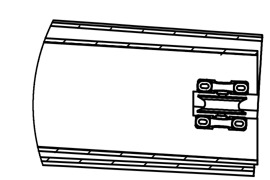




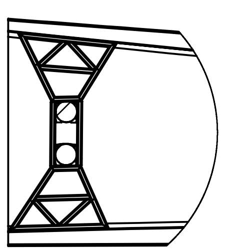




SECTION A-A



DETAIL B
SCALE 1 : 20



DETAIL C
SCALE 1 : 75

PROPRIETARY AND CONFIDENTIAL

THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF CLIPPER WINDOWER TEAM 9. ANY REPRODUCTION IN PART OR AS A WHOLE WITHOUT THE WRITTEN PERMISSION OF CLIPPER WINDPOWER TEAM 9 IS PROHIBITED.

NEXT ASSY | USED ON

APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN METERS

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MLR	3/8/2009
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

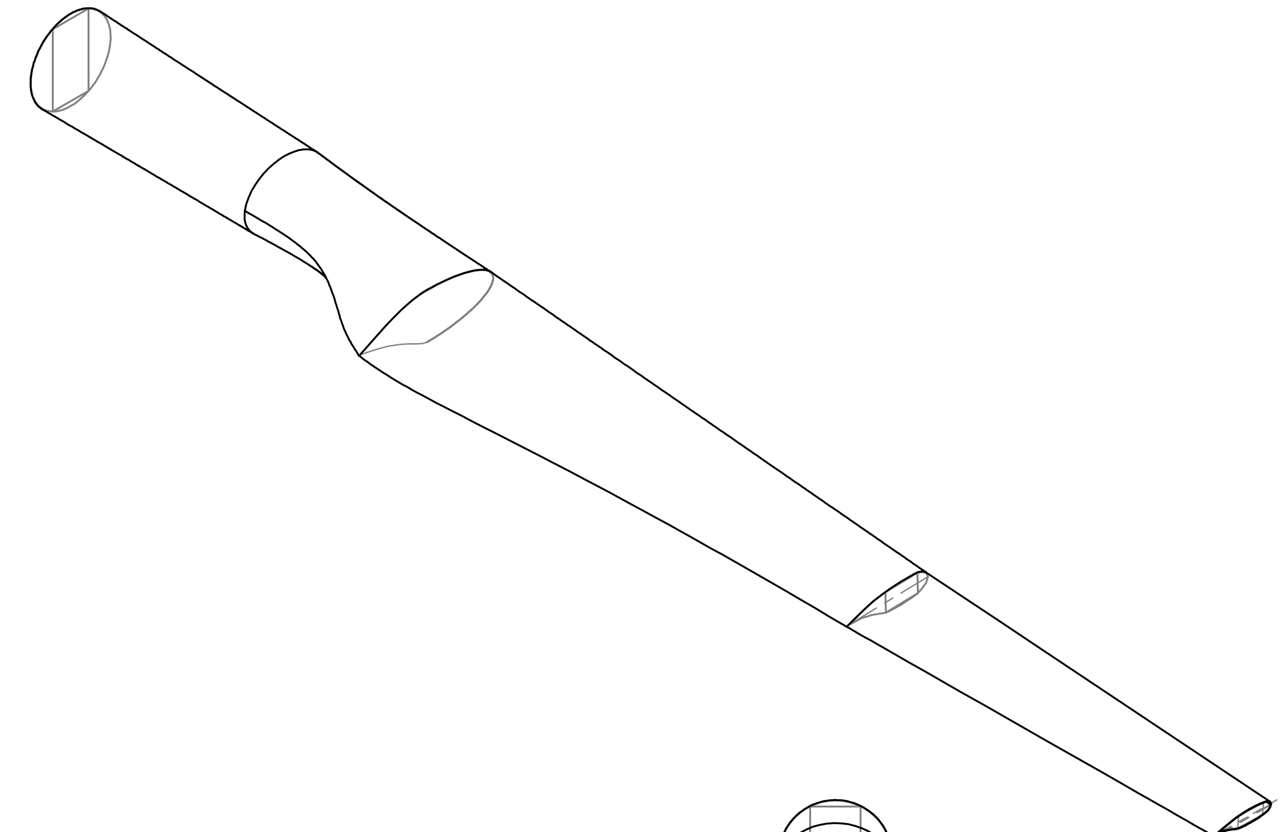
TEAM 9

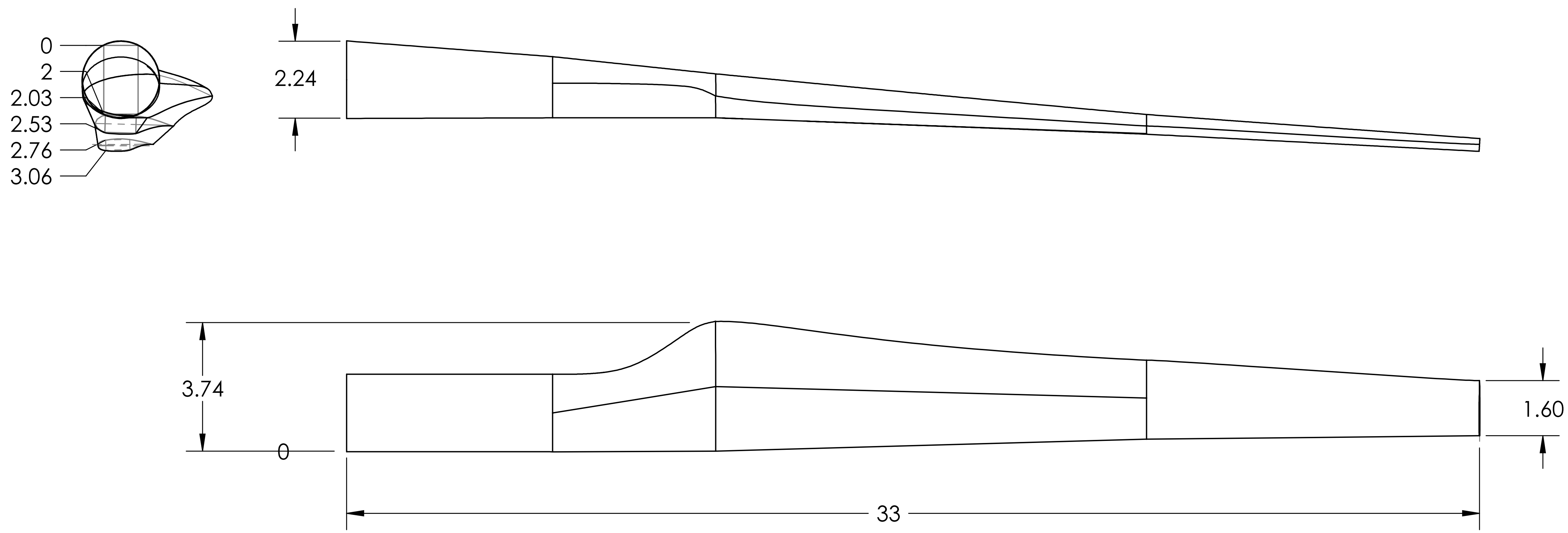
TITLE:
CLIPPER WINDPOWER BLADE MODIFICATIONS SUBASSEMBLY

SIZE **B** | DWG. NO. 1 | REV

SCALE: 1:150 | WEIGHT: | SHEET 1 OF 1

NOTE:
1. MODEL BUILT IN REFERENCE TO CLIPPERWINDPOWER'S LIBERTY BLADE
2. DIMENSION AIRFOIL FROM MODEL AT CROSS-SECTIONS

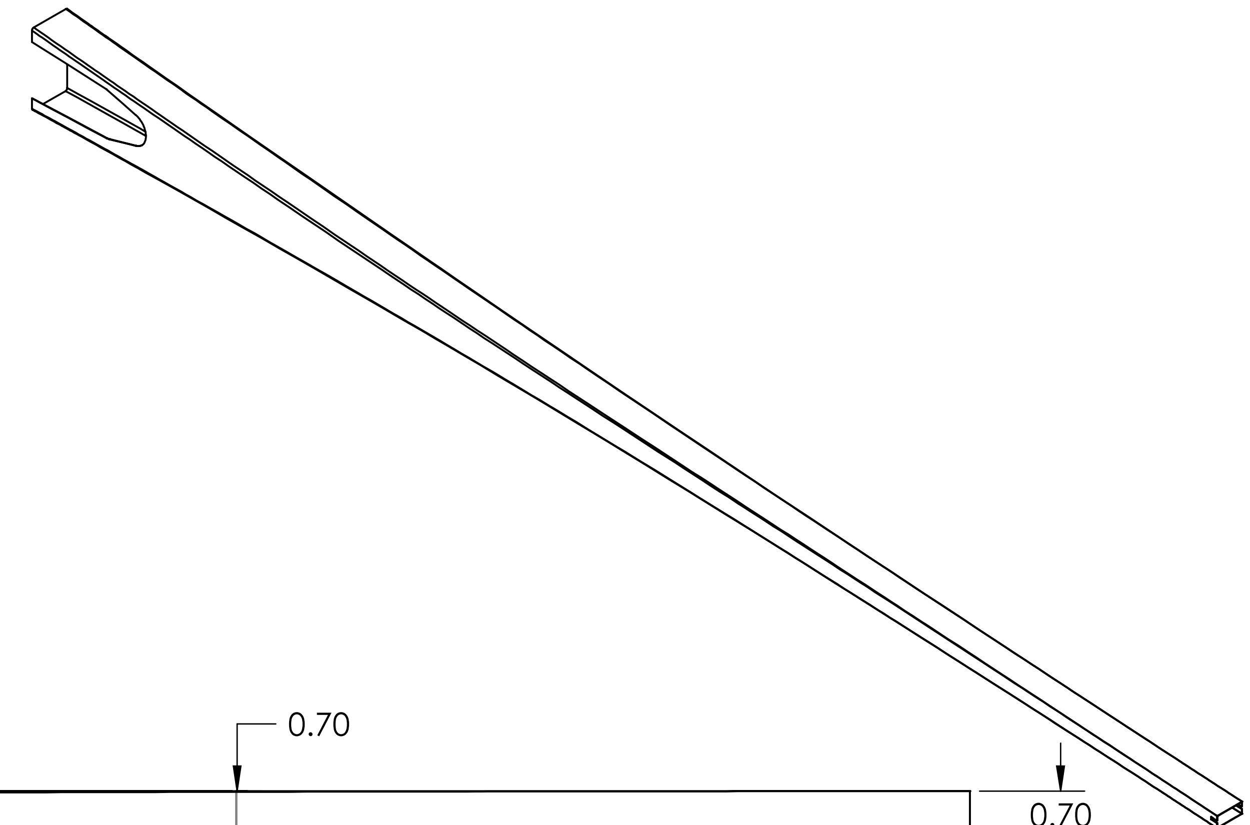


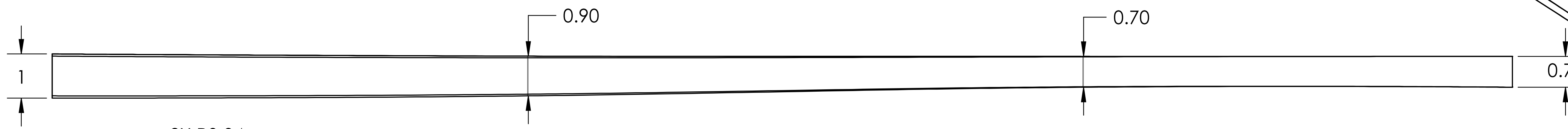


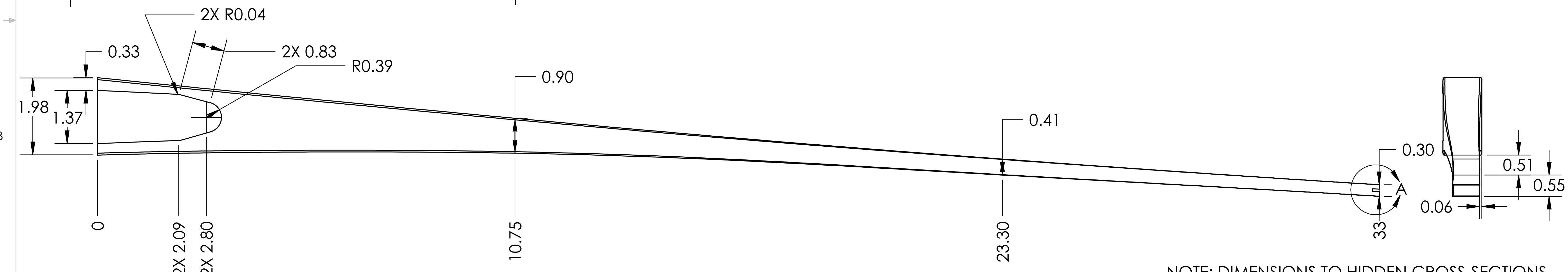
UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TEAM 9	
DIMENSIONS ARE IN METERS TOLERANCES: TWO PLACE DECIMAL: ± .005	DRAWN	MLR	3/8/2009	TITLE: CLIPPERWIND AIRFOIL	
	CHECKED				
	ENG APPR.				
	MFG APPR.				
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.				
MATERIAL GRFM	COMMENTS:			SIZE B	DWG. NO. 1-001	REV
FINISH SMOOTH CLEAR COAT					
DO NOT SCALE DRAWING				SCALE: 1:150	WEIGHT:	SHEET 1 OF 1

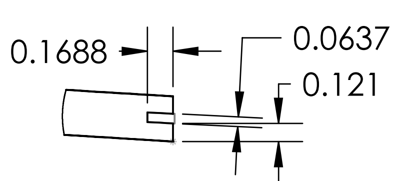
NEXT ASSY	USED ON
APPLICATION	

NOTE:
1. UNIFORM THICKNESS THROUGHOUT 0.01
2. MODEL DIMENSIONS BASED ON CLIPPER WINDPOWER'S LIBERTY BLADE
3. REFER TO CAD MODEL FOR VARIABLE RADIUS DIMENSIONS









DETAIL A
SCALE 1 : 50
CUT THRU ALL

NOTE: DIMENSIONS TO HIDDEN CROSS-SECTIONS AT ORDINATE LOCATIONS

NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN METERS
TOLERANCES:
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .002
FOUR PLACE DECIMAL ± .0005

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: GRFM

FINISH: AS MOLDED

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MLR	3/8/2009
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

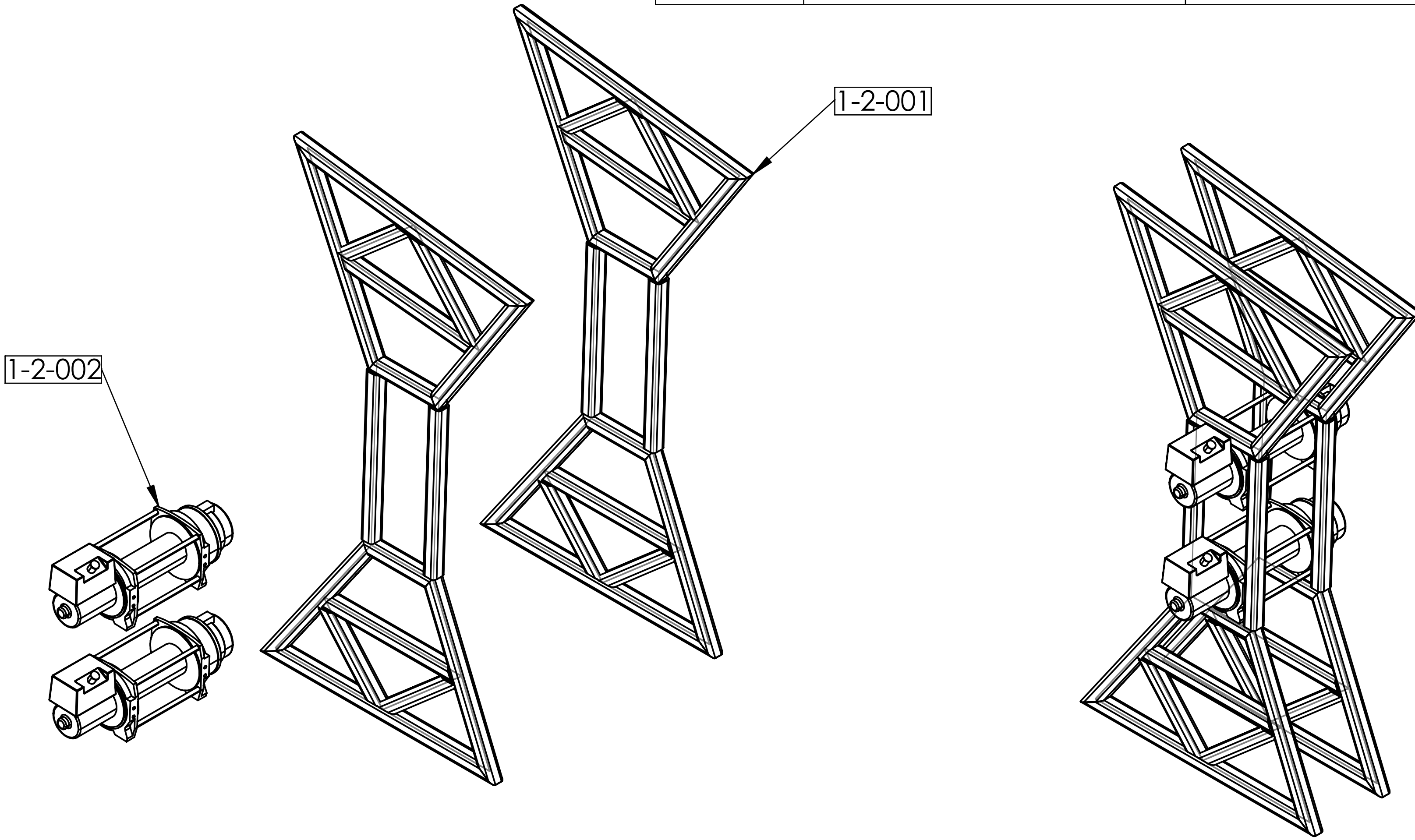
TEAM 9

TITLE: CLIPPERWIND BOXBEAM MOD3

SIZE	DWG. NO.	REV
B	1-002	

SCALE: 1:100 WEIGHT: SHEET 1 OF 1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1-2-001	WINCHBRACKET	2" SQ. TUBING WELDMENT	2
1-2-002	SW3dPS-INDUSTRIAL_WINCH	ELECTRICWINCH	2



UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MLR	3/8/2009
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL	COMMENTS:		
FINISH			
DO NOT SCALE DRAWING			

NEXT ASSY	USED ON
APPLICATION	

PROPRIETARY AND CONFIDENTIAL

TEAM 9

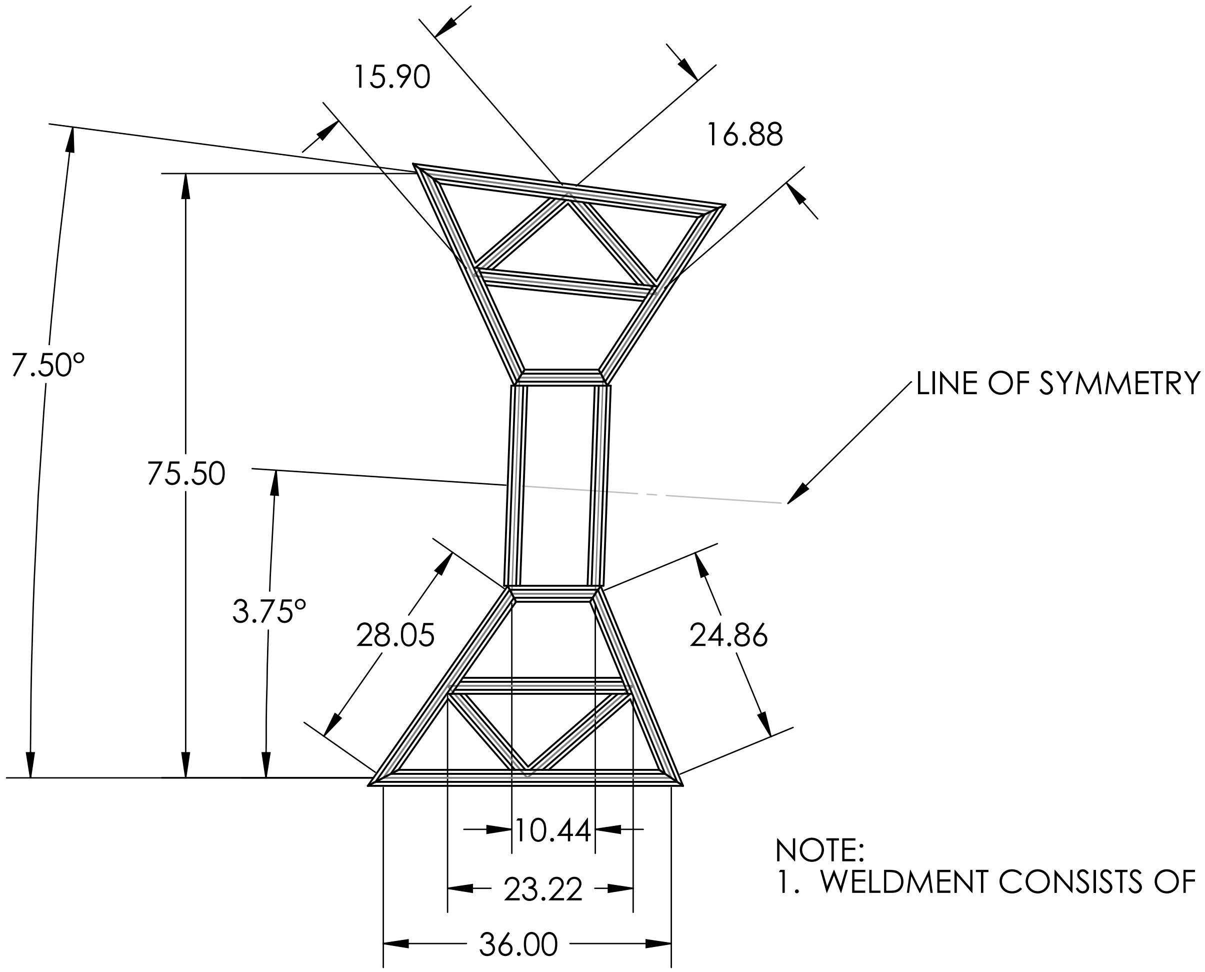
TITLE:

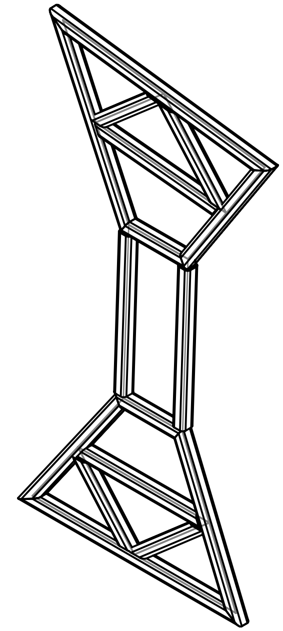
WINCH ASSEMBLY

SIZE A | DWG. NO. 1-2 | REV

SCALE: 1:20 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1





NOTE:
1. WELDMENT CONSISTS OF 2" SQ. TUBING WITH .25" WALL

PROPRIETARY AND CONFIDENTIAL

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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:

ANGULAR: MACH ± 1°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
STEEL

FINISH
AS CUT

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MLR	3/10/2009
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TEAM 9

TITLE:

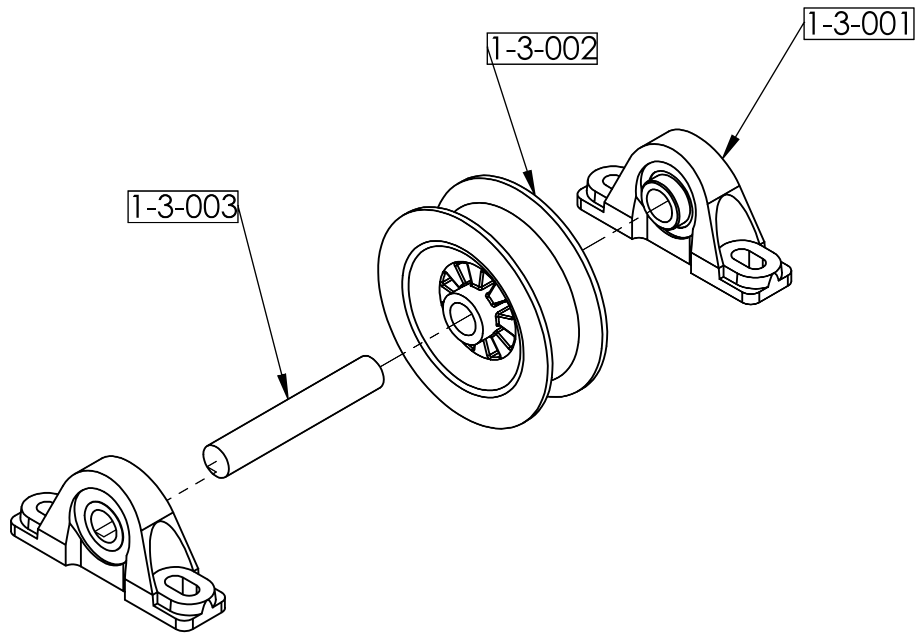
WINCH BRACKET

SIZE	DWG. NO.	REV
A	1-2-001	

SCALE: 1:24 WEIGHT: SHEET 1 OF 1

5 4 3 2 1

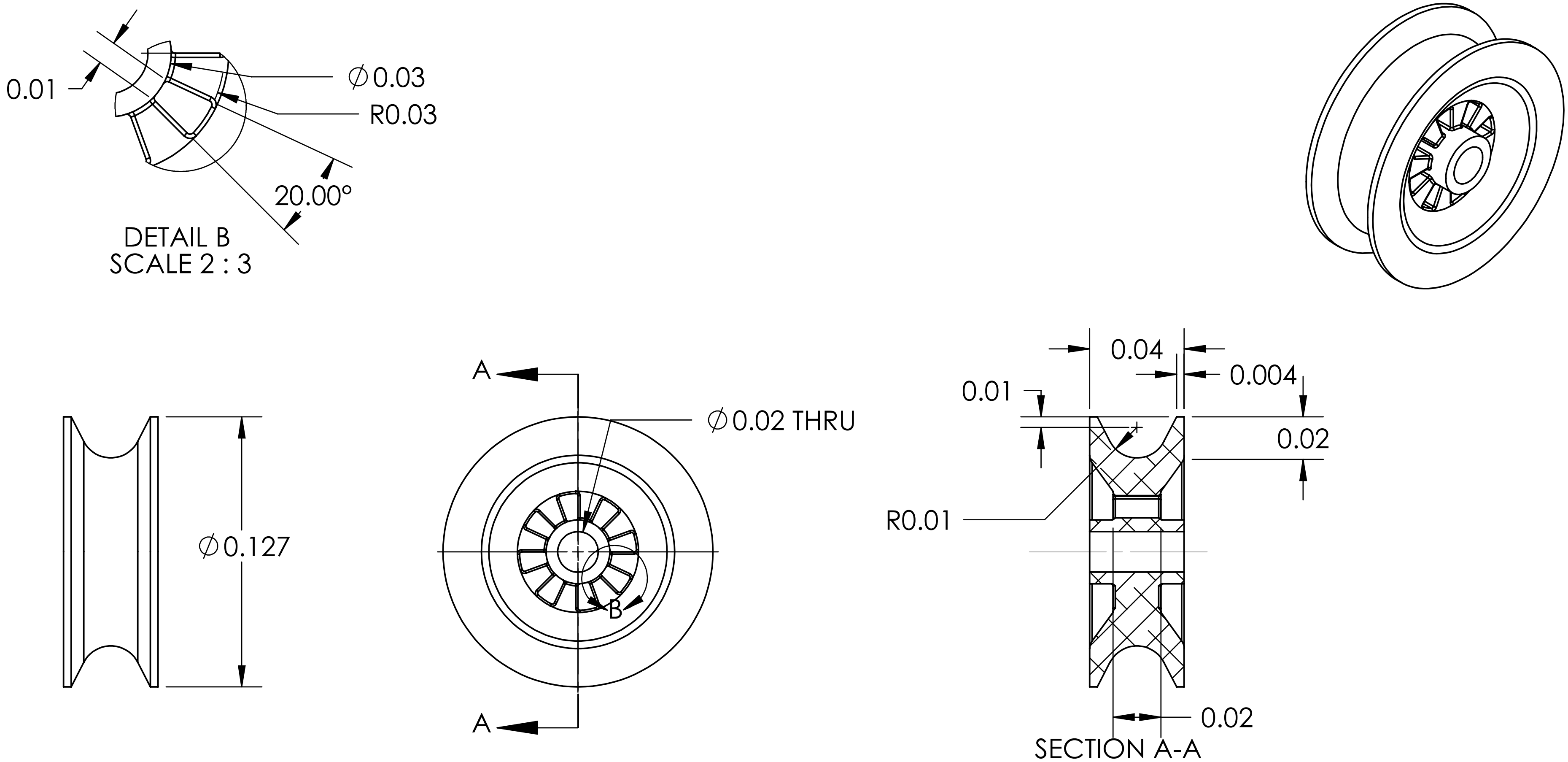
ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1-3-001	SW3dPS-BEARING O.75, PILLOW BLOCK	BEARING Ø.75, PILLOW BLOCK	2
1-3-002	5INCH_PULLEY	5" DIA. ROPE PULLEY	1
1-3-003	5INCH_PULLEYSHAFT	0.75" SHAFTING	1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TEAM 9	
	DRAWN	MLR	3/8/2009	TITLE: 5" PULLEY SUBASSEMBLY	
	CHECKED				
	ENG APPR.				
	MFG APPR.				
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.				
MATERIAL	COMMENTS:			SIZE A	DWG. NO. 1-3	REV
FINISH					
DO NOT SCALE DRAWING				SCALE: 1:4	WEIGHT:	SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	

5 4 3 2 1



NOTE: ALL FILLETS RADIUS .001METERS

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN METERS TOLERANCES: ANGULAR: MACH ± 1° TWO PLACE DECIMAL ± .01 THREE PLACE DECIMAL ± .001		NAME / DATE
	DRAWN	MLR / 3/8/2009
	CHECKED	
	ENG APPR.	
	MFG APPR.	
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.	
MATERIAL ALUMINIUM 6061-T6	COMMENTS:	
FINISH POLISHED		
DO NOT SCALE DRAWING		

NEXT ASSY	USED ON
APPLICATION	

PROPRIETARY AND CONFIDENTIAL

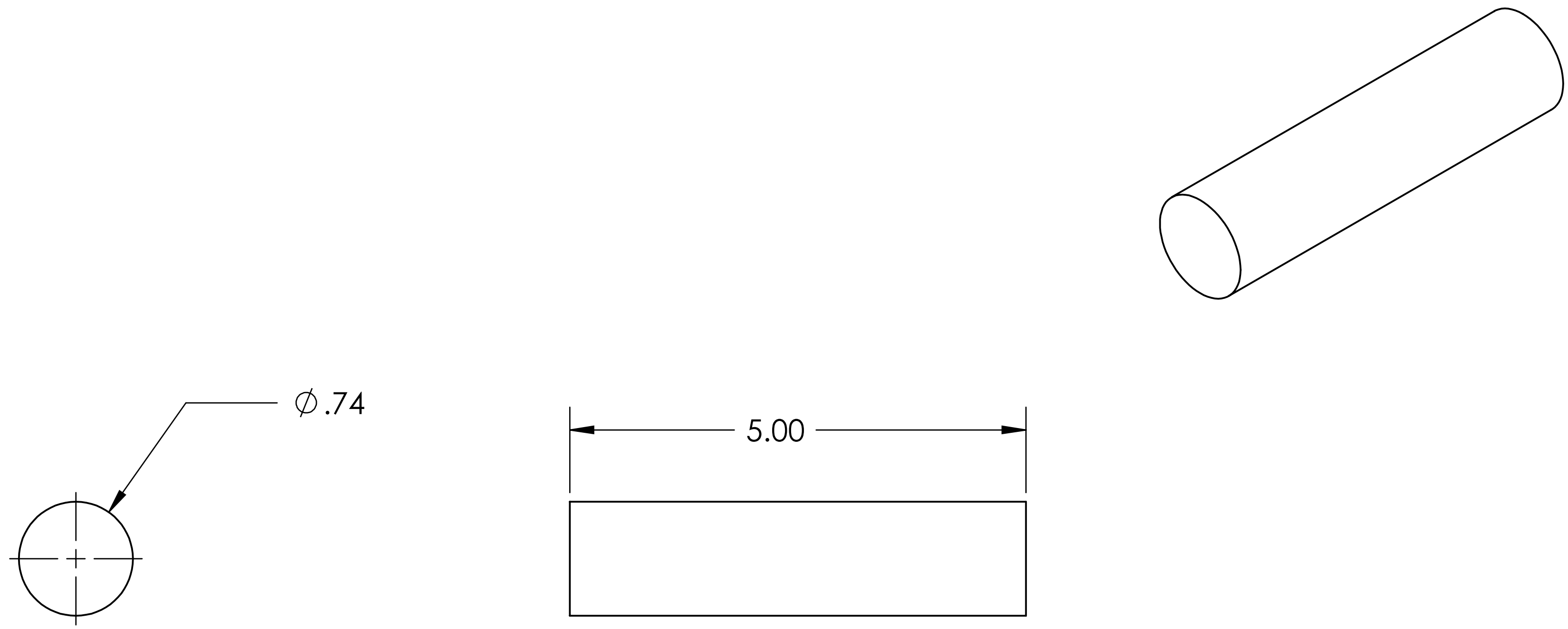
TEAM 9

TITLE:

5" PULLEY

SIZE	DWG. NO.	REV
A	1-3-002	
SCALE: 1:3	WEIGHT:	SHEET 1 OF 1

5 4 3 2 1



PROPRIETARY AND CONFIDENTIAL

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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
TWO PLACE DECIMAL ± .002

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
STAINLESS HARDENED STEEL

FINISH
AS MACHINED

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MLR	3/10/2009
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TEAM 9

TITLE:

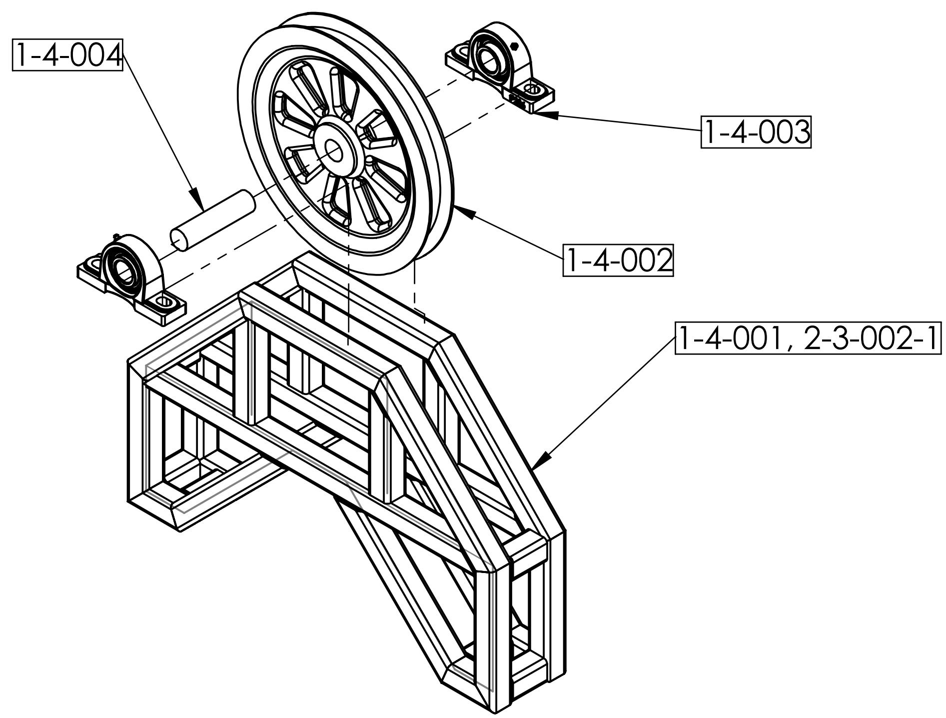
.75" PULLEY SHAFT

SIZE A | DWG. NO. 1-3-003 | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

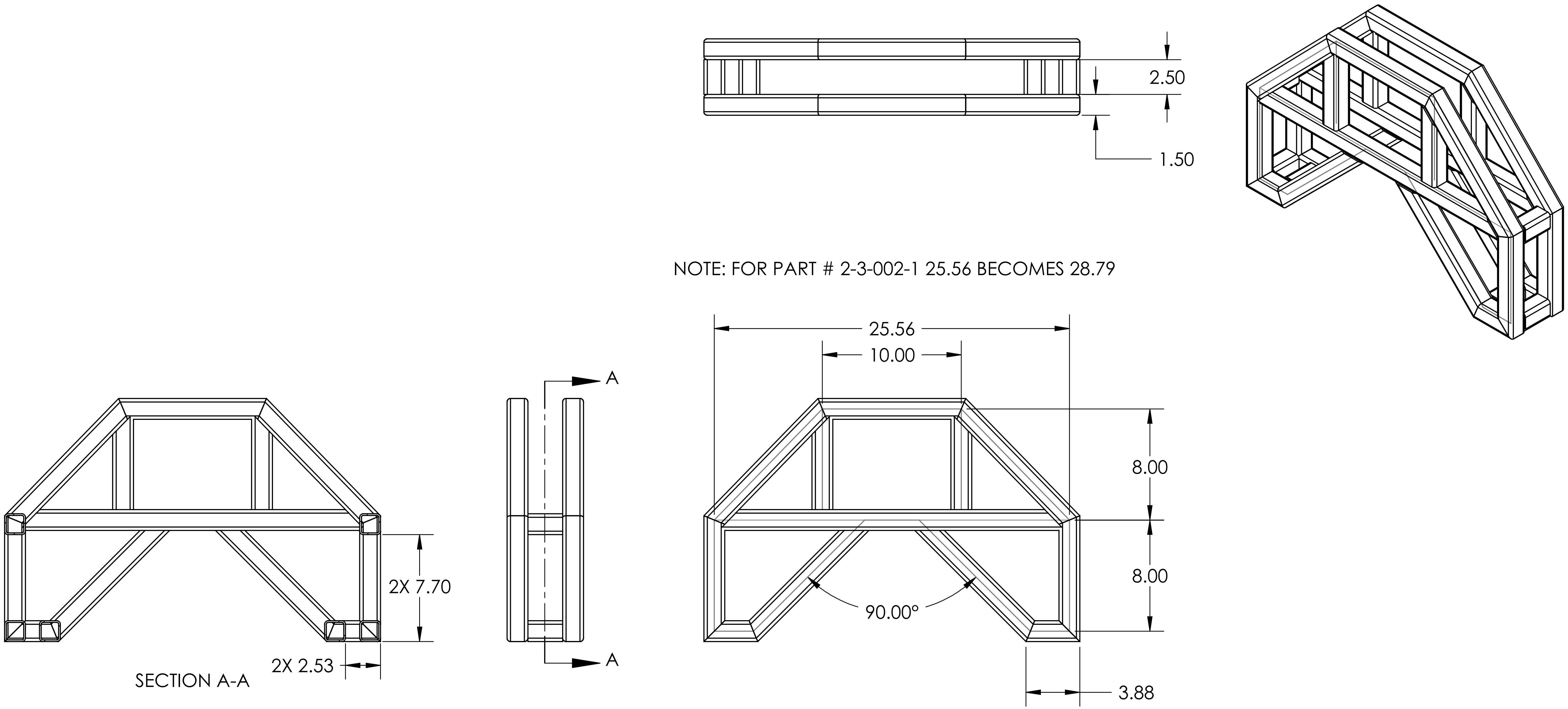
5 4 3 2 1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1-4-001, 2-3-002-1	14INCH_PULLEY_BRACKET1AND 2	1.5" SQ. TUBING WELDMENT	1
1-4-002	14INCH_PULLEY	14' DIA. PULLEY	1
1-4-003	001303	PILLOW BLOCK BEARING,UCP-206-20	2
1-4-004	1.5INCH_DIA_PULLEYSHAFT	1.5" DIA. SHAFTING	1



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TEAM 9
			DRAWN	MLR	3/8/2009	TITLE: 14" PULLEY SUBASSEMBLY
			CHECKED			
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL	COMMENTS:			SIZE A · DWG. NO. 1-4, 2-3-002 · REV
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:10 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1



NOTE:
1. COMPOSED OF 1.5'' SQ TUBING .125'' WALL
2. ALL FILLETS RADIUS .25

NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
ANGULAR: MACH ± 1°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: ALUMINIUM 6061-T6

FINISH: AS CUT

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MLR	3/8/2009
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

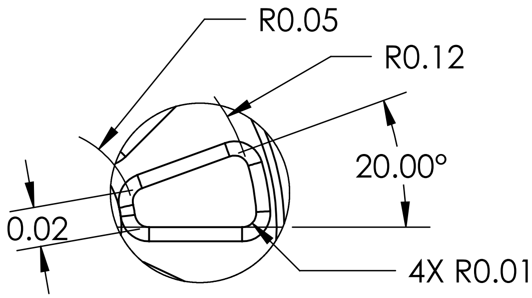
TEAM 9

TITLE:

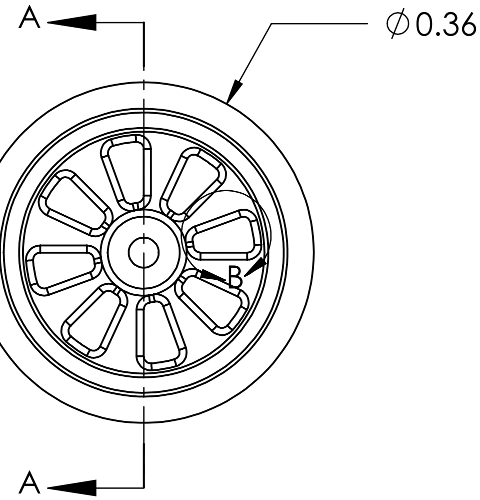
PULLEY BRACKET1

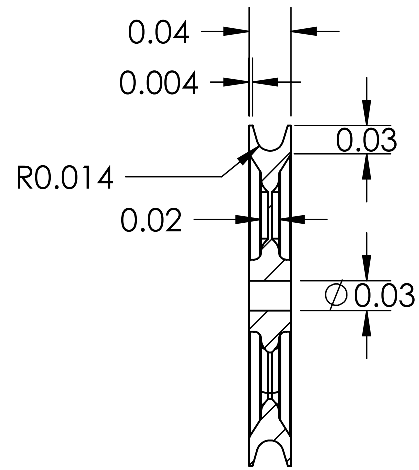
SIZE	DWG. NO.	REV
B	1-4-001, 2-3-002-1	

SCALE: 1:8 WEIGHT: SHEET 1 OF 1

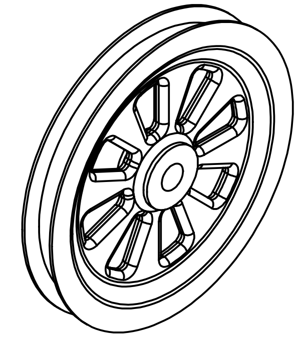


DETAIL B
SCALE 1 : 4
8 PLACES





SECTION A-A



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN METERS
TOLERANCES:
ANGULAR: MACH ± 1°
TWO PLACE DECIMAL ± .005
THREE PLACE DECIMAL ± .002
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING

NEXT ASSY | USED ON
APPLICATION

	NAME	DATE
DRAWN	MLR	3/8/2009
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

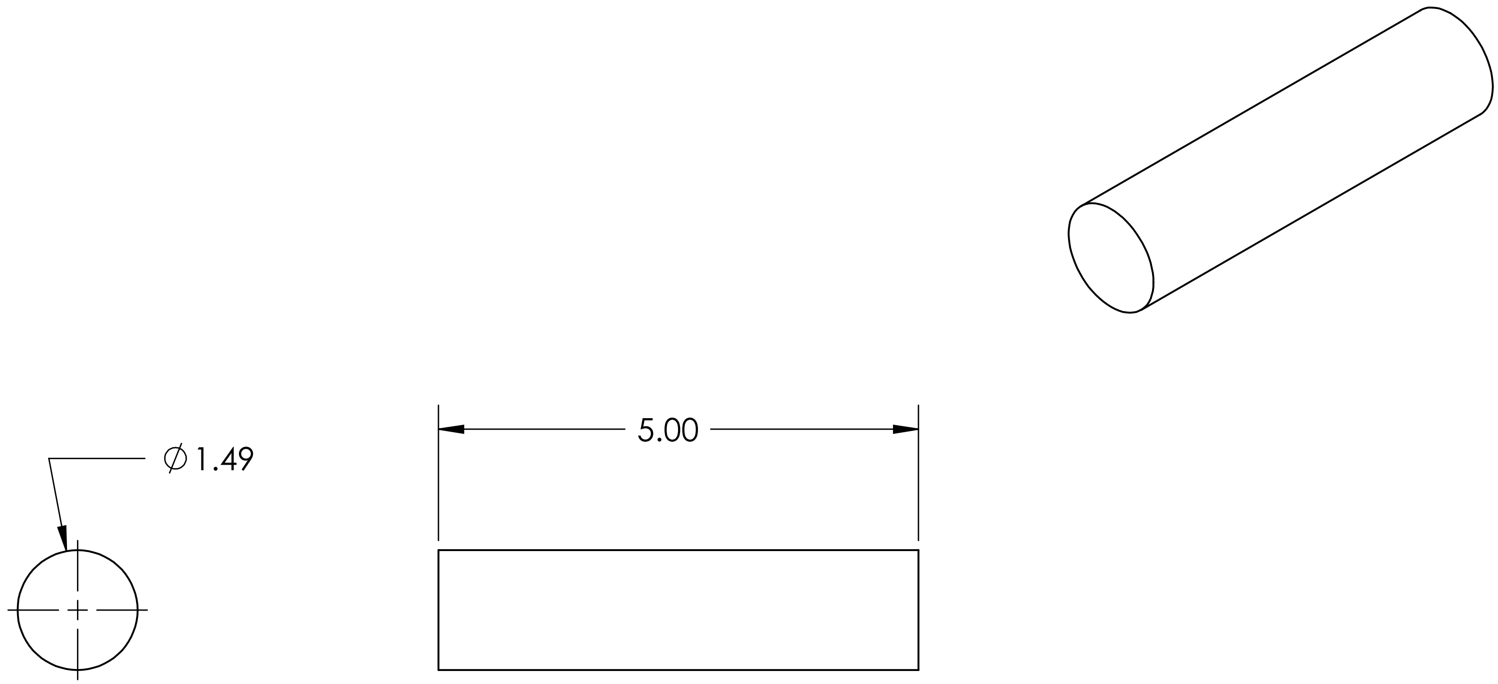
COMMENTS:

TEAM 9

TITLE:
14" DIA. PULLEY

SIZE A | DWG. NO. 1-4-002 | REV
SCALE: 1:8 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TEAM 9
		DIMENSIONS ARE IN INCHES TOLERANCES: TWO PLACE DECIMAL ±.002	DRAWN	MLR	3/10/2009	TITLE:
			CHECKED			1.5" PULLEY SHAFT
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL STAINLESS HARDENED STEEL	COMMENTS:			SIZE A · DWG. NO. 1-4-004 · REV
NEXT ASSY	USED ON	FINISH AS MACHINED				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:2 · WEIGHT: · SHEET 1 OF 1

PROPRIETARY AND CONFIDENTIAL

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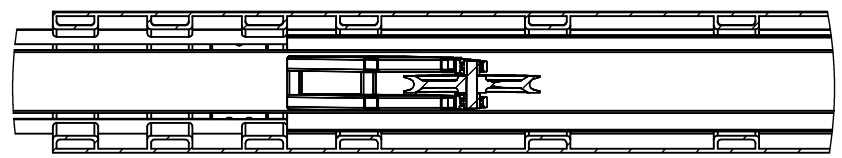
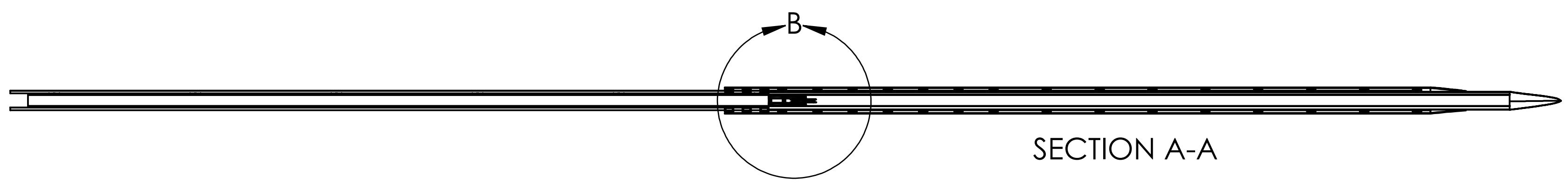
5 4 3 2 1

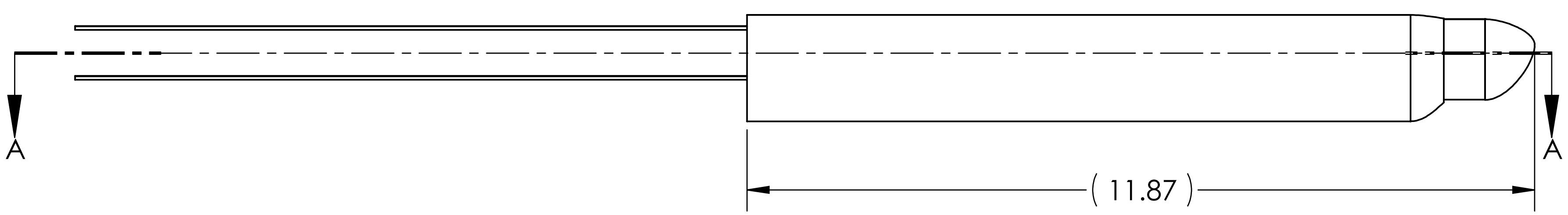
ITEM NO.	PART NUMBER	DESCRIPTION	Default/QTY.
2-1	RECEIVER_REV3	RIBS AND STRINGERS EXTENSION HOUSING	1
2-2	TRACK	U-CHANNEL SUPPORTS TIP EXTENSION LOADS	4
2-3	TIPEXT_SUB1	TIP EXTENSION, STRUCTURE, CASTERS AND PULLEY	1

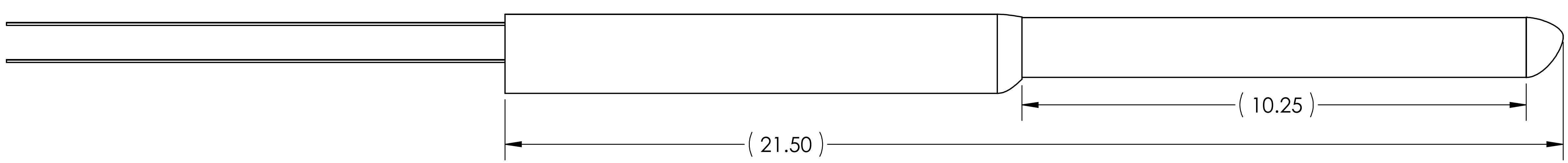


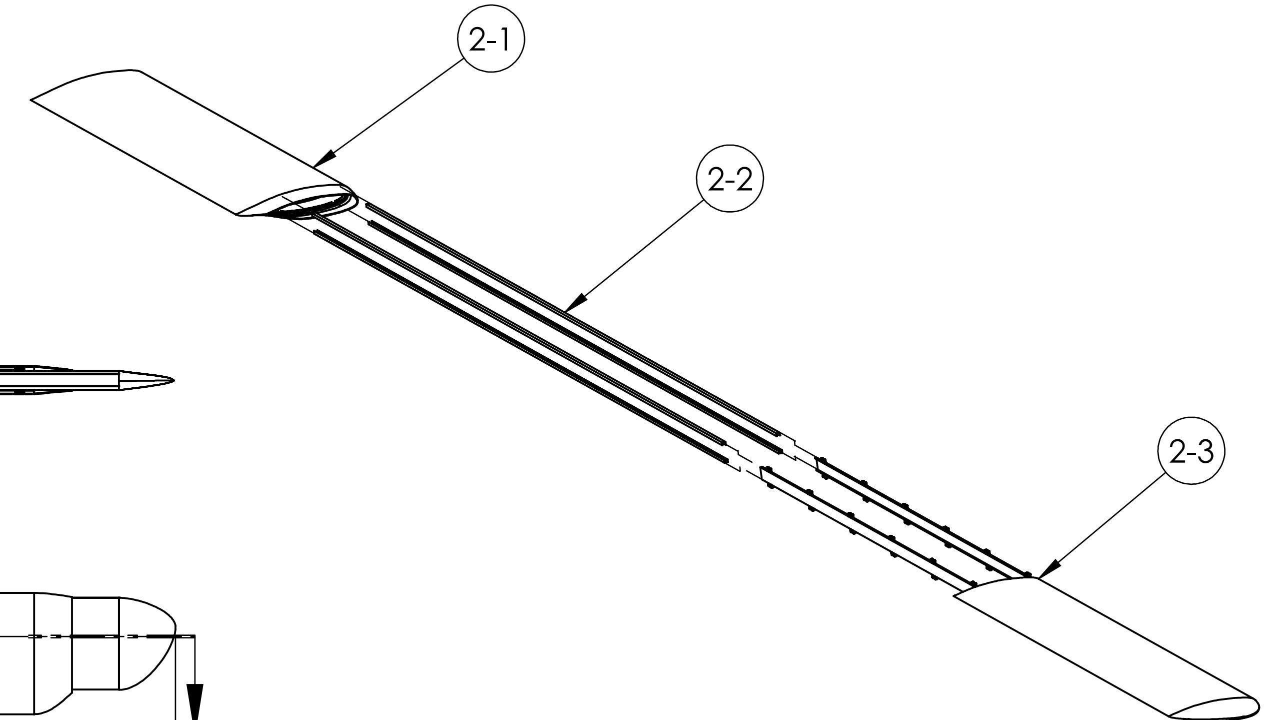
DETAIL B
SCALE 1 : 20



SECTION A-A







NEXT ASSY | USED ON
APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN METERS
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MLR	3/8/2009
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

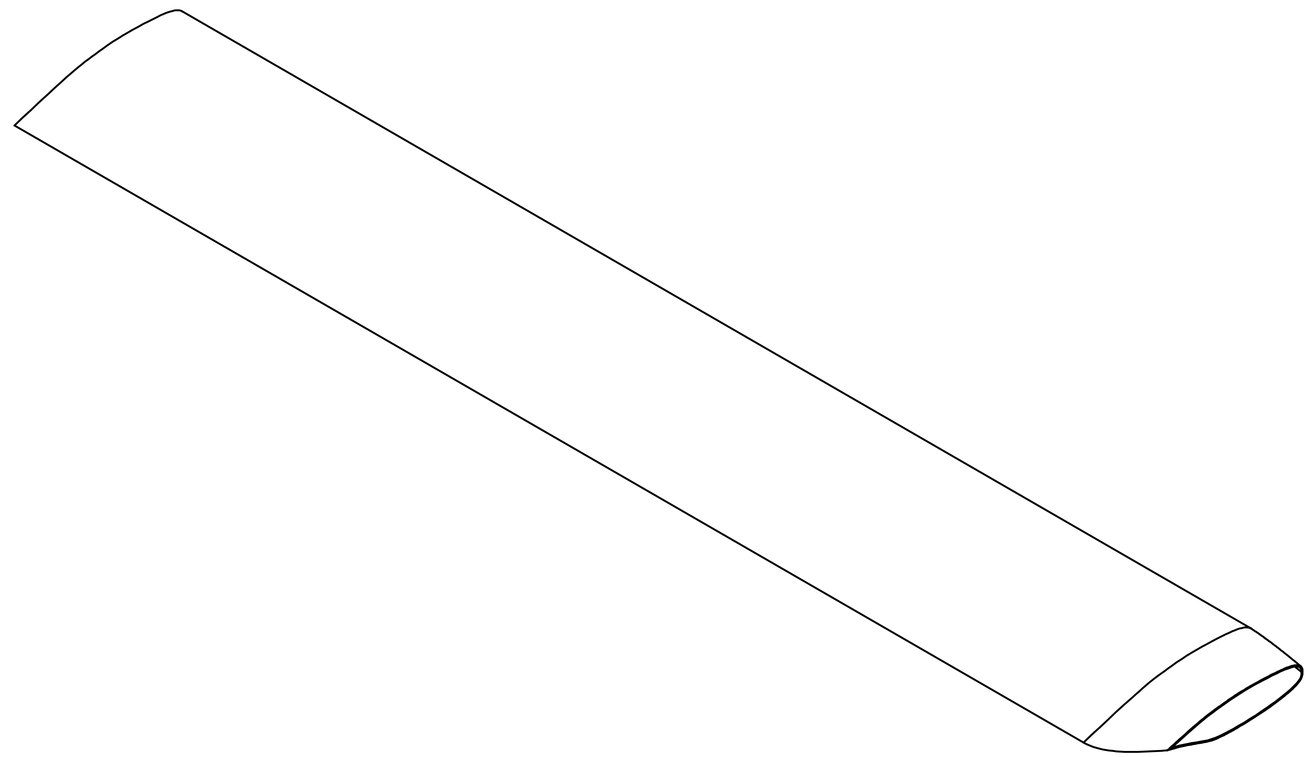
TEAM 9

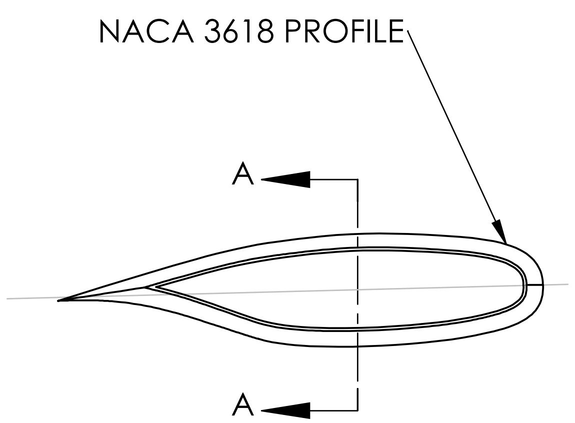
TITLE:

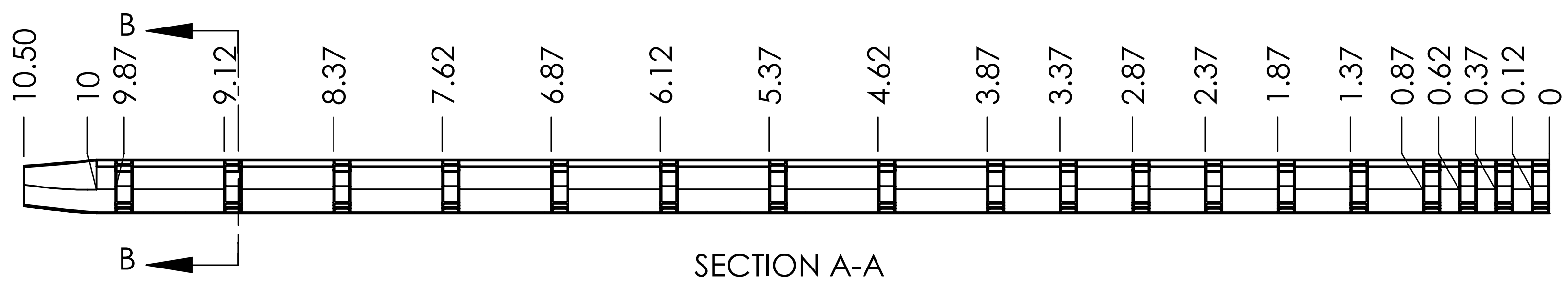
TIP EXTENSION

SIZE **B** | DWG. NO. 2 | REV

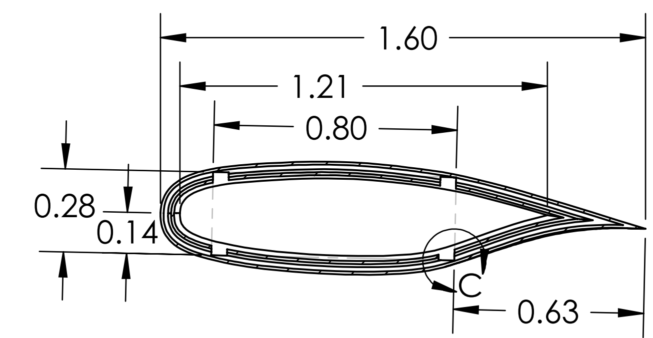
SCALE: 1:100 | WEIGHT: | SHEET 1 OF 1



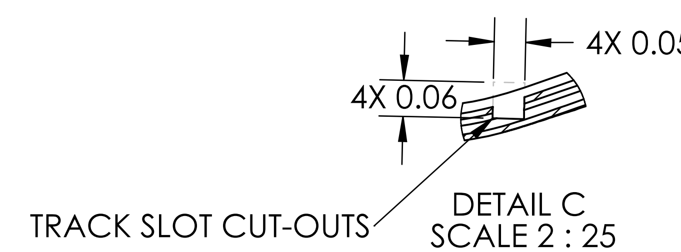




SECTION A-A



SECTION B-B
SCALE 1 : 25



DETAIL C
SCALE 2 : 25

NOTE:
1. NACA # DENOTES AIRFOIL DIMENSIONS
2. PART MOLDED AS ONE PIECE
3. REFER TO CAD MODEL FOR CRITICAL DIMENSIONS
4. STRUCTURE HAS UNIFOR THICKNESS THROUGHOUT

NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN METERS
TOLERANCES:
TWO PLACE DECIMAL ± .005
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL GRFM
FINISH WHITE CLEAR COAT
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MLR	3/8/2009
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

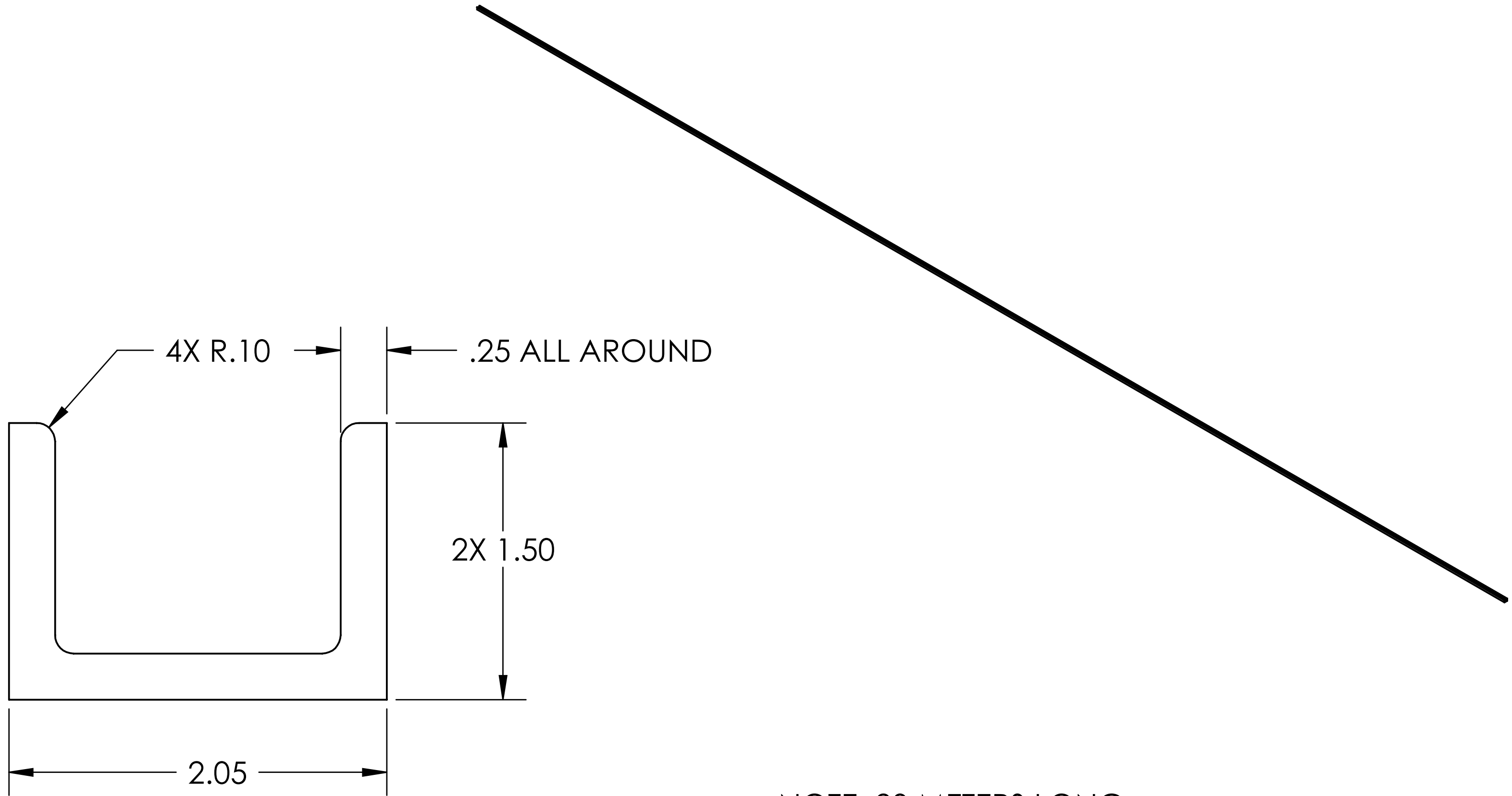
TEAM 9

TITLE:

RECEIVER_REV3

SIZE	DWG. NO.	REV
B	2-1	

SCALE: 1:50 WEIGHT: SHEET 1 OF 1



NOTE: 20 METERS LONG

		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: TWO PLACE DECIMAL ± .005	DRAWN	MLR	3/8/2009
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL GRFM	COMMENTS:		
NEXT ASSY	USED ON	FINISH CLEAR COAT			
APPLICATION		DO NOT SCALE DRAWING			

TEAM 9

TITLE:

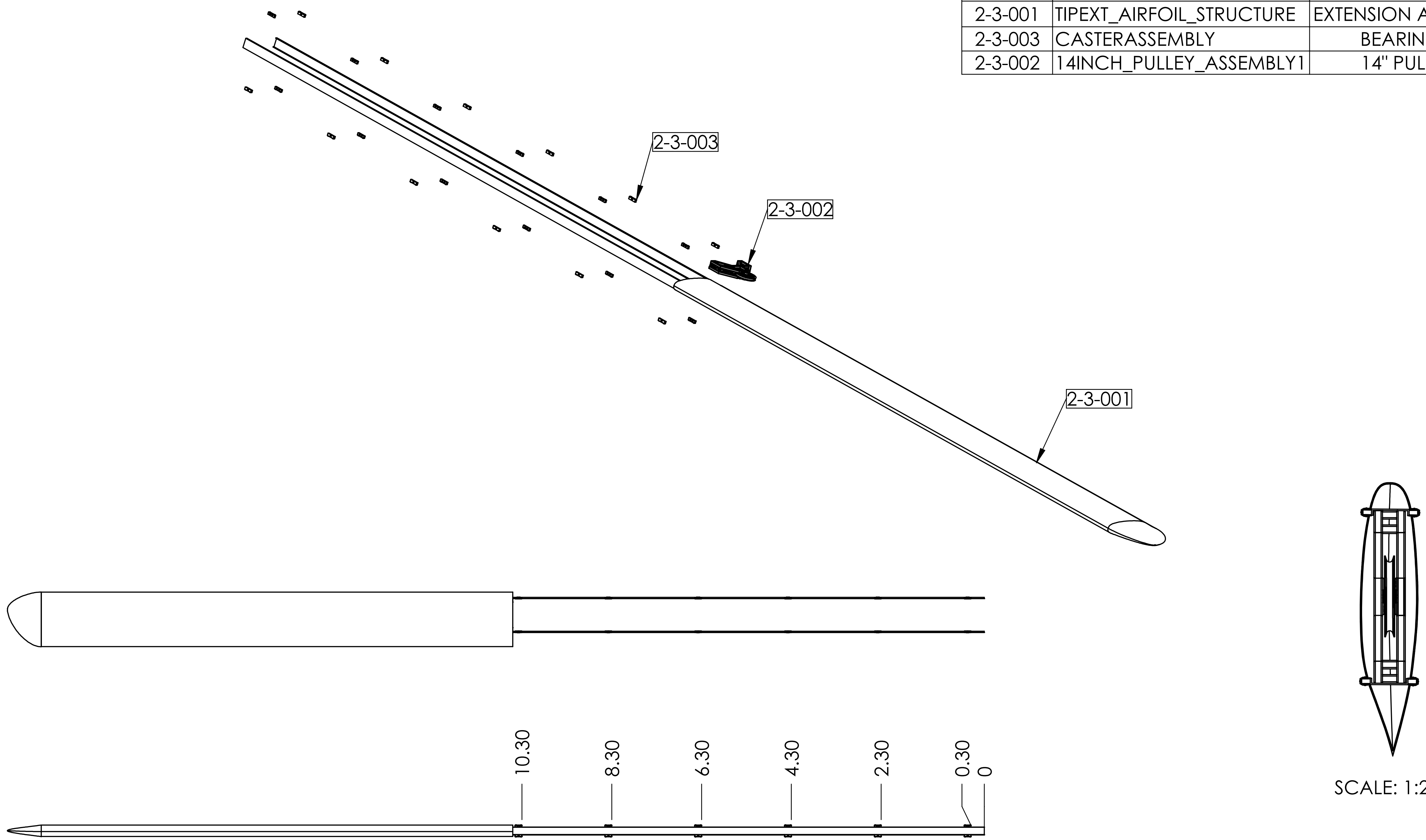
TRACK

SIZE	DWG. NO.	REV
A	2-2	
SCALE: 1:1	WEIGHT:	SHEET 1 OF 1

PROPRIETARY AND CONFIDENTIAL

5 4 3 2 1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
2-3-001	TIPEXT_AIRFOIL_STRUCTURE	EXTENSION AIRFOIL AND STRUCTURE	1
2-3-003	CASTERASSEMBLY	BEARINGS AND HOUSING	24
2-3-002	14INCH_PULLEY_ASSEMBLY1	14" PULLEY AND BRACKET	1



NEXT ASSY | USED ON
APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN METERS
TOLERANCES:
TWO PLACE DECIMAL: ± .005
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MLR	3/8/2009
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

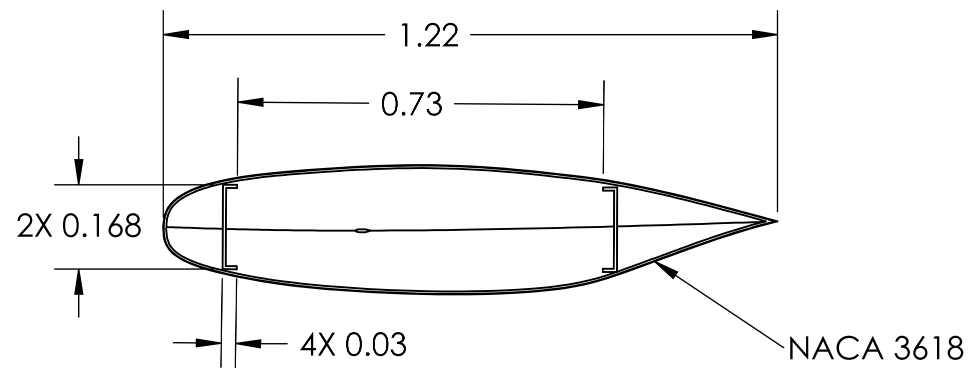
COMMENTS:

TEAM 9

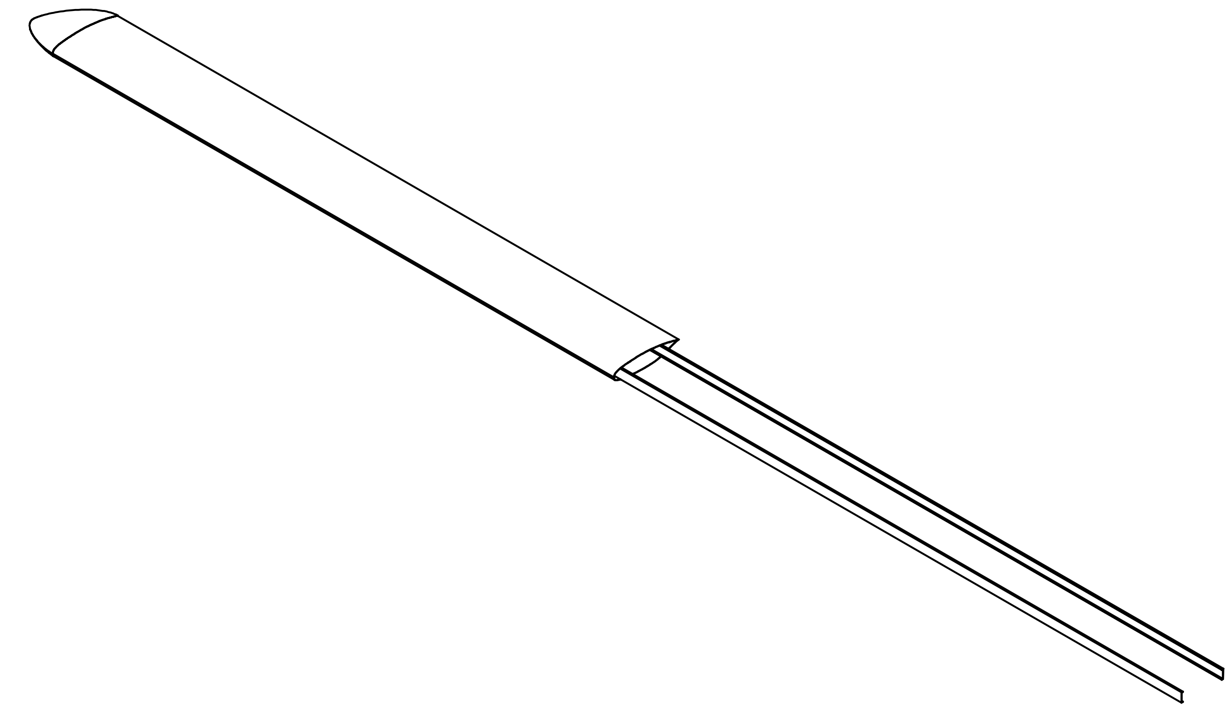
TITLE:
TIP EXTENSION SUBASSEMBLY

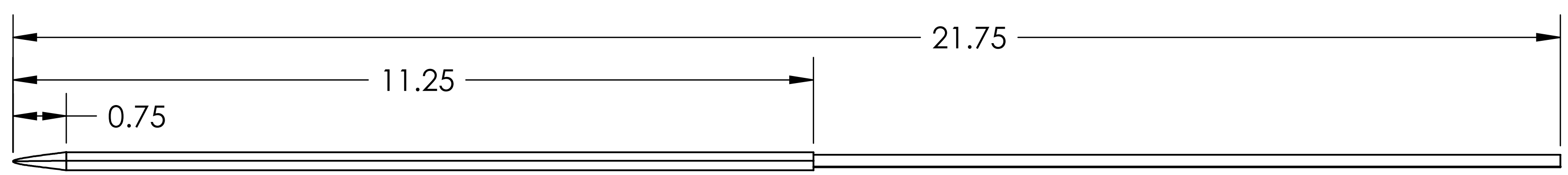
SIZE **B** | DWG. NO. 2-3 | REV

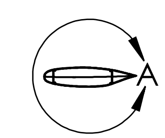
SCALE: 1:100 | WEIGHT: | SHEET 1 OF 1



DETAIL A
SCALE 1 : 15







NOTE:
1. NACA # DENOTES AIRFOIL DIMENSIONS
2. AIRFOIL THICKNESS 0.005
3. C-CHANNEL THICKNESS .015

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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN METERS
TOLERANCES:
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL GRFM

FINISH CLEAR COAT

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MLR	3/8/2009
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

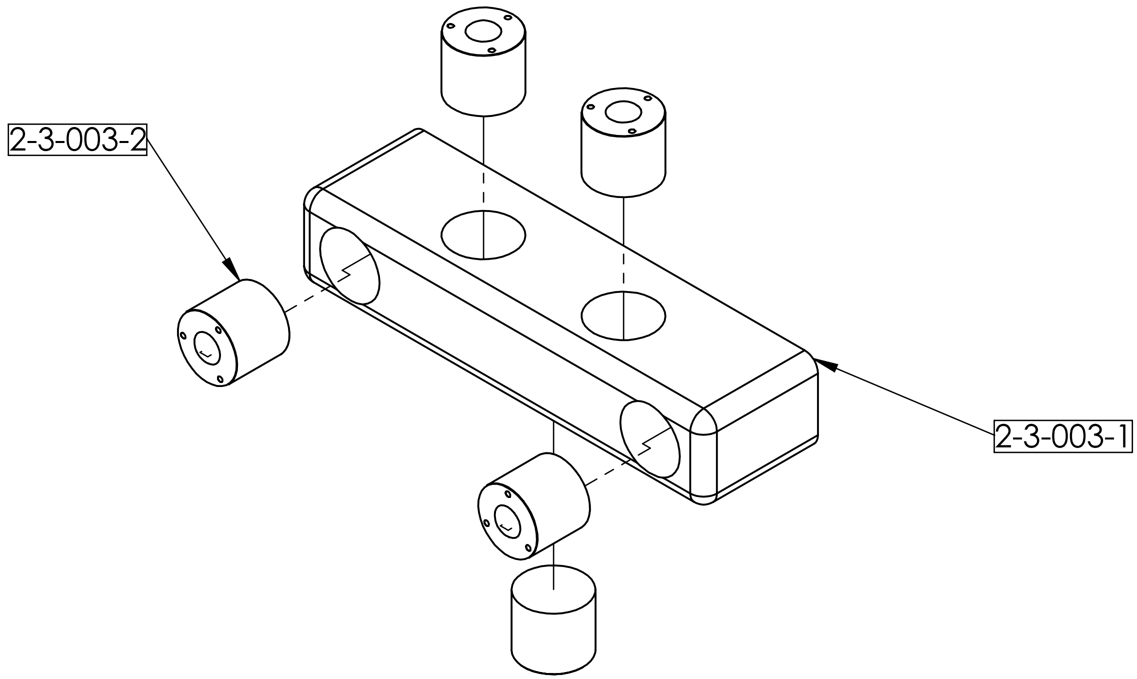
TEAM 9

TITLE:
TIP EXTENSION C-CHANNELS

SIZE	DWG. NO.	REV
B	2-3-001	

SCALE: 1:100 WEIGHT: SHEET 1 OF 1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
2-3-003-1	BALLTRANSFER_CASE	RECTANGULAR HOUSING	1
2-3-003-2	BALLTRANSFER_6460K53	BALL TRANSFER BEARING	5



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
			DRAWN	MLR	3/8/2009
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

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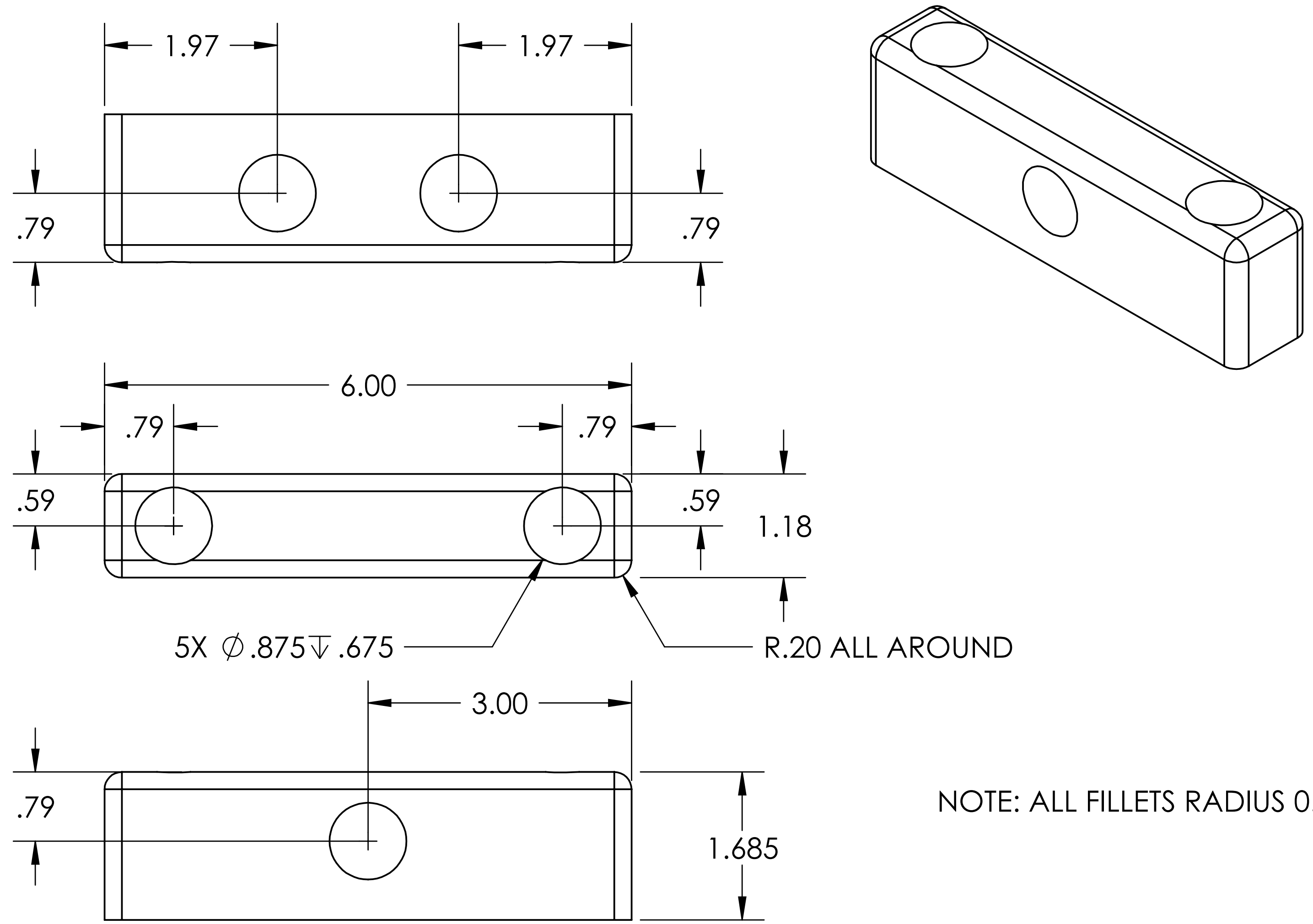
TEAM 9

TITLE: CASTER ASSEMBLY

SIZE	DWG. NO.	REV
A	2-3-003	
SCALE: 1:2	WEIGHT:	SHEET 1 OF 1

5 4 3 2 1



NOTE: ALL FILLETS RADIUS 0.2"

PROPRIETARY AND CONFIDENTIAL

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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .001

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
ALUMINUM 6061-T6

FINISH
AS MACHINED

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MLR	3/8/2009
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TEAM 9

TITLE:

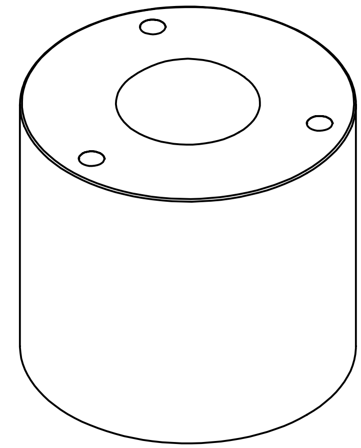
BALL TRANSFER CASE

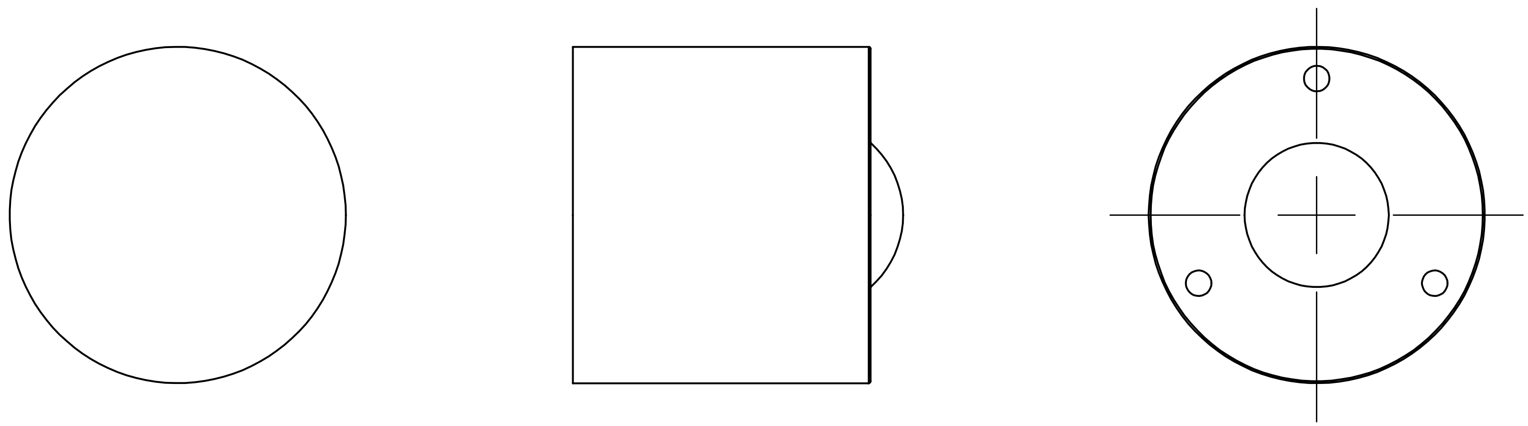
SIZE	DWG. NO.	REV
A	2-3-003-1	

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1

NOTE: MCMASTER PART NUMBER 6460K53





		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	TEAM 9
			DRAWN	MLR	3/8/2009	TITLE: BALL TRANSFER 6460K53
			CHECKED			
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL	COMMENTS:			SIZE A — DWG. NO. 2-3-003-2 — REV
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 2:1 — WEIGHT: — SHEET 1 OF 1

5 4 3 2 1