Comprehensive Design Review Report

Team #28

NREL Collegiate Wind Competition (CWC)

University of Colorado Boulder
Department of Mechanical Engineering
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Director: Roark Lanning
Claire Isenhart - Project Manager
Simon Grzebien - Finance Manager
Alec Kostovny - Logistics Manager
Aaron Schwan - Systems Engineer
Anika Levy - Manufacturing Engineer
Charles Candon - Test Engineer
Graham Blanco - CAD Engineer
Kirollos Gerges - Electromechanical Engineer
Erik Feiereisen - Electromechanical Engineer
Luke Walker - Electromechanical Engineer
Mariam Gopalani - Civil Engineer

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Table 1. List of important acronyms and abbreviations.

Important Acronym or Abbreviation	**Meaning**
NREL	National Renewable Energy Laboratory
CWC	Collegiate Wind Competition
DOE	Department of Energy
PI	Proportional Integral

Background

The Collegiate Wind Competition (CWC) is put on by the Department of Energy (DOE) in collaboration with the National Renewable Energy Laboratory (NREL). The competition was created to provide real-world challenges and a competitive work environment for undergraduates to create solutions for particular wind energy tasks. The first competition took place in 2014 as a way for the DOE to inspire young engineers to gain experience in the wind energy field. This was motivated by a goal for the U.S. to increase the electrical share of wind energy in the grid from 4% to 20% by 2030 [1]. The competition hopes to introduce students to the current challenges and barriers of creating wind turbines and working in the wind industry.

The inaugural competition in 2014 required that the competitors create a wind turbine to power small electronics such as phones and laptops. In addition, they were asked to recognize the market for the device and create a business plan to support the design [2]. Each year the competition has evolved to become more relevant to the demand in the wind energy industry. An example of this adaptive competition is in 2020, the teams were required to create an onshore wind turbine. In 2020, President Biden created a goal of 100% clean energy by 2035. The Biden administration began its term by announcing a new offshore wind energy plant in the New York Bight area, creating a goal to employ tens of thousands of employees in the offshore wind energy field, and deploy 30 Gigawatts of offshore wind by 2030 [3]. To reflect the rapid change in the wind energy industry, the CWC changed the competition for 2021 to incorporate the emphasis on offshore wind turbines. The 2021 competition challenge is to create an offshore wind turbine prototype as well as a highly efficient wind farm off the coast of Texas in the Gulf of Mexico. This will allow students to pursue and learn offshore technology and analysis.

Introduction

For the 2021-2022 CWC, the competition has shifted to a focus on offshore wind. In this scenario, each team's turbine will have to sit in a large tank full of sand and water to mimic the conditions of an offshore wind turbine. With this new scope, the competition committee placed a greater emphasis on the importance of a stable and unique offshore foundation. Teams are now able to commit greater focus to foundation as the competition reduced the importance of a robust yaw system. This provides important context for the design decisions discussed throughout the report. The CU Wind Team is broken up into six subteams to better tackle this year's challenge: Electrical, Yaw, Pitching, Blades, Nacelle, and Foundation/Tower subteams. The CWC will allow our team to grow in their fundamental understanding of wind energy technology.

Electrical

Design Requirements

There are numerous requirements that need to be followed to ensure our design is safe for use in the competition. One of the main requirements is that all turbine controls need to be contained within enclosures that meet the NEMA Type 1 rating. Along with this, all electrical components must be mechanically secured to the enclosure and must be electrically insulated from the enclosure. Going off of this, all cables must use cable glands that provide strain and chafe protection and must have connectors that are not submerged in the water. Another competition requirement is that the output voltage from the turbine must be DC at the point of common coupling (PCC) and below 48 Volts. The wires should be terminated with Anderson Powerpole connectors and all cables should be the correct gauge size to allow for proper current carrying capacity.

Specific to the turbine controls, there must be no more than 10 Joules of energy storage on the turbine side of the PCC. This means no batteries are allowed on the turbine side of the controls. However, bulk energy storage is allowed in the load. The turbine may draw power from the load if needed but must register a zero state of charge at the beginning of the competition test and all wired connections between the turbine and load that are external to the PCC must be optically isolated as to allow for no power transfer except through the PCC.

Lastly, the turbine must be capable of shutting down on command through two different initiation sequences. First, a physical button will be pressed and the emergency stop sequence will be initiated. Second, the load will be disconnected and again the emergency stop sequence must be initiated automatically. We must provide a cable terminated with a JST RCY female receptacle housing to transfer the E-Stop signal from the PCC to the turbine controls.

Preliminary Design

The electrical system must be designed to safely and efficiently deliver power to all of the turbine controls as well as the load. The first part of the electrical design was to choose a generator to convert the rotational energy from the wind into electrical energy. We determined two options for choosing a generator. We can either design and manufacture a generator ourselves or purchase this component. There are advantages to both options, for example, building a generator allows the team to have a lot more control over the specs that will be optimized. On the other hand, buying a generator will likely result in higher efficiencies due to the manufacturing expertise of motor suppliers, which is ultimately why our team decided to purchase this component. This decision also was due to the limited resources of our team (i.e. fewer members than other competition teams) and the lack of electrical engineering expertise.

The next choice for our team was to choose the type of motor that would work best when used as a generator. The obvious candidates for this were brushed DC motors, brushless DC motors, or stepper motors. From the start, the team decided to eliminate stepper motors as an option because of the extremely high voltages they generate. One source explained that stepper motors can generate about 20 V when turned by hand [5]. This was also verified experimentally by feeding the output of a stepper motor into a full-wave rectifier and observing the voltage.

The next decision was choosing between a DC motor and a BLDC motor. The team decided to use a 3-phase BLDC motor because it can be used to slow the rotation of the turbine during the emergency stop. A property of 3-phase motors is that a large magnetic field is created when the terminals are shorted. This magnetic field adds significant resistance to rotation and can be used to arrest motion. This is important for the emergency stop scenario which requires the turbine to reduce its speed to 10 percent of the original speed in 10 seconds. The emergency stop scenario occurs when the load is disconnected. This means that no power can be drawn from the load. Given that the generated power will be diminishing as the turbine slows to a stop, there will be no steady source of power during this scenario. This means that 'active' emergency stop methods are very difficult. 'Active' emergency stop methods use power to slow rotation. An example of an active emergency stop is a servo that presses on the rotor to apply a braking force. This emergency stop approach was avoided because of the limited power when the load is disconnected. The team decided to use the passive method described above because it takes very little power to shorten the terminals of a 3-phase generator.

Additional considerations for choosing a specific BLDC are optimizing specifications. The team determined that the most important specifications were to minimize the speed constant, minimize the cogging torque and rotor inertia, and minimize the terminal resistance. The speed constant specification determines the output voltage at a given rotational speed. The specification is minimized so that the generator will generate maximum power. The next considerations were the cogging torque and the rotor inertia. Cogging torque refers to the torque needed to overcome the internal resistance in the generator to begin rotation. The internal resistance is caused by friction in the bearings and the magnetic flux that is generated by the windings. This specification, as well as rotor inertia, are minimized to ensure that the turbine can begin rotating faster and at lower wind speeds. The final important specification to consider is the terminal resistance. Minimizing this is desirable because the current capacity decreases with increasing resistance [6]. We desire a high current carrying capacity so that we can extract as much energy from the wind as possible.

The next major decisions for the electrical system considered how to implement an electrical control system that would meet the CWC rules and regulations as well as accomplish all the tasks that are needed to generate power. The preliminary design for the turbine controls is detailed in the functional block diagram shown in Figure 1. The choice to use a 3-phase generator necessitates a circuit to rectify and filter the 3-phase sinusoidal output. Additionally, because of the varying wind speeds, the generator will not always be outputting a steady voltage. This means that a regulator circuit is needed to turn the varying voltage into a steady DC voltage. The next circuit that is necessary is a current sensor that can be used to monitor the connection status of the load (for the emergency stop scenario mentioned earlier) and to monitor the power output to the PCC. The PCC connection is where the turbine controls connect to the competition DAQ. The competition scores are based on power output and thus they need to monitor the connection between the turbine controls and the load.

The load must be designed by the team prior to competition. The purpose of this system is to simulate the grid load that a full-scale wind turbine would connect to. The specifications for the load given by the CWC are limited which means the team has a large degree of freedom when designing. The purpose of the load is to sink the power generated by the turbine. Some preliminary ideas included using generated power to charge a battery or simply using a power resistor to turn generated energy into heat. Other design considerations for the load are whether or not to draw power from the load. The competition rules state that we can plug our load into the wall to draw power when needed. Alternatively, if the load was designed around charging a battery, power could then be drawn from the load when needed.

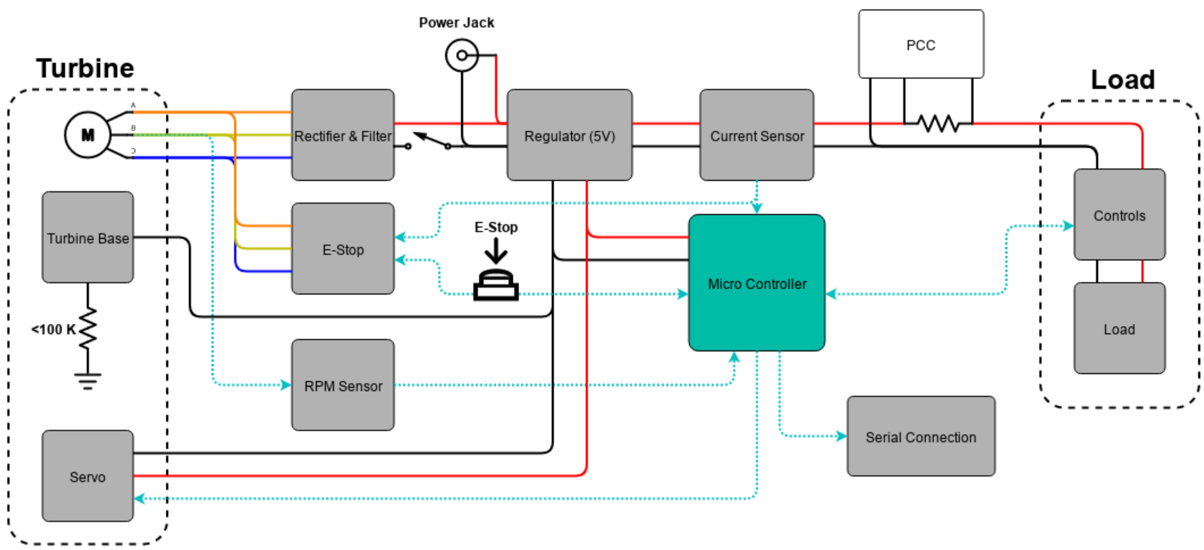


Figure 1. Preliminary design of the electrical system. The red and black connections show the transfer of power and the dotted blue lines show the transfer of information.

Final Design

This section will focus on the current design and prototyping status of the turbine controls and load. It is necessary to begin by discussing an updated version of the block diagram from the preliminary design phase (figure 1) shown in figure 2. The biggest change between the two versions is that critical decisions have been made about the load and how to handle the emergency stop scenarios. As discussed previously, there are two emergency stop scenarios: the emergency stop button is pushed or the load is disconnected. The turbine must be able to restart once an emergency stop has occurred. To accomplish this, a bimodal approach was taken. The first mode (figure 2, top), deemed “startup”, is active once an emergency stop signal is received or when the system loses power. In this mode, the turbine controls draw power from the load.

The second mode (figure 2, bottom) is called “normal operation” and in this scenario, the turbine controls draw power from the generator, and generated power is also sent to the load. These two modes are controlled by a set of switches in the circuit. These switches will either be NMOS or PMOS depending on if they are used in a pull-up or pull-down configuration. They will be logic level MOSFETS so that they can be controlled with a single Arduino signal (figure 2, dashed purple lines). When the signal is low (0V) the system is in startup mode. When the signal is high (5V) the system is in normal operation mode. The benefit of this approach is that even if the turbine controls lose power because the load has been disconnected, the system will default to startup mode so that load power will be available once the load is reconnected.

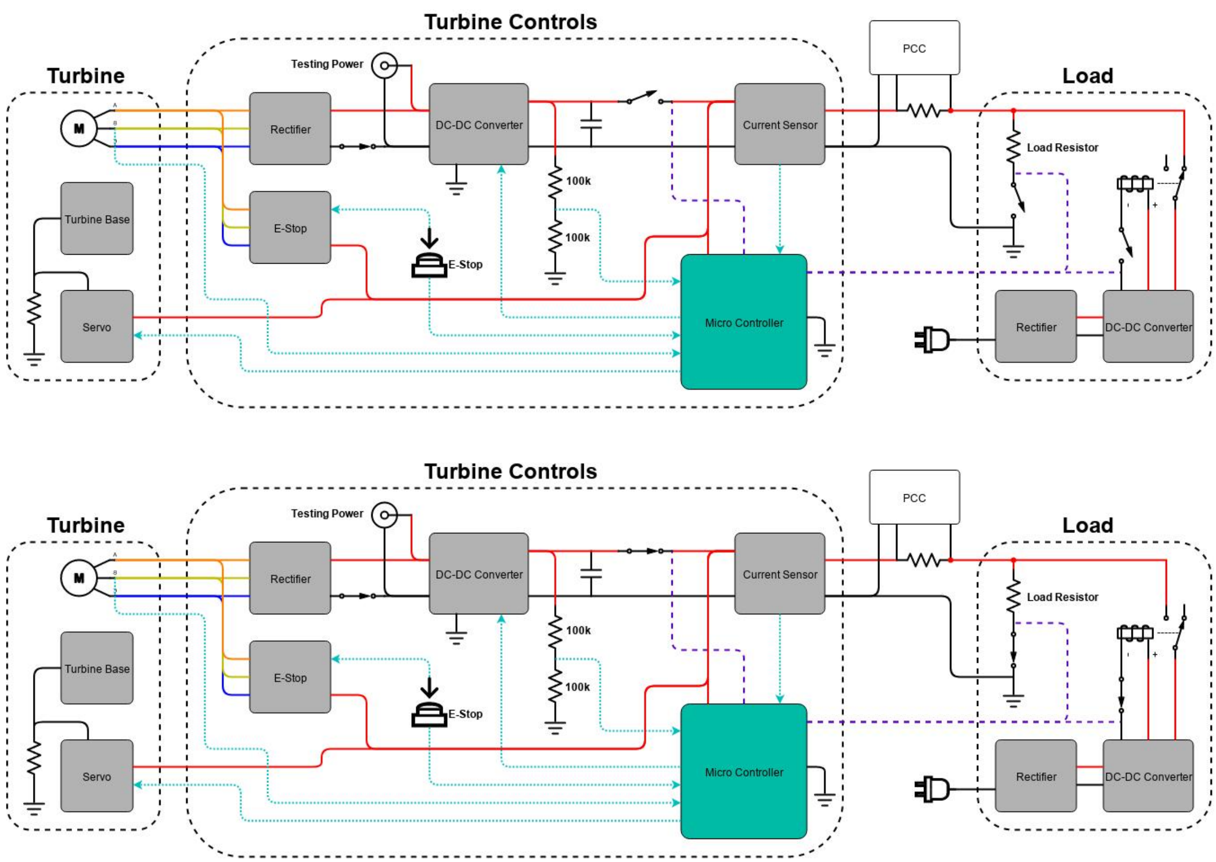


Figure 2. Updated electrical system block diagram. “Startup” mode is shown on the top and “normal operation” mode is shown on the bottom. Red and black connections represent a transfer of power and dotted blue lines represent a transfer of information. Dashed purple lines are the signal that changes operation modes.

Generator

The first major decision that was made was choosing the specific generator to use with our turbine. The team chose the EC-i 40, brushless, 100 Watt motor from Maxon (PN: 488607).

This motor features a speed constant k_n of 105 $\frac{RPM}{V}$ which is low compared to other generators. A low-speed constant is imperative as our turbine will be spinning at a maximum of 2000 RPM and we would like to generate the highest voltage possible within the RPM range our turbine will be operating in. This means that at a nominal rotational speed of 2000 RPM the generator would be outputting approximately 19V DC. Next, this generator has minimal cogging torque and rotor inertia of 44 gcm^2. These specs will ensure that the turbine can begin rotating at low wind speeds. Finally, the terminal resistance of this generator is 1.01 Ω. Again, a low terminal resistance is necessary because it means more current can theoretically be drawn from the windings.

Rectifier

The circuit that is needed to convert the 3-phase AC generator output into a DC voltage is a 3-phase full-wave rectifier. The schematic is shown in figure 3. This circuit consists of six Schottky diodes that effectively take the absolute value of the input voltage. That is, the negative portions of the input sinusoid from each phase will be flipped so that they are positive. Then a smoothing capacitor is placed across the output of the rectifier to create a steady DC voltage. The diodes used in the circuit will have the lowest possible forward voltage drop while at the same time having a reverse voltage higher than the maximum voltage output from the generator. The chosen diodes (PN: B340B-13-F) have V_F=0.5 V and V_R=40 V. This will help minimize losses in the rectifier circuit while making sure the diodes do not enter the breakdown region.

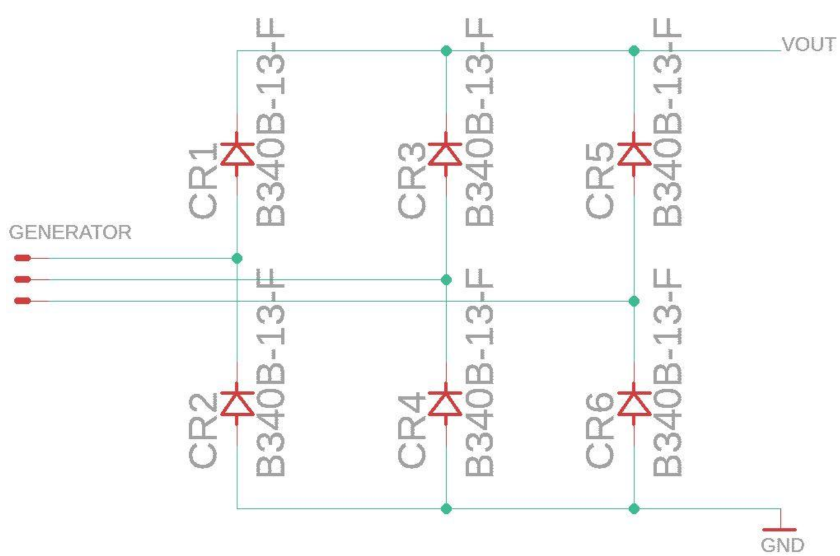


Figure 3. Schematic of 3-phase full-wave rectifier.

DC-DC Converter

The DC-DC converter that will be used is a single-ended primary-inductor converter (SEPIC). This circuit is necessary because the output voltage of the rectifier is proportional to the rotational speed of the generator. The turbine controls need a constant voltage to operate and therefore the varying output voltage of the rectifier is not sufficient. The advantages of the SEPIC are that it has buck-boost capabilities, meaning it can maintain a steady output voltage with input voltages both below and above the output voltage. Additionally, the SEPIC does not have an inverted output like a traditional buck-boost converter. A schematic of the SEPIC is shown in figure 4. The circuit element values have not been fully decided upon because the SEPIC output current is still unknown [9]. This value is largely a function of the load resistance which is discussed in a later section of this report. The power MOSFET that will be used is the IRL520NPBF because of its high continuous current, low drain-source resistance, and good thermal characteristics. During operation, a switching signal is sent to the gate of Q1 with a microcontroller. The signal is a square wave with a frequency in the hundreds of kilohertz range. The actual switching frequency that will be used in the turbine controls has not yet been decided. The output voltage of the SEPIC is controlled by the duty cycle of the switching signal. Therefore, a feedback loop will be utilized to measure the output voltage and adjust the duty cycle to keep it at a steady value.

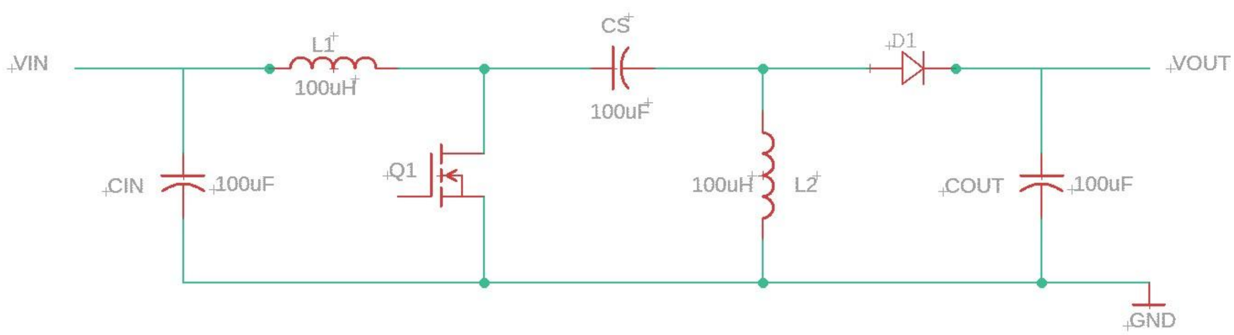


Figure 4. SEPIC schematic. Circuit element values reflect current testing values, not final values.

E-Stop

After some deliberation, it was decided that the emergency stop system would resemble last year's (CU Wind Team 2020) system. Last year's emergency stop was mainly electrical and used the fact that shorting the terminals of a three-phase BLDC motor would add a lot of resistance to the shaft, therefore, slowing the turbine. The reason we decided on an electrical emergency stop system was mainly due to power consumption considerations. For the emergency stop scenario in which the load is disconnected, there is no way to draw power from the load to power our controls and mechanically slow the shaft. It is understood that the turbine will continue to generate power as the turbine slows, and therefore the microcontroller will still be able to be used, but the power output of the microcontroller will decrease as the shaft slows.

Our design uses an N-Channel Metal-Oxide-Semiconductor Field-Effect Transistor (MOSFET) that will read a small output voltage from the microcontroller. A MOSFET is similar to a Bipolar Junction Transistor (BJT) in how it functions by allowing current to flow when a threshold input is read at the gate, but the difference between a BJT and a MOSFET is that a MOSFET is voltage controlled and will allow current to flow when a threshold voltage is reached. When a threshold voltage of 0.5 Volts is applied to the MOSFET, current will begin to flow through the transistor thus shorting the source and drain of the MOSFET. This will allow current to flow through an opto-isolator. An opto-isolator is an electrical component that transfers electrical signals between two isolated circuits using an infrared (IR) LED. Once current begins flowing through the MOSFET (after a threshold voltage is reached at the gate), the IR LED will light and transfer a signal to the photo MOS which will open the circuit allowing current to flow through each terminal of the generator thus shorting the terminals and applying a resistance to the shaft. However, a question did arise in that as the terminals to the BLDC motor are shorted, will the current flow in the winding cause the motor to heat up to an unsafe temperature? To answer this question, a calculation was performed to determine the absolute worst possible scenario for change in temperature of the motor, and the equation (Equation 1) is shown below.

$$Q = mc_p \Delta T \tag{1}$$

Here, $\Delta T$is the unknown value that we would like to figure out. We used a mass *m* of 39.78 grams. This value is the mass of copper wire in one motor and was found after consulting with a sales engineer at Maxon. We also used the specific heat of copper $c_{p,\, copper}$ = 387.279 $\frac{J}{kgK}$. Lastly, to determine the heat transfer Q, the output power must be known by using equation 2.

$$P = \frac{V^2}{R} \tag{2}$$

Using the speed constant k_n of the motor, which was 105 $\frac{RPM}{Volt}$, we can determine the voltage output of the motor at a specific RPM. Due to mechanical constraints discussed later in the report, the max RPM of our turbine is 2000 RPM. At this speed, the terminals would likely overheat the generator, therefore, this emergency stop method would only be implemented after the blades have been pitched parallel to the wind and the RPM of the shaft drops below 1000 RPM. Therefore, at 1000 RPM, we will likely get an output voltage of around 9.5 volts. Dividing this value by the terminal resistance of the wires, which is 1.01Ω, we get a power output of 89.8 Watts. Now, the emergency stop system must slow the shaft to 10% of the max speed of our shaft within 10 seconds. Continuing with the worst-case scenario, it is assumed this braking method will last for the entire 10 seconds. Therefore 89.8 W * 10 sec =898 Joules. This value of 898 J is equal to the heat transfer from the wires to the motor and is Q. Now, solving for ΔTin equation 1, we find that the worst-case change in temperature for 1000 RPM is $\Delta T = 58.3°C$. The max operating temperature of the generator is 150°C, so as long as the nominal operating temperature of the generator is below 90°C, the generator will not overheat and fail.

Components were chosen carefully to ensure that this system will function properly. The MOSFET (PN: RD3S100CN) was chosen based on its low threshold voltage needed to initiate current flow through the circuit. The threshold voltage of this MOSFET is 0.5V meaning an input of half of a volt to the gate will allow current to flow. After running tests with multiple types of MOSFETS, it was determined that the input voltage to the gate should be higher than the threshold voltage to ensure proper current flow. Therefore, we carefully chose resistor values that allowed for a voltage input of 2 volts at the gate, which is well above the threshold voltage and will allow proper current flow.

After careful consideration and research, we decided to use the same opto-isolator as last year's team (PN: AQZ202). This opto-isolator had reasonable characteristics that would allow for low power usage during emergency stops including a small LED operating current and forward voltage (V_F) drop. Figure 5 illustrates the three circuit systems that will short all three wires together and figure 6 shows a single subsection of the three circuit systems.

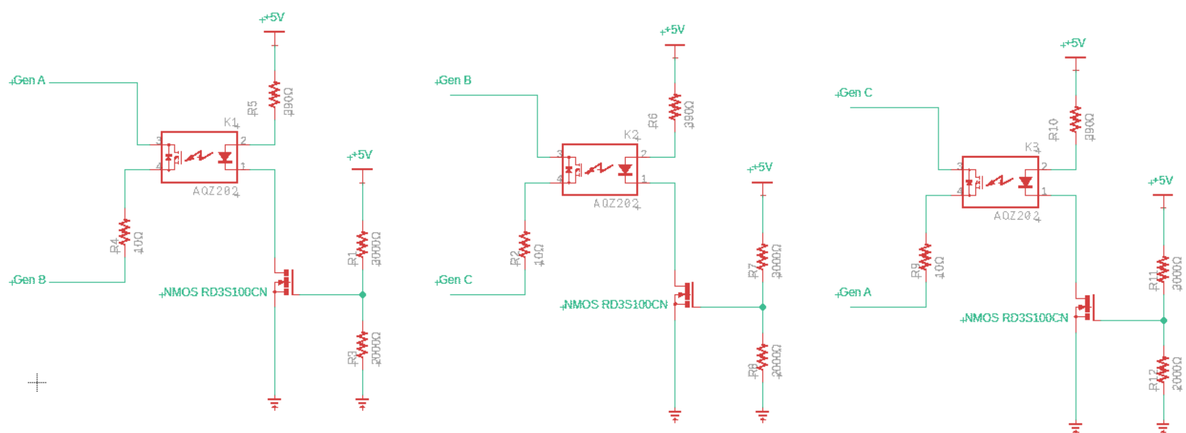


Figure 5. Emergency stop system showing all three terminal subsections shorted together.

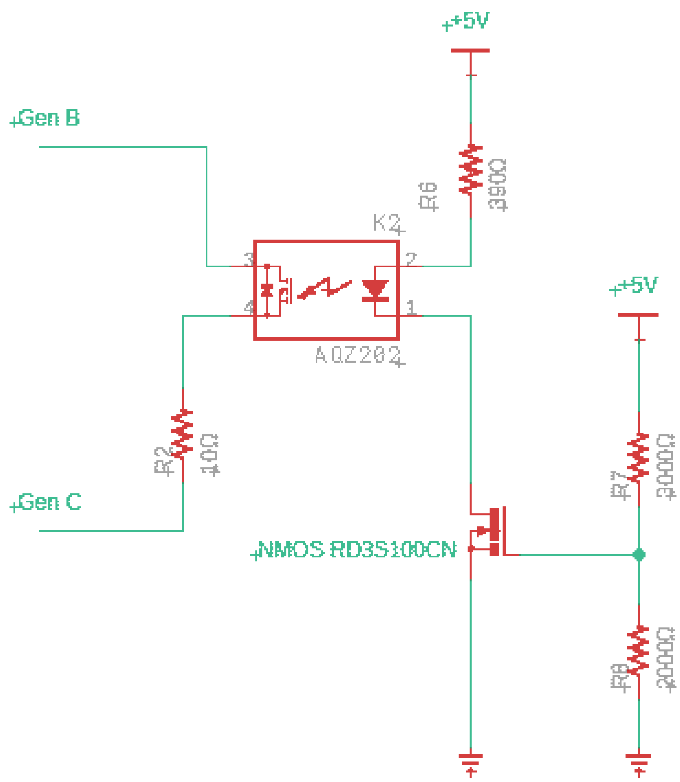


Figure 6. One subsection of the emergency stop system showing the parts numbers and their corresponding values.

Current Sensor

The current sensor is a major component of note (Figure 1) in the electrical system block diagram. The current sensor is responsible for:

1. Calculating the electrical power output that is fed to the PCC and then to the load.
2. Providing information on whether the load is connected or disconnected.
3. Determine whether to run our microcontroller and the servo motor with the available power from the generator or if we need to withdraw power from the wall.

This component will be a Sparkfun current sensor (PN: ACS723) which can sense low to moderate AC and DC current by using a Hall effect sensor to output a voltage that is proportional to the current registered by the sensor. The sensor can sense as low as 10 mA, and as high as 5 A of current passing through it, and is rated up to 2.4kV. Therefore, the current sensor perfectly suits the application. In addition, it has a built-in gain amplifier that could be adjusted to limit the current to a specific range to improve the accuracy of the control system. It also has an adjustable reference voltage that lets us choose between measuring AC or DC current.

Load

The load is an open-ended portion of the design as there are only two specifications outlined in the Rules and Regulations.

1) Bulk energy storage is allowed, such as a battery or capacitor to store charge
2) It is permissible to lug into the wall and use the 120 volts AC to run the load.

The team's load design, shown on the right side of figure 2, is relatively simple in theory. After the direct current runs through the PCC and is counted towards the power generation of our turbine, it is then dissipated in the form of heat over a power resistor. This design was chosen over a battery as there was no apparent need for bulk energy storage since we can be connected to a wall outlet of 120 volts AC. As seen in figure 2, the load changes between "Startup" and "Normal Operation" modes. In a startup, the turbine will initially not be generating any power, so we use the 120 volts AC provided to us. We rectify and regulate the AC and turn it into 5 volts DC where it is then sent backward through the PCC and into the turbine control system to power the microcontroller and servo for pitching. The microcontroller then tells the servo to pitch the blades to an optimal angle of attack. This optimal angle will then generate a force on the blades thus rotating the shaft and generating power. The startup phase continues until the microcontroller reads a value of 300 rpm from the hall sensor. Based on the speed constant of our generator, there should be about 3V DC at the output of the rectifier. At this point, the microcontroller will begin generating the switching signal needed to control the SEPIC converter with a duty cycle that corresponds to 3V input. Simultaneously the microcontroller will set the mode control signal to 5V to switch into normal operation mode. This change in modes will disconnect the wall power input and create a closed circuit from the generator through the PCC to the load power resistor.

The load resistor must be chosen carefully to ensure we find the right balance between drawing enough current from the generator and not generating too much torque on the shaft. If the load resistance is small, the generator will be able to output more power (see equation 2). This increase in power generation comes at a cost though as the smaller the load resistance, the more current we will draw from the generator. When more current flows through the winding around the generator, a larger induced magnetic field will be generated therefore adding unwanted torque to the output shaft thus decreasing the RPMs of the shaft resulting in a lower output voltage from the generator. To find the correct load resistance, we will test multiple resistors ranging from 1 - 10000Ω and record the power output and current out of the DC-DC converter to find the optimal balance between shaft torque and power generation.

Yaw

Design Requirements

According to the Rules and Regulations of the CWC, one must be able to manually adjust the nacelle into the direction of the wind during installation. This simple test of the yaw system suggests that it does not need to be an overly robust component of the turbine. Therefore, the nacelle must be light and the bearing for the yaw will have relatively low amounts of friction. As the bearing must sit below the nacelle and above the tower, there must be an additional connector piece between the bearing and the nacelle and a connection between the bearing and the tower.

Preliminary Design

The yaw system must be designed to provide easy rotation of the nacelle such that the drive shaft housed in the nacelle always remains parallel to the wind. This ensures that the system remains stable and the optimal amount of wind energy can be transferred into electrical power generation. The yaw system can be either passive or active, however, there are costs and benefits associated with each. The passive yaw system requires zero electrical power input but does not have a large degree of operational control that is present in an active yaw system. The team eventually decided on the passive yaw system due to its easy installation as well as its independent motion such that the controls team can prioritize blade pitching and emergency stop.

During the initial design phases of the passive yaw system, careful consideration was given to the bearing that would be located between the tower and the nacelle. This component must be lightweight and provide easy rotation of the nacelle. Initially, the team looked into the pillow block bearing chosen by the previous team, but it was eventually decided that this would be too heavy for the system. Eventually, a turntable bearing was chosen as this component is lightweight and easily rotates under about 10 pounds of loading (approximate weight of the nacelle).

Whilst the nacelle team considered the traditional yawing system of a bearing situated above the tower, the foundation and tower team began looking for a more unique approach. Instead of solely rotating the nacelle, another option would be to alter the tower. Instead of a standard cylinder, the team considered altering the tower such that it would have an airfoil along the length of the tower. The wind would then contact the tower and this would rotate the entire tower plus the nacelle. While this would allow for easy rotation and eliminate the need for an added bearing, there were concerns about the stability of the rotation. Specifically, the main question was whether the tower would cease rotation once the nacelle was aligned with the wind.

While the two options for the passive yaw system had equal merit, the team had to reanalyze the competition requirements to come to a decision.

Final Design

After re-evaluating the Rules and Regulations for the competition, the team noted the caveat that the turbine must be able to rotate manually during installation. Given this information, it was decided that to reduce the complexity of the project, the yaw system would consist of the turntable bearing on top of a cylindrical tower. However, as this turntable bearing has a large uncertainty of functionality, the bearing must be easily removed and replaced in case there is a sudden increase in friction within the part. So, the nacelle will be attached to a connector piece that is attached to the bearing with #6-32 fasteners.

Pitching

Design Requirements

The purpose of the pitching system is twofold. First, the system must angle each blade to optimize the revolutions per minute of the generator shaft. This will maximize the amount of wind energy converted to electrical power without rotating at speeds large enough to reach a no-load condition within the generator. Second, the pitching system is an integral part of the Emergency Stop such that pitching the blades to an angle of 0° will abruptly slow the rotation of the turbine shaft.

The pitching system shall adjust the angle of the blades to a precise location as directed by the pitching control system. The optimal pitch angle shall be determined by the pitch control algorithm and shall autonomously adjust the blade pitch angle. The pitching system shall maintain the optimal pitch angle as the wind speed increases and adds additional stress to the pitching system.

Preliminary Design

The initial design for the pitching system was inspired by the previous team's turbine. This system included a stepper motor that would rotate by a certain amount and push a bearing surrounding the shaft forward or backward based on the wind speed and desired rpm. After considering this design, the team decided to replace the stepper with a servo motor to increase available space within the nacelle. As this servo determines angular position, a linkage arm is necessary for the pitching system to convert this angular motion into linear translation. With this design change, the next step was to determine an appropriate mechanism that would change the angle of the blades as it moves linearly along the shaft. Initial consideration was given to manufacturing these components from scratch, but it was eventually decided that purchasing a pitching system would be the most efficient. This component is a helicopter pitching system (PN. 1653921) in which the base is driven along the shaft by the servo. After choosing the mechanical motion of the pitching system, the design of the control algorithm is the subsequent task. The controls govern the motion of the servo motor and the associated angle of the blade such that the optimal rpm value is reached. This algorithm must maximize the energy transfer to the generator without reaching an unstable point of a no-load condition.

Final Design

The following section provides an overview of the pitching algorithm that will stabilize the system throughout the operation and optimize the electrical energy produced by the generator.

Pitch Angle Control

Variable-speed wind turbines require a pitching angle controller to optimize power output. There are three goals the controller aims to achieve:

1. Optimizing the power output at low wind speeds.
2. Maintaining the rotor power at design limits at wind speeds that are higher than the rated wind speed of the wind turbine model.
3. Minimizing the fatigue loads of the turbine's mechanical components.

To achieve the three goals mentioned above, the pitch angle should vary to find an optimum tip speed ratio, λ, that corresponds to the optimum power coefficient the wind turbine can achieve at a given wind speed. By testing the wind turbine and recording the power coefficient at various pitch angles at a specified wind speed, the optimum pitch angle could be referenced for any wind speed later on for maximum power output. However, this control strategy is not effective enough as the effective wind speed cannot be measured accurately.

The control algorithm requires the following models that will describe the relations affecting the pitch angle:

The wind power is proportional to the cube of the wind turbine speed and is expressed as follows:

$$P = \frac{1}{2}\rho A v^3$$

Where ρis the air density, $A$is the area swept by the blades, and $v$is the wind speed. According to Betz's law, the wind turbine can only extract a fraction of the wind power (maximum of 59%). This fraction is described by the power coefficient, C_p, of the wind turbine which is a function of the blade pitch angle and the tip speed ratio. Therefore, the mechanical power achieved by the wind turbine is expressed as:

$$P_m = \frac{1}{2}\rho A v^3 C_p\,(\beta,\ \lambda)$$

Where C_p is the power coefficient of the wind turbine, β is the blade pitch angle, and λ is the tip speed ratio.

C_p varies with the wind speed and is dependent on the rotational speed of the rotor and the pitch angle of the turbine. The goal will be to change the pitch angle to increase the power coefficient of the wind turbine. This corresponds to an increase in the electric power achieved by the wind turbine model. C_p is calculated as follows:

$$C_p = 0.5176\left(\frac{116}{\lambda_i} - 0.4\beta - 5\right) e^{\frac{-21}{\lambda_i}} + 0.0068\,\lambda$$

Where:

$$\frac{1}{\lambda_i} = \frac{1}{\lambda+0.08\beta} - \frac{0.035}{\beta^3+1}$$

The tip speed ratio, λ, is calculated as:

$$\lambda = \frac{\omega_r R}{v}$$

Where ω_r is the turbine rotor speed, and R is the radius of the wind turbine blade, and v is the wind speed.

The equations above are implemented in MATLAB/Simulink to see the response of the system and the change in the C_p and P_m values given inputs of wind speed (v), rotor speed (ω_r), and the blade radius(R). Furthermore, a feedback control loop is implemented to control and change the pitch angle of the blade for two schemes:

1) The actual wind speed is below the rated wind speed of our turbine model.
2) The actual wind speed is above the rated wind speed of our turbine model.

Case I: Wind speed is below the wind turbine's rated wind speed:

Below rated wind speed, the power output is optimized using quadratic control law. Here, the optimization is achieved by using a torque control scheme in which the yield controller's (PI) gain is as follows [8]:

$$K_p = \frac{2\beta}{\omega_{r(ref)} - \omega_r},\ and\ K_i = 0$$

For the feedback control loop, the servo motor is modeled as a first-order system with a transfer function of:

$$\frac{1}{\tau_\beta s + 1}$$

Where τ_β is the response time of the servo motor to achieve the requested new pitch angle. Therefore, from Fig.7, the PI Pitch controller is modeled as follows:

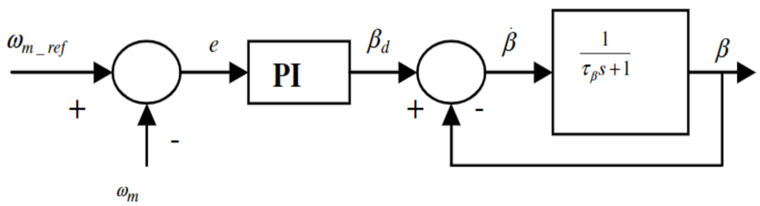


Figure 7. PI Pitch controller.

Case II: Wind speed is above the wind turbine's rated wind speed:

The PI controller's gain values are changed to maintain the rotor power below design limits. This is achieved by a scheduling algorithm that takes the wind speed threshold and decides which gains values to use. The controller gains K_p and K_i are calculated through the simulation from the following relation to be [8]:

$$\frac{\beta(s)}{E(s)} = \left(K_p + \frac{K_i}{s}\right)\left(\frac{1}{s+2}\right)$$

The feedback control loop is put along the models discussed above to yield our pitching angle control algorithm as shown in Fig.8, the control algorithm also involves an emergency stop that will pitch the angle to 90 degrees in case an emergency stop button is pressed. Eventually, the control algorithm Simulink model is connected to a microcontroller (Arduino) for taking the input of wind speed, and the rotor speed. Where it will also in return give an analog signal with the new pitch angle requested to the microcontroller.

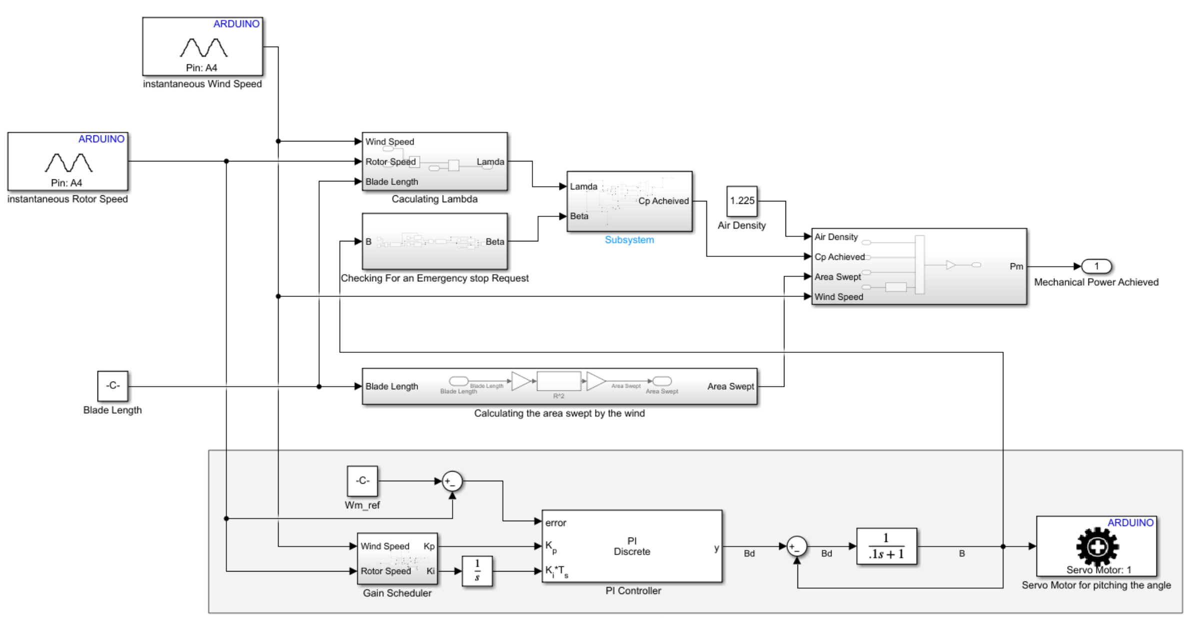


Figure 8. Simulink model for the pitching control algorithm.

Wind Speed Calculation

The flow velocity shall be determined by measuring the pressure differential around the tower. We will use the SEN0343 pressure sensor along with a microcontroller to measure the pressure differential. The pressure will be measured by drilling holes in the tower at points S and T shown in figure 9. Due to the flow characteristics around a cylinder, the flow velocity at point T will be twice the upstream velocity.[11] Therefore the flow velocity can be calculated using equation 9. To compensate for unexpected variance in turbulence and the sensor, the system will be calibrated in the wind tunnel with known wind speed.

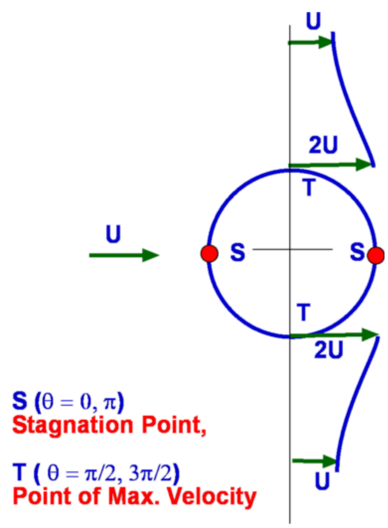


Figure 9. Stagnation Points for Flow about a Circular Cylinder [11]

$$U = \frac{1}{2}\sqrt{2 * (p_S - p_T)/\rho} \qquad [9]$$

Blades

Design Requirements

The turbine blade design needs to efficiently turn the wind energy into mechanical energy to power the generator. The blades must efficiently supply power at wind speeds ranging from 5-22 m/s. The entire nacelle head including the blades in their final position must be contained within the required volume to comply with the competition rules. In addition, the blades must be designed to be compatible with the pitching system. To meet both the requirements, the blade may only be 8.35 in. long, which does not include the adapter piece that interacts with the pitching system.

Table 2. Design specifications for the blades.

ID	Specification (Requirements)	Verification and Validation
BLD-1	The blades shall be optimized to produce the maximum power for a range of 6-13m/s wind speeds.	Inspection and testing
BLD-2	The blades must not break when subjected to the highest wind speeds or the highest RPMs	Inspection and testing at generated RPMs using a motor as well as static force testing

Preliminary Design

The first step in designing the blades was to pick an airfoil that would generate the most lift in various wind speeds while also reducing drag which would increase the stress in the blade. There are a large number of available airfoils which have associated data at different Reynolds numbers. To narrow down the selection we focused on blades commonly used in turbine design. The three types of blades that are the most common in the literature are the NACA series, S series, and DU series. To choose between these three, we narrowed our search to look for airfoils that excelled at wind speeds that would generate the most points during the competition. NACA airfoils are the series that had the most amount of publicly available data making them a solid starting point for our first design iteration. Once the airfoils had been narrowed to NACA series airfoils, an analysis was run in Q-Blade, a software that models wind turbine blade designs using the Blade Element Momentum Theory (BEM). This helped to narrow down to a shortlist of airfoils that are most likely to be effective in our application. Three airfoils that were shown in a research paper to be efficient at low wind speeds were the NACA 4412, 23012, and 0009 (Hong). Putting these three airfoils in Q-Blade and running an analysis on the relationship between the lift coefficient and drag coefficient, it can be seen that the NACA 4412 and 23012 airfoil were the most optimal airfoil choice.

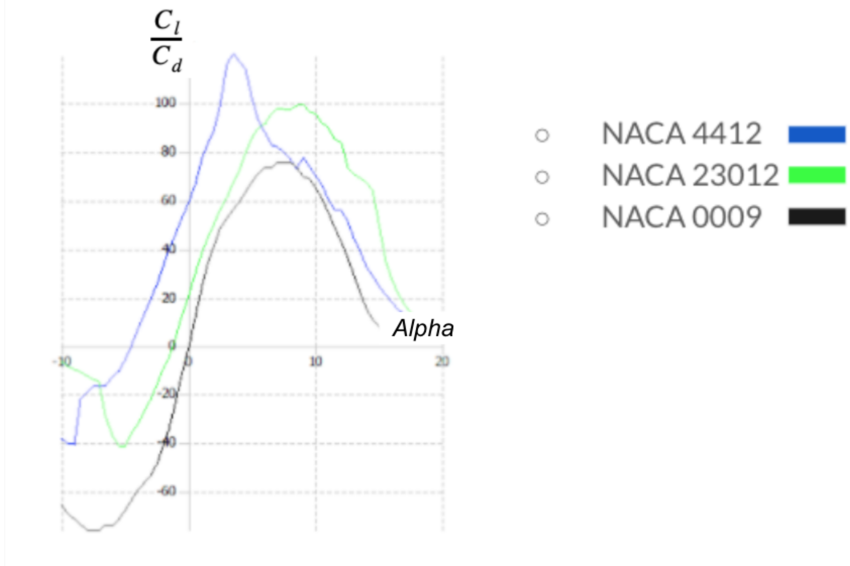


Figure 10. Lift coefficient divided by drag coefficient vs angles of attack (alpha) for 3 NACA series airfoils.

Figure 10 shows that the NACA 4412 airfoil has the desired characteristics at the wind speeds expected. This primarily is a high potential ratio between the coefficient of lift and coefficient of drag. However, the lift drops off at larger angles of attack. Furthermore, this airfoil also has a thicker overall profile making it structurally beneficial to be placed near the root of the blade. The NACA 23012 airfoil shows strong lift at larger angles of attack where the 4412 drops off so it would be beneficial to add to our blade profile near the tip. Since the NACA 0009 airfoil shows less lift than the other two at any angle of attack we removed this airfoil from our design considerations.

The tip speed ratio (TSR) calculation of the blade is significant because it allows for optimization for a targeted wind speed using a dimensionless quantity. This allows us to calculate the ideal blade measurements of twist and chord diameter. This subsequently allows for more control of the power curve. The turbine will only generate power and be effective for a range of wind speeds which will also create a range of tip speed ratios. By choosing the optimal tip speed ratio for the targeted wind speed, the turbine will generate the most power at the chosen wind speed and the surrounding wind speeds. The chosen wind speed is 7 m/s in hopes of strategically gaining points for the competition for both the power curve performance task as well as the durability task. There will be a test of the wind speeds from 6-22 m/s for the durability task, but the majority of the time will be spent on wind speeds between 6-13 m/s. For the power curve performance task, wind speeds from 5-11 m/s will be tested and the highest scaling factor for the awarded points is for power generation at wind speeds of 6 & 7 m/s. The optimal range for the power curve to cover would be from 5-13 m/s. Therefore, considering that it would be beneficial to have the turbine generate power for the greatest power curve including wind speeds of 5-13 m/s, the final optimized wind speed was chosen to be 7 m/s. The formulas for the TSR at 7 m/s are shown in **the centripetal force appendix.** From these calculations, it was determined that any TSR above 4 would cause the motor to move faster than 2,000 rpm, which would break the motor and many other components. From this, a TSR of 4 or less will be used for the following calculations.

Once the airfoils were chosen and the TSR range calculated, the twist and chord of the blade needed to be chosen and analyzed. The Schmitz Theory and formulas were utilized which relied on several factors that would help to specify a twist and chord that would be optimal. The formulas used are shown in Equation 3.1-3.3.

$$\varphi = \frac{2}{3}\tan^{-1}\left(\frac{1}{\lambda_r}\right) \quad \text{(Equation 3.1)}$$

$$c = \frac{8\pi r}{BC_l}(1 - cos\varphi)$$

(Equation 3.2)

$$\theta_P = \varphi - \alpha$$

(Equation 3.3)

To find the angle of the relative wind to the chord line of the blade, (φ), the formula above shows the relationship with the local tip speed ratio(λ_r). The chord (C),is reliant on the number of blades(B),the radius from the center of each airfoil being analyzed (r), as well as the lift coefficient (C_l). The twist of each section (θ_P),is calculated with values of the difference between the relative wind angle to the initial angle of attack (α). From these formulas, a spreadsheet was made to get a basic understanding of how each of these values would affect the shape of the blade. Simultaneously, a program was written to iterate through each of the design possibilities and used the spreadsheet as a way to check the reliability of the code. A shortened version of the values from the code is shown in table 3.

Table 3. Values of chord length and twist angles for airfoils at different radii for airfoil NACA4412.

r (in)	**Chord (in)**	**Twist (◦)**
0.492126	1.060602	36.2195
1.326334	1.559162	22.52935
2.160542	1.445205	13.34996
2.994751	1.236749	7.410792
3.828959	1.052311	3.432128
4.663167	0.905561	0.636666

5.497375	0.790387	-1.41486
6.331584	0.699111	-2.97625
7.165792	0.625628	-4.20064
8	0.565498	-5.18462

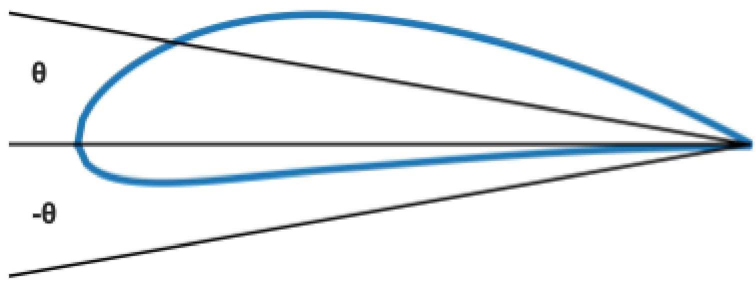


Figure 11. An image of the twist of each airfoil through the length of a blade on NACA 4412 airfoil.

Final Design

The final design of the airfoil will be determined by empirical data after testing. To characterize the many variables related to the blade, we must perform tests in the specific conditions our turbine will experience. Based on this data and the design outlined above, we will finalize our blade to perform best under the required conditions.

Nacelle

Design Requirements

The nacelle is not directly associated with gaining any points however it is the primary supportive structure of the turbine. This means that almost all the parts that can gain points are located on the nacelle making it one of the most critical parts of the turbine. This means that the design requirements are primarily to ensure the assumptions made by the other subteams are valid such as a rigid body attachment for the generator and yaw subteams, alignment of the central shaft for the generator subteam, and a small cross-sectional area assumed by the blade team.

Table 4. Nacelle Specifications.

ID	Specification (Requirements)	Verification and Validation
NAC-1	The nacelle may contain turbine electronics if desired	Inspection
NAC-2	The nacelle shall have the smallest possible cross-sectional area given the components that must be stored on it	Inspection

NAC-3	The Nacelle and blades must fit in a 45 cm x 45 cm box	Inspection and measurement

Preliminary Design

The nacelle was created to prioritize low mass. This was important due to the update that this year would be offshore and would have to rest on a base that, with too much mass, could become unstable. Another motivator for this decision was to overall improve the previous years' turbine where both their bearing and nacelle weights combined came out to 3 lbs 5 oz. The team felt that this weight could easily be improved upon and decided to search for a much thinner and smaller nacelle and a much lighter bearing. The team chose a 0.25'' material because this would be sufficient to hold the components while being thick enough to allow items to be fastened to the sides (using #4-40 fasteners). To find a size for the nacelle, the team made an outer boundary condition suggestion for the size of the components resting on the nacelle. The team subsequently created a visual aid to understand the general size and location of each component on the nacelle in figure 12 shown below.

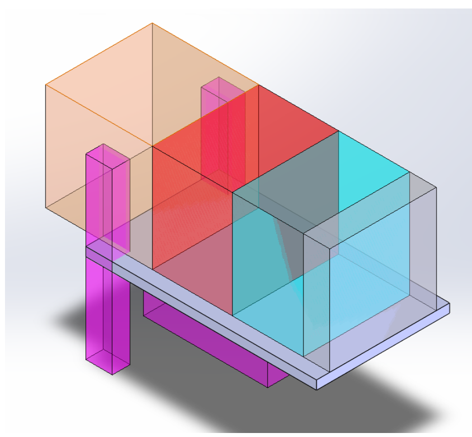


Figure 12. Colored representation of component spacing on the Nacelle. Purple represents the yaw system, yellow represents the generator, red represents the driveshaft, blue and grey represent the pitching mechanism.

Final Design

Next, the team needed to decide on the bearing that would attach to the tower to allow pitching. The previous year, the turbine used a cast iron pillow block weighing more than 2 lbs and was 5'' x 5'' which restricted how small the nacelle could be made and was the baseline bearing the team hoped to improve upon. The three goals for the bearing would be that it can support the weight of the nacelle while rotating freely, is easy for the manufacturer to install or attach, and is smaller in size than the previous year to reduce the unused space on the outskirts of the nacelle. The team researched many bearings such as large ball bearings or roller bearings.

These would require press-fitting into the nacelle and while considering the ease of manufacturing, the team decided against them. The turntable bearing weighs 0.32 oz. and is easily attached to the nacelle with four fasteners and could support a weight of 200 lbs, which the nacelle should be vastly below.

Foundation/Tower

Design Requirements

The foundation and tower have numerous design requirements that must be accounted for in the design. This comprehensive list can be found in table 5 below:

Table 5. A comprehensive list of the foundation and tower design requirements set by the Collegiate Wind Competition.

ID	Specification (Requirements)	Verification and Validation
FND-1	The anchoring system shall fit within the 25x25x15 cm^3 foundation envelope specified by R&R's inside the offshore simulation tank.	Inspection
FND-2	The anchoring system shall interface with the competition-provided Transition Piece (stub) according to the R&R's.	Inspection and testing
FND-3	The turbine tower shall interface with the Transition Piece (stub) according to the R&R's.	Inspection and testing
FND-4	The transition piece shall not move more than 6mm during the second portion of the durability test with 8 - 22m/s wind speeds.	Testing
FND-5	The foundation shall be installed within 15 minutes.	Testing/practice
FND-6	No body parts shall touch the water during the installation of the foundation.	Practice
FND-7	The post that connects the foundation to the transition piece above the water must be within 3 cm of the top of the container that the water and sand are contained in.	Inspection and Testing
FND-8	Any structure below the transition piece must be made of ferrous material.	Material Selection
TWR-1	The tower shall support the nacelle under any loading conditions experienced during competition testing.	Testing at measured/calculated loads for the nacelle and full testing
TWR-2	The tower shall have an inner diameter of 1 in to allow for electrical wiring to pass through it.	Inspection (measurement)

TWR-3	The tower shall mechanically interface with the transition piece and the nacelle.	Inspection

In this year's Collegiate Wind Competition the offshore foundation is a completely new aspect and therefore has a lot of unique and new requirements compared to the rest of the turbine prototype. This means that the specifications outlined in Table 5 needed to be studied extensively and implemented accordingly in the preliminary and final design.

Preliminary Design

Foundation

The preliminary design of the foundation is a simple upside-down bucket (Appendix A.1), with an attachment post. This design is similar to an industry-standard suction bucket but without the suction. This design provides the most structural stability while minimizing the mass of the foundation and eliminates the need for the design of a suction apparatus. The curved outside edge of the bucket provides a large surface area to interact with the sand when it undergoes forces from the wind and gravitational forces from the turbine. Other designs including a monopile and tripod configuration were considered but during prototyping, in a mock tank filled with sand, the bucket design far outperformed the other two. Placing each prototype in the sand container and exerting a force with our hand it was clear that the structural stability of the monopile and tripod would not be sufficient to stand up to much force, and the bucket design was more stable, leading us to continue with that design almost immediately.

One of the other critical design considerations is height adjustability for the top of the foundation due to the 3 cm constraint at the top of the container (FND-7). Multiple designs have been considered in regards to how this would be accomplished, shown in table 6 below.

Table 6. Design choices for height adjustability of foundation.

Design	**Description**
Spring Pin (Preliminary Idea)	An inner and outer tube where the inner tube has several holes along the shaft that allow a spring pin to be inserted. The spring pin will be mounted on the stationary outer tube and can be pulled back to allow the inner tube to slide.
Pin Slot (Preliminary Idea)	An inner and outer tube with the inner tube having two threaded holes on opposite sides and the outer tube having two slots on opposite sides. The inner tube can be adjusted to the desired height and will be held in place with two bolts that enter through

	the slot and screw into the two threaded holes. The frictional force between the heads of the bolts and the outer tube holds the nacelle and tower in place.
Clamping Shaft Collar (Appendix A, 3)	An inner and outer tube where the outer tube is stationary and the inner slides up and down. The outer post has 2 slots of removed material at the top to allow the clamping shaft collar to tighten the tube and hold the inner tube in place, similar to the seat height adjustability mechanism on a bike.
Threaded (Appendix A, 2)	The post is broken up into two parts, a top, and a bottom, and a set screw is threaded into the top section. The bottom section has a plate that has a threaded hole in the center where the set screw from the top can interface with and move up and down as the top post section is twisted. A clamping ring offset from the bottom plate using standoffs encircles the top post section and will clamp the post when the desired height is reached.

Tower

The preliminary design of the tower is based on one-inch inner diameter tubing with an adapter at each end to attach to the Nacelle and the Transition Piece. Three materials have been considered for the tower: Aluminum 6061, Fiberglass, and Carbon Fiber. The inner diameter was chosen based on the quantity of electrical wiring, while also minimizing the cross-sectional area of the tower to reduce drag when the turbine is in operation.

Final Design

Foundation

Based on manufacturing considerations and prototyping we pursued the general idea of the bucket foundation. The final design incorporates supports to better distribute load and create more structural rigidity. Based on our trade study for the height adjustment we chose the threaded design. This design allows for the best adjustability, low mass construction, and the most stability for the rest of the turbine.

The design of the bucket will be made of two separate sheets of low-carbon steel that will be cut into a circular and rectangular shape and welded together to form the upside-down bucket. Thin ribs will be used to add stability to the bucket for stiffness when undergoing the moment from the force of the wind. To choose the final design for adjusting the height of the foundation post a trade study was conducted which weighted desired traits and compared different designs (Appendix C, 1). The chosen design is the threaded post which had strong results in the ease of adjustability during competition and was lighter than the next closest design, the clamped tube. This height adjustment design will rest on top of the bucket, with welded ribs going to the edge of the top of the bucket and up to the beginning of the threaded adjustment subassembly components. Providing stability to the post and the top of the bucket, the ribs along with the lower section of the upside-down bucket will counteract the moment from the drag force of the wind in the sand.

Tower

By performing an analysis on the three material options for the tower, we have selected carbon fiber based on our trade study that weighed the results of the analysis, price, and weight (Appendix C, 2). This is mainly due to its strength-to-weight ratio providing a high yield and ultimate strength at a very low mass. Mass is of vital importance as it will be measured at the competition and assigned a score relative to other teams' masses. This rules out aluminum as it is significantly heavier than fiberglass and carbon fiber. Furthermore, the rigidity of carbon fiber is desirable as we don't want the tower to flex under loads.

Testing Plan

Blade Test

The blades will be tested in the CU Aerospace Engineering Department's wind tunnel. This tunnel has a test chamber with dimensions of 12"x12"x12". While the complete rotor with all three blades will not fit in this section, there is sufficient space to test an individual blade on the nacelle. The blades will be fixed to a sting balance system borrowed from the testing facility. The properties of the blades will be tested using the recommended procedure provided by the testing facility. The sting balance will transfer the forces from the blade to the strain gauges which will output a voltage that can be analyzed. From this data, we will be able to extract the lift and drag forces exerted on various aerofoil designs. Each aerofoil will be tested at a range of wind speeds between 5 and 22 m/s and at pitch angles between 0 and 90 degrees.

Pitot Tube Calibration

Again using the wind tunnel we will calibrate the pitot tube velocity measurement system. For this test, we will place the velocity measurement device in the wind tunnel and vary the wind speed. By comparing our measured wind velocity to the velocity measured by the wind tunnel we will be able to calibrate our sensor and validate that it is functioning accurately.

Drag Test

To determine the typical and maximum drag forces exerted on the system we will subject the turbine rotor and nacelle to a 5 m/s wind speed and measure the force exerted on it by the wind. The force will be measured by load cells placed under the nacelle. From this force, we will be able to calculate the turbine's drag coefficient using the drag equation (Appendix B.1). From there we will be able to estimate the drag force on the system at different speeds including the maximum wind speed of 22 m/s. The drag force will be important for testing the foundation as this directly correlates to the bending moment of the body.

This test requires the full rotor so we will have to use an alternate testing method than the wind tunnel. To generate 5 m/s wind speeds, we will use the industrial fan in the CU Idea Forge. This fan moves 4,000 cubic feet of air per minute (cfm). We will constrict this flow through a 20"x20" cardboard tube sealed to the end of the fan to maintain flow velocity. Using the continuity equation[2] we can determine the maximum wind speed through this tunnel to be ~7.3 m/s assuming minimal pressure drop in the short tube. To measure the actual wind speed in the tube we will use a Vernier anemometer. The possibility exists to design a larger wind tunnel that would be able to generate wind speeds >20m/s.

Pitch Control Test

For this test, we will validate that the autonomous pitching system is functioning correctly. We will vary the wind speed between 5 m/s and 22 m/s and record the rpm of the rotor and pitch angle of the blades. From this data, we will be able to tune the blade pitch control algorithm so that the optimum tip speed ratio is maintained.

Ideally, this test will be performed over the full wind speed range. To do this we will attach the rotor to the top of a vehicle and drive at various speeds. The turbine base plate will be bolted to plywood that will be securely strapped to the vehicle.

Full System Test

This is a full test of the system. It is similar to the pitch control test except that the motor and electrical system will be connected. In this test, we will validate that the turbine functions autonomously and generates high-quality power. From this test, we will also be able to determine the turbine's power curve.

Foundation Pull Test

To test the stability of our foundation, we must perform a pull test. The foundation will be placed in a tub filled with sand and water that is at the same height specifications as the competition. A tape measure is placed next to the tower on the sand to record the displacement of the tower as it is pulled. The pulling mechanism will involve a string that is attached to the top of the tower which will be horizontally guided over a stationary pulley. At the other end of the string, weights will be added in increasingly heavy weights to pull the tower with increasing force. This mimics the force of the wind that the turbine will experience and thus will provide us with accurate test results to iterate our design in terms of the weight of the foundation. The critical weight of the foundation that will be needed to withstand wind speeds from 5-22 m/s will ultimately be determined in wind tunnel tests.

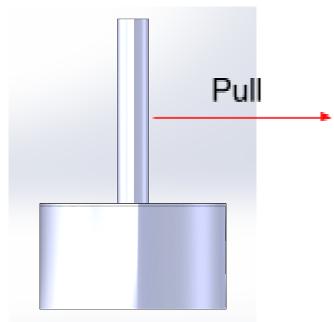


Figure 13. Pull Test Diagram.

Foundation Installation Rehearsal

For the competition, we will only have 15 minutes to fully install our turbine in the testing tank along with making all the electrical connections that are needed for full functionality. Thus, it is vital that the foundation itself is easy to install and can be done relatively quickly to allow sufficient time for other portions of the wind turbine to be installed. The best way to ensure this is to simulate the conditions of competition and practice installing the foundation ahead of time. We will have a large tub filled with the exact height of sand and water to mimic the conditions that will be found in the competition and practice several times to ensure timely installation.

Generator Test

To test the operation of the generator we will mechanically connect the generator to a motor. We will then turn the motor on at varying speeds to simulate the rotation of the wind turbine rotor. We will measure the power output of the generator at various speeds. This will provide us with a power curve for the generator and verify that the generator is functioning properly and producing power.

Power Quality Test

It is also important that the power generated is stable. For a given generator rpm, the power output should be smooth with minimal noise. To test this, we will record the output signal from the rectifier. We will observe the noise in this signal and use filters to minimize the noise.

Emergency Stop Test

The turbine should be able to shut down to 10% of its average rpm within 10 seconds. To test this we will simulate operation by rotating the rotor with a motor. Once the rotor is spinning at an operational frequency, we will actuate the emergency stop system. By pitching the blades to increase drag and shorting the generator leads, the turbine should quickly slow. We will monitor the rpm of the rotor with the tachometer to observe how quickly the shutdown procedure was completed.

Project Financial Status

Bill of Materials

Table 7 shows the preliminary bill of materials for the wind turbine design. It is broken up into the five critical subsystems of the wind turbine. Despite allotting $200 towards project prototyping, the team has not needed to use this money because of all the resources and materials left to us from last year's project team. These proved to be sufficient in helping us understand how all the different components of a turbine worked and have assisted us in being able to design, analyze, and select our custom and purchased components for the final design. Furthermore, much of the prototyping could be done via various software such as QBlade and SolidWorks as well as handwritten equations which have no associated costs.

Table 7. Initial bill of materials.

Foundation							
Notes	Description	Vendor	Part No.	Quantity	Price Per Unit	Shipping (Est)	Total (Est)
At least14x29" sheet of metal	Low-Carbon Steel Sheet, 24" x 48" x 0.0300"	McMaster	6544K19	1	43.02		43.02
At least14x29" sheet of metal	Low-Carbon Steel Sheet, 24" x 48" x 0.0600"	McMaster	6544K21	1	93.65		93.65
3 ft. length	Easy-to-Weld 4130 Alloy Steel Round Tube, 0.035" Wall Thickness, 1-1/2" OD	McMaster	89955K389	1	36.06		36.06
For height adjustment	Clamping Shaft Collar 1 5/8" shaft diameter	McMaster	6435K68	1	46.43		46.43
For custom machined	Stainless steel sheet 6"x12" x 0.75"	McMaster	8983K982	1	105.46		105.46

threaded adjustment parts							
Threaded adjustment standoff	Female Threaded Round Standoff	McMaster	91125A473	1	4.11		4.11
Fasteners for threaded adjustment	Variety of SHCS	McMaster	Various	1	15		15
						Subtotal	343.73

Electrical							
Notes	**Description**	**Vendor**	**Part No.**	**Quantity**	**Price Per Unit**	**Shipping (Est)**	**Total (Est)**
	100 W Brushless Motor, Diameter = 40mm, high torque density with low cogging torque	Maxon	488607	2	302.5		605
Used to measure air speed at the turbine	Differential pressure sensor	Mouser	426-SEN0343	1	39.9		39.9
						Subtotal	644.9

Nacelle							
Notes	**Description**	**Vendor**	**Part No.**	**Quantity**	**Price Per Unit**	**Shipping (Est)**	**Total (Est)**
Rotary Shaft that connects the pitching mechanism to the generator	1/4" Stainless Steel Shaft (9" long)	McMaster	1327K118	2	8.82		17.64
Bearing for rotational shaft	Mounted Open Needle-Roller Bearing for 1/4" Shaft Diameter	McMaster	2031N11	1	25.75		25.75
Screws for turntable, bearing mount, L-bracket mount	1/4" 6-32 Button head screws	McMaster	92949A145	1	3.53		3.53

Screws for mounting needle rollar bearing	5/16" 6-32 Button head screws	McMaster	92949A148	1	3.56		3.56
Turn table bearing used to yaw the system	Turn Table Bearing (3 in x 3 in)	McMaster	6031K16	3	2.84		8.52
Coupling unit to the generator	Shaft coupling	McMaster	61005k2	2	28.02		56.04
Generator L-Bracket Connection	L-channel	McMaster	8982K123-8982K232	1	6.06		6.06
Washers to center 6-32 screws from bearing to the because nacelle	Washers for bearing attachment	McMaster	92141A008	1	1.2		1.2
Screws for mounting the bearing attachment to the nacelle	1/2" 6-32 Button head screws	McMaster	91251A148	1	10.17		10.17
						Subtotal	126.41

Blades

Notes	**Description**	**Vendor**	**Part No.**	**Quantity**	**Price Per Unit**	**Shipping (Est)**	**Total (Est)**
	PLA filament for test blades	Hatchbox		1	28.99		28.99
	SLA Resin for final blades	Amazon		2	32.99		65.98
						Subtotal	94.97

Pitching

Notes	**Description**	**Vendor**	**Part No.**	**Quantity**	**Price Per Unit**	**Shipping (Est)**	**Total (Est)**
	Pitching Mechanism	Banggood	1653921	1	98.99	3.79	102.78

	Subtotal	102.78

Table 8. A summary of all expenses to date.

Expense Summary	
Foundation/Tower	$ 343.73
Electrical	$ 644.90
Nacelle	$ 132.47
Pitching	$ 102.78
Blades	$ 94.97
Contingency 7%	$ 91.90
Total Project Estimate	**$ 1,411.17**

Table 9. A summary of all funding sources. Currently awaiting the final decision of the EEF board about our proposal

Funding Sources:	
CU Provided	$2,000
EEF Grant from Last Year	$1,000
EEF Grant 2021 (waiting for approval)	$3,000
Total Project Funding	**$3,000**

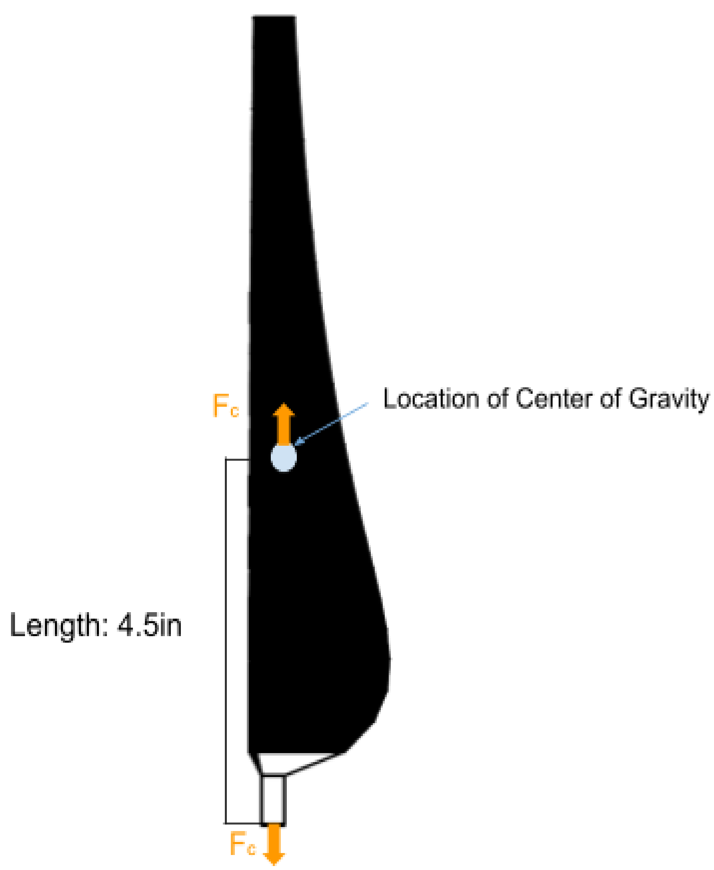
Financial Status Assessment

Currently, the project is well within budget due to the remaining funds from last year's project team's Engineering Excellence Fund Grant (EEF) which totaled $1000 (Table 3). The team also submitted their own EEF Grant proposal in October and is awaiting the board's final decision which will come December 1st. If approved, the team can receive up to an additional $3000 in funds.

It is important to note that despite still having about $1600 (Table 2), we have not taken into account the costs associated with manufacturing and shipping. Manufacturing costs will be determined during the spring semester and shipping costs will be determined when ordering all materials and components. Because a majority of the materials and components are being bought from McMaster-Carr, we plan to order everything at once to reduce the shipping costs as much as possible. With this in mind, the team is not concerned about going over budget and we have enough funds to cover the costs of any design revisions and unforeseen costs that may arise later in the project.

Analysis of Components

Centripetal Force on Blades



We assume that the blade is a rectangular blade with equal density throughout making the center of mass half the overall length assumed to be 9 inches. This agrees with the assumed worst-case condition because the overall blade will be a consistent density with a tapering chord diameter indicating that the center of mass will be located closer to the base of the blade than indicated. Using the centripetal force equation shown below we can see that as the radius of the mass increases the force will increase.

$$F_c = m * r * \omega^2$$

For the properties of the blade, we will use last year's blades as a mass estimate which was found to be 1.1 oz per blade. To pad this we assumed a total mass for our blades to be around 2 oz. For the blade RPM, we will use the specifications found for the rotor hub to be at 2000 RPM maximum.

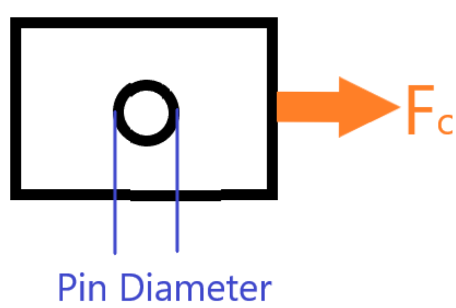
m = ⅛ lb

r = 4.5/12 in

$\omega =$ 2000 RPM = 209.3 rad/s

Using these values we can calculate the force on the base of the blade is approximately bounded by $F_c \approx 2053.4\ lbf$. This can be used as an upper estimate for the forces the pins are subjected to and the hub

Centripetal Force on Rotor Pins



The pin area is characterized as a through-hole and therefore the effective area is twice the cross-sectional area of the pins. Due to the fact we are buying the rotor hub, we have an assumed pin diameter of ⅛ inch and a max shear comparable to many screws available on McMaster Carr of 84,000 psi.

$$A_c = 2 * A_{pin} = \frac{1}{2} * \pi * D^2 = \frac{1}{2} * \pi * (1/8)^2 = \frac{\pi}{128} in^2$$

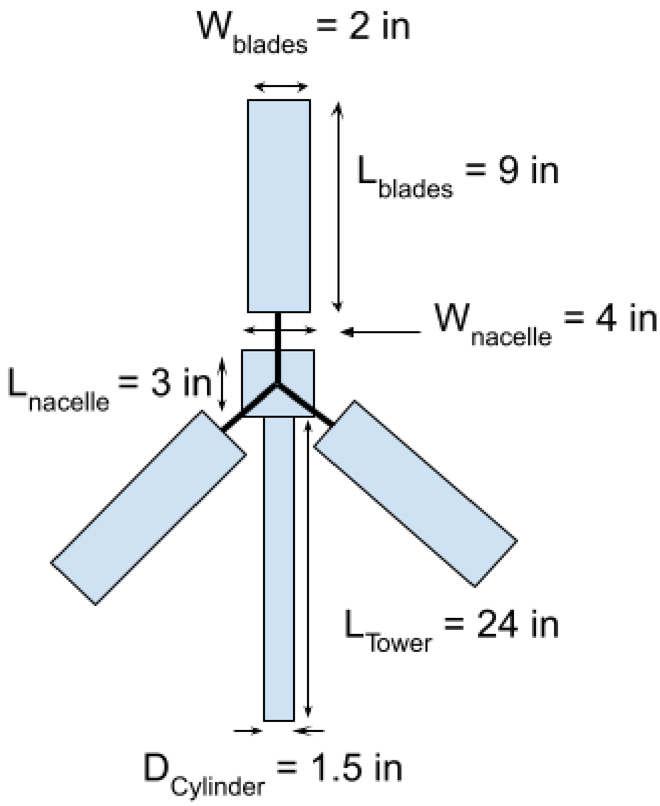
Shear force

$$\tau = F_c/A_c = 2053.4/(\pi/128) = 83663psi$$

Safety Factor

$$N = \tau_{max}/\tau = 84000/83663 \approx 1$$

Although a safety factor of 1 is unacceptable this assumes that the blades are rough twice the weight expected from last year's tests and that we are running at a speed of 2000 RPM which is 500 RPM faster than last year's tests these factors along with a radially closer center of gravity are expected to lower the centripetal forces. We also believe this is acceptable due to the fact that the blade is the most likely component to fail due to the high forces subjected to it and the material it will be made of, likely a 3D printed plastic.



To calculate the maximum forces exerted on the base of the tower a square geometry was assumed for the nacelle, blades, and tower. This will yield the highest forces due to the high drag coefficients on a flat plate. The dimensions are the absolute maximums for the blades and tower where the nacelle is a reasonable approximation of the largest area that would be required to house the generator and will likely be smaller making the total force smaller.

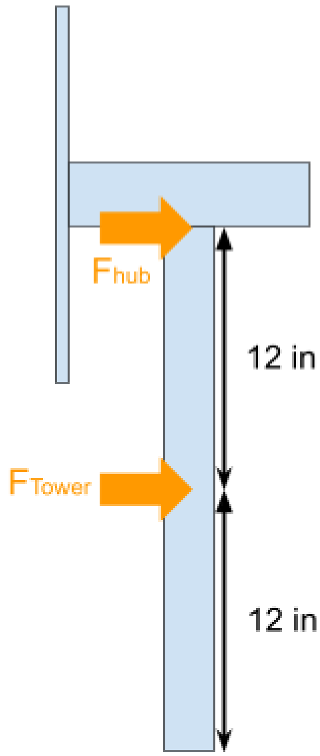
$$A_{nacelle} = 3 * 4 = 12\, in^2 A_{blade} = 2 * 9 = 18\, in^2$$

$$A_{hub} = A_{nacelle} + 3 * A_{blade} = 12 + 3 * 18 = 66\, in^2$$

$$A_{tower} = 1.5 * 24 = 36\, in^2$$

We can break the forces into two components when analyzing the forces on the base of the tower.

Max Force on the Base of Tower



To calculate the force exerted on the tower base we have to convert the worst-case situation of max airspeed pressure to the force

$$P = 0.00256 * 50^2 \approx 6.4\, psf = 6.4/144\, lb/in^2$$

From the pressure, we can calculate the forces by multiplying the pressure by area.

$$F_{hub} = PA_{hub} = 6.4 * 66/144 = 2.9\bar{3} lbf$$

$$F_{tower} = P * A_{tower} = 1.6\, lbf$$

The reacting force from the base of the tower is the total force

$$F_{base} = F_{hub} + F_{tower} = 4.53\, lbf$$

The moment can be calculated using the distances from the base for the two forces

$$M_{base} = 1.6 * 12 + 2.93 * 24 = 89.52\, lbin$$

Torque on Shaft

Yield Strength: σ_y = 70 ksi

Calculating Polar Inertia

$$J = \frac{\pi d^4}{32} = 3.83E - 4\, in^4$$

$\tau_{all} = \frac{1}{2}(70) \ = \ 35\,ksi$

$T_{max} \ = \frac{\tau J}{c} \ = \ \frac{(35E3)*(3.83E-4)}{(0.25/2)} = 107.38 lbin$

Designing for a TSR value of 4

$\lambda = \frac{\omega R}{u_\infty}$

Wind Speed

$u_\infty \ = \ 866.14\, in/s$

Largest Designed Length of Blades

$R \ = \ 8.86\, in$

Calculated Angular Rotational

$\omega \ = \ 391.03\, rad/s$

To determine the rotational moment on the shaft from the blades, the following equations provided by the Technical University of Denmark Department of Wind Energy will be used once a blade design has been finalized through empirical testing. [10]

$V_{rel}^{\ 2} = (1 - a)^2 V_0 \ + \ (1 + a')^2 \omega^2 r^2$

$L \ = \ 0.5\rho c C_L V_{rel}^{\ 2}$(Lift Force on the Blade)

$D \ = \ 0.5\rho c C_D$(Drag Force on the Blade)

$P_n \ = L\cos\phi \ + \ D\sin\phi$ (Normal Force used in moment calculation)

$P_t \ = \ L\sin\phi \ - \ D\cos\phi$ (Tangent Force)

Sources

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[7] de Oliveira, A. R. et al. "Development of a Wind Turbine Blade with Dedicated Profiles by Schmitz's Optimum Dimensioning Systematization". Proceedings of the 10th International Conference on Rotor Dynamics -- IFToMM. Red Katia Lucchesi Cavalca en Hans Ingo Weber. Cham: Springer International Publishing, 2019. 544–559. Print.
[8]Abdulhamed Hwas, Reza Katebi, Wind Turbine Control Using PI Pitch Angle Controller, https://www.sciencedirect.com/science/article/pii/S147466701631031X.
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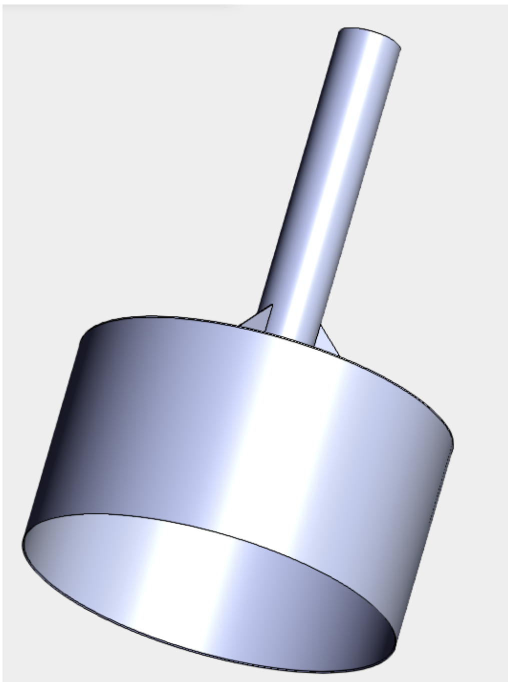
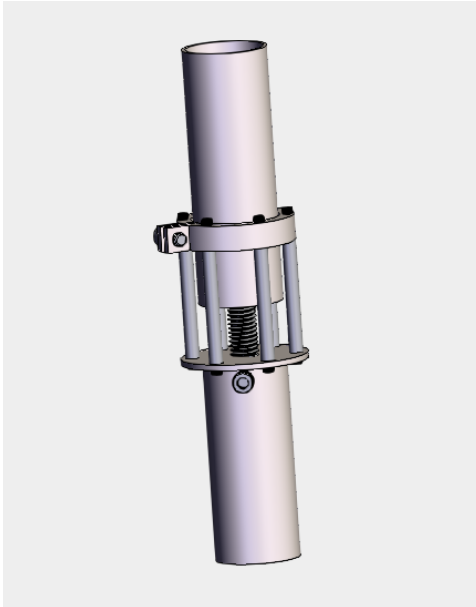
[10] Bredmose, Henrik. "Flow and Forces around a Wind Turbine Blade." *YouTube*, DTU Wind Energy, 24 Feb. 2016, https://www.youtube.com/watch?v=wOkm1oqpRhk.
[11]http://www-mdp.eng.cam.ac.uk/web/library/enginfo/aerothermal_dvd_only/aero/fprops/poten/node37.html

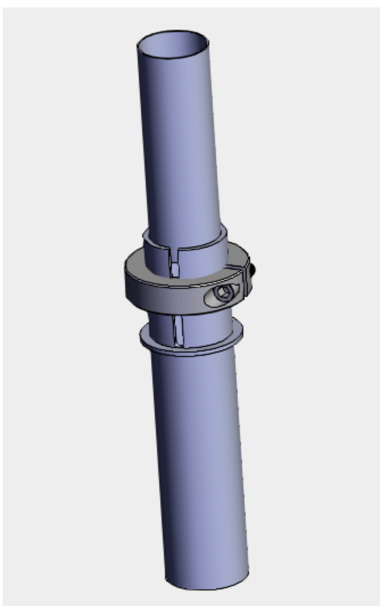
Appendix A (CAD and Drawings)

1. 

Rendering of the upside-down bucket assembly

2. 

Rendering of the threaded post design of the height adjustment feature on the foundation.

3. 

Rendering of the clamped design of the height adjustment feature on the foundation.

Appendix B (Equations)

1 Drag Equation

$D = (1/2)C_d \rho V^2 A$

2 Continuity Equation

$A_1 V_1 = A_2 V_2$

Appendix C (Trade Studies)

1. Post Design Trade Study

			Double Slotted Screw		Cylinder Clamp		Threaded		Spring Pin	
Identification	**Metric [unit]**	**Weight**	**Score**	**Weighted Score**	**Score**	**Weighted Score**	**Score**	**Weighted Score**	**Score**	**Weighted Score**
Adjustability	time (minutes)	0.3	2	0.6	2.5	0.75	3	0.9	2	0.6
Manufacturability	time (weeks)	0.05	2	0.1	2	0.1	1	0.05	2	0.1
Height precision	cm	0.1	2	0.2	2	0.2	3	0.3	2	0.2
Mass	kg	0.05	3	0.15	3	0.15	3	0.15	2	0.1
Movement potential	mm	0.2	1	0.2	3	0.6	2	0.4	1	0.2
Safety factor	safety factor	0.3	2	0.6	3	0.9	3	0.9	2	0.6
TOTAL				1.85		2.7		2.7		1.8
Adjustability score: 1-5min = 3, 5-10min = 2, 10-15min = 1, 15-20+min = 0. Adjustability during competition.										
Manufacturability score: 1-2wks = 3, 2-3wks = 2, 3-4wks = 1, 4+wks = 0										
Height precision score: +-0.5cm = 3, +-1cm = 2, +-1.5cm = 1, +-2cm+ = 0										
Mass score: Relative to other designs										
Movement potential: 0-2mm = 3, 2-4mm = 2, 4-6mm = 1, 6+mm = 0. Requirement based on R&Rs										
Safety factor score: safety factor based on Solidworks analysis										

2. Tower Trade Study

			Aluminum		Fiberglass		Carbon Fiber	
Identification	**Metric [unit]**	**Weight**	**Score**	**Weighted Score**	**Score**	**Weighted Score**	**Score**	**Weighted Score**
Mass	kg	0.2	1	0.2	3	0.6	3	0.6
Safety factor	Safety factor	0.4	3	1.2	3	1.2	3	1.2
Manufacturab	time	0.2	2	0.4	3	0.6	3	0.6

ility	(weeks)							
Cost	$	0.2	3	0.6	1	0.2	1	0.2
Conductivity	Yes/No		0	0	1	0	0	0
TOTAL				2.4		2.6		2.6
Mass score: 0-1kg = 3, 1-2kg = 2, 2-3kg = 1, 3+kg = 0								
Safety factor score: Safety factor based on Solidworks analysis								
Manufacturability score: 1-2wks = 3, 2-3wks = 2, 3-4wks = 1, 4+wks = 0								
Cost score: $0-50 = 3, $50-100 = 2, $100-150 = 1, $150+ = 0								
Conductivity score: Yes = 0, No = 1								