Wind Turbine Blade Final Report

Lab 411 Group 4

Lola Anderson (lra57), Kate Ng (kln46), Jeff Yang (py64), Luke Brust (lmb388)

Executive Summary

Our group was tasked with designing a blade optimized for maximum power production at a specific wind speed.

To accomplish this, we began by making assumptions about the fluid flow in our environment. We expected that the wind experienced by the turbine would not be constant, varying in speed according to a Weibull distribution ranging from 0 m/s to 8 m/s. To determine the optimal wind speed for design, we needed to identify the speed that was both most likely to occur and capable of producing the highest theoretical power for a given turbine size. Our approach involved multiplying the theoretical power output by the probability of each wind speed occurring. This analysis indicated that the optimal design wind speed was 5.4 m/s.

Next, we selected an airfoil for our blade design. We agreed that our application required an airfoil with a high lift-to-drag ratio (Cl/Cd). Based on previous research on small-scale wind turbine blades, we identified the SG6043 airfoil, which demonstrated an impressive Cl/Cd ratio of 65 [1]. Then to determine the optimal rotational rate, we used the MATLAB script provided in class for power calculations. Inputting the SG6043 airfoil properties and testing various rotation rates led us to select an optimal rotation rate of 1250 RPM.

We then performed finite element analysis (FEA) on the initial blade geometry to evaluate yield stresses and deflection magnitudes at the blade tip. Two loading scenarios were analyzed: one under optimal design conditions and another under maximum possible operating conditions. Our analysis revealed that the initial blade design experienced significant deflection at the tip, even under optimal conditions. To address this, we scaled the chord length using a function that increased chord more significantly toward the tip than at the base, resulting in a fatter, more rectangular blade shape.

For wind tunnel testing, we selected a low wind speed and applied a braking torque to achieve a rotation rate as close to 2000 RPM as possible. We then incrementally increased the braking torque until the blade stalled. Once stall occurred, we incremented the wind speed in steps of approximately 1 m/s, repeating this process as many times as the allotted testing time permitted.

Finally, we compared our results to our design conditions by plotting power output versus RPM and power output versus wind speed. While our blade performed optimally under the designed conditions, some technical limitations encountered during the lab testing prevented us from conclusively verifying this outcome. These limitations are explored in greater detail in this report.

Design Objectives and Goals

The overall goal was to design a wind turbine that was optimized for power production in a given environment. The conditions in this hypothetical environment were given via a probability density function of the incoming wind velocity, described by a Weibull distribution (equation below). It was up to each group to decide what criteria to use to determine what constituted the maximum power produced in this environment.

$$p(U) = \left(\frac{k}{c}\right)\left(\frac{U}{c}\right)^{k-1} \exp\left[-\left(\frac{U}{c}\right)^{k}\right]$$

Weibull probability distribution function describing wind velocity. c = 5, k = 5

In addition to optimizing the blade design for the given environment, we had to meet several key manufacturing and testing requirements. The blade's maximum radius was limited to 6 inches to fit the available wind tunnel setup, and the root of the blade had to match the geometry of the provided hub for secure mounting. Testing conditions were constrained by the limits of lab equipment, ensuring the wind turbine did not exceed 2500 RPM or 40 oz-in of torque, as set by the B2 Magnetic Particle Brake specifications [2].

We also established self-imposed design requirements. The maximum equivalent stress under peak loading conditions could not exceed the yield stress of the PLA material used to 3D print the blade. Additionally, we set a maximum deflection limit of 2 mm under theoretical optimal conditions to prevent deviations in experimental performance caused by excessive blade deflection. These constraints ensured the blade remained functional, safe, and aligned with theoretical predictions during testing.

Operating Conditions and Assumptions

Operating Conditions

We needed to decide on operating conditions at which to optimize our blade design. To do this, we visualized the expected power at each velocity by plotting the theoretical power output of a turbine in each wind velocity (equation below) multiplied by the probability that the wind would reach that velocity. This yielded the expected power curve shown below which peaked at 5.4 m/s which is the velocity that we decided to optimize for.

$$Power = \frac{1}{2}\rho A v^{3} 4a(1 - a)^{2}$$

Theoretical power output of a wind turbine for a given wind velocity.

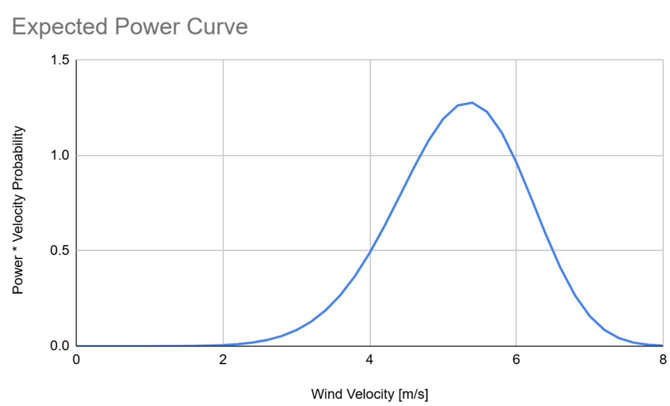

Fig 1. Graph to visualize the probability of producing power at each wind velocity.

Assumptions

Our initial blade design assumed a 1D analysis, ignoring the blade's wake by setting the angular induction factor (a') to zero and the axial induction factor (a) to 1/3. The axial induction factor represents how much air slows as it approaches the turbine, while the angular induction factor indicates the amount of rotational velocity imparted to the wake by the spinning wind turbine. While accurate for a 1D model, in reality, a and a' vary along the blade's radius. To improve accuracy, we used a blade element momentum (BEM) code, which combines conservation of energy, momentum, and airfoil theory to iteratively refine a and a' until both models agree. This approach comes from the model of a 2D wind turbine blade and incorporates the effects of angular velocity imparted by the spinning of the wind turbine.

The BEM ignores 3D effects such as root and tip vortices. These vortices appear when high-pressure air from below the blade circulates to the low-pressure region above, increasing drag and reducing lift. To account for this, we tapered the blade and made the chord larger at the root than the tip. Additionally, when calculating power coefficients from experimental data, we only consider the axial induction factor, neglecting angular effects.

Design Process

Theoretical Design

We started our design process by doing some research on airfoils that performed well when used for small scale wind turbine blades. We were looking for a blade with a high lift to drag ratio (C_L/C_D) in order to maximize the useful force that we could produce with the turbine. We found that the airfoil SG6043 had the highest lift to drag ratio of all of the airfoils that we looked at and was accredited by a research paper to perform well for small scale, horizontal axis wind turbines [1].

As outlined above in *Assumptions*, we began our geometry design process by using 1D assumptions, using an axial induction factor of ⅓ and an angular induction factor of 0. This allowed us to use the Betz Blade equations for chord and angle of attack as a function of radius which we derived during Lab 5/6 . These equations are as follows:

$$\theta(r) = tan^{-1}(\frac{U_1}{r\omega}) - \alpha_{max\ cl/cd}$$

$$c(r) = \frac{{U_1}^2}{{U_2}^2} * \frac{4\pi ar(1-a)}{3(C_l cos(\theta) - C_d sin(\theta))}$$

These equations were initially used where the axial induction factor was assumed to be ideal and equal to ⅓. These equations do not account for 2D effects of angular velocity in the wake. The angular velocity of the wake not being accounted for means that the effective angle of attack is actually lower than what we intend it to be. To account for this we increased our pitch angle by a small percentage throughout the radius of the blade. We also iteratively found that we ended up extracting the highest power at a higher rotation rate than what we designed for. We were able to iterate through different wind speeds and rotation rates and optimize our blade with the goal of achieving the highest power coefficient at 5.4 meters per second. This led us to our first theoretical design. However, due to choosing a very thin airfoil, our first design did not pass our structural requirements. We then scaled the chord by a factor which increased linearly with the radius. The equation for this increase is given below in *Design Verification*.

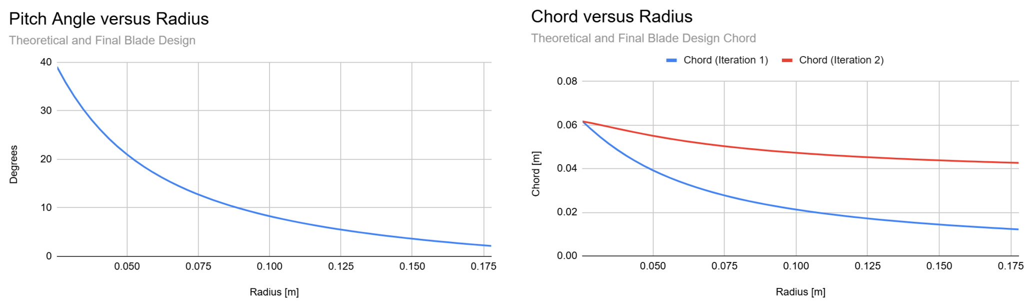

Fig 2. Pitch Angle (left) and Chord (right) as a function of turbine radius for our initial and final geometry iterations.

Design Verification

We imposed two structural requirements on our design to validate that the blade would be able to withstand the expected loads without breaking or deviating too far from the expected performance.

Requirement 1 - Optimal Loading Conditions: At the theoretical optimal operating conditions of the wind turbine (U_1 = 5.4 m/s and Ω = 1250 RPM), no part of the blade should deflect more than 2 mm.
Requirement 2 - Worst Case Loading Conditions: At the theoretically highest loading conditions (U_1 = 7.745 m/s and Ω = 1000 RPM), the equivalent stress within the blade structure should not exceed the flexural yield strength of PLA - 77.4 MPa.

Requirement 1 was determined to prevent the performance of the blade at optimal conditions from suffering due to deflection. The loads were determined based on the conditions that we optimized the blade for. Requirement 2 was to prevent the structure of our blade from failing under any realistic conditions. The worst case incoming wind velocity was determined to be 3 standard deviations above the mean velocity in our predicted operating environment (this covered 99.7% of expected wind velocities). The conditions that cause the highest forces are dependent on two factors, the incoming wind velocity incident on the airfoil, and the angle of attack of that incident air (which determines the C_L and C_D). To find where those two factors resulted in a maximum torque, we calculated the total torque produced by the blade in 7.745 m/s wind over a range of rotation rates. This revealed that our blade would produce the maximum amount of torque - 0.123 Nm - at a rotation rate of 1000 RPM.

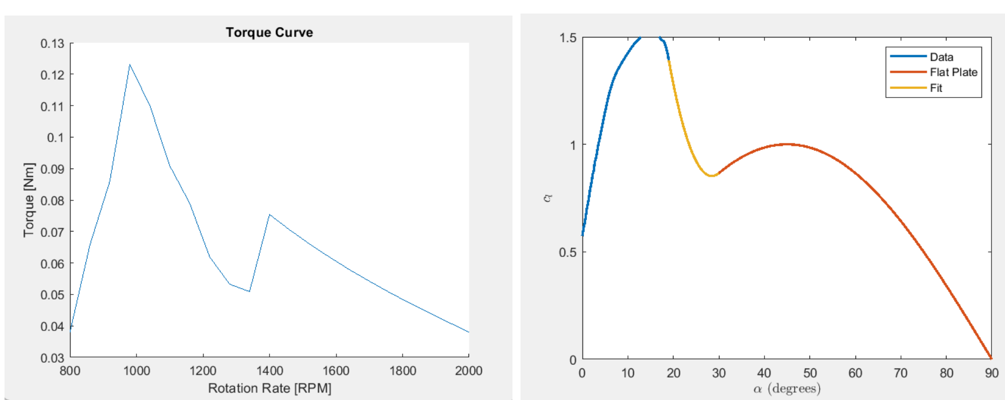

Fig 3. Torque as a function of rotation rate compared to C_L of the SG6043 airfoil as a function of α

At first, the torque curve appears to have a very strange shape, but upon closer inspection, it appears very similar in shape to the C_L curve of the SG 6043 airfoil that we used to design our blade. This shows that the torque on the turbine is a direct result of the angle of attack of the blade cross sections which changes with rotation rate.

We performed ANSYS Static Structural simulations to model the loads on our blade during testing. We modeled two cases: the optimal design case to verify requirement 1 and the worst case loads case to verify requirement 2. We modeled those conditions in MATLAB to obtain the expected axial and useful force in each scenario. The loads were applied to 10 increments along the blade using force loads and the connection of the dovetail joint to the hub was modeled as a compression only constraint. We measured the maximum deflection of the tip of the blade for the optimal conditions case and the maximum equivalent stress for the worst loads case to determine whether our design met the requirements.

PLA Material Properties [3]:

PLA Properties:		
Density	1.3	g/cm^2
Young's Modulus	2.35	GPa
Poisson's Ratio	0.3	
Ultimate Tensile Strength	57.3	MPa
Yield Strength	45.2	MPa
Flexural Yield Strength	77.4	MPa

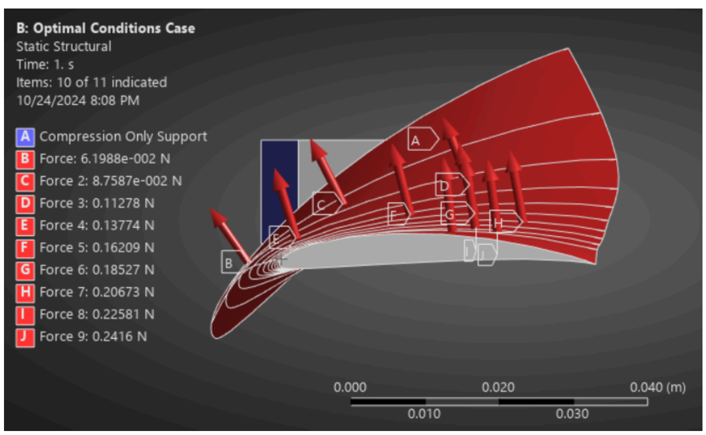

Fig 4. Boundary conditions applied to the blade in the Optimal Conditions Case

After running the Optimal Conditions Case on our theoretical blade design, it was clear that the deflection at the tip of the blade was too large, failing requirement 1. In order to decrease the deflection, we chose to increase the moment of inertia of the blade's cross section. Since the deflection was seen disproportionately close to the tip of the blade, we chose to scale the original chord using a scale factor that increased linearly along the radius of the blade.

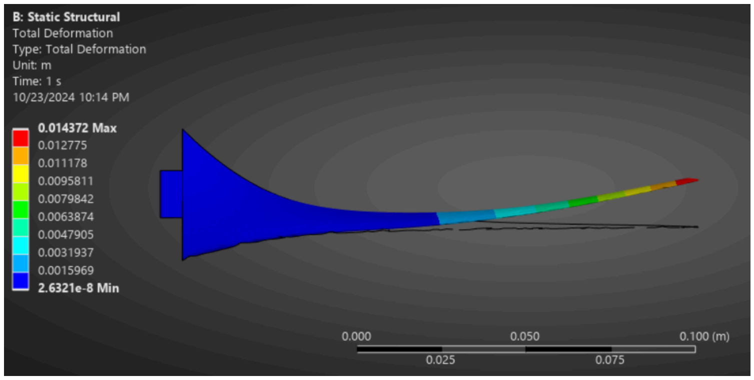

Fig 5. Deflection results of the original blade design. δ_{max} = 14.4 mm

$$C'(r) = C(r)[1 + \frac{r - r_{hub}}{r_{tip} - r_{hub}} \cdot b]$$

Function to scale the blade chord length

Using the new blade geometry, we then re-ran the ANSYS simulation and verified that the blade met both requirements. We found that the maximum deflection expected at the optimal operating conditions was 1.7 mm, and the structure had a margin to yield of 14.2 even in the worst loading case with a factor of safety of 1.5.

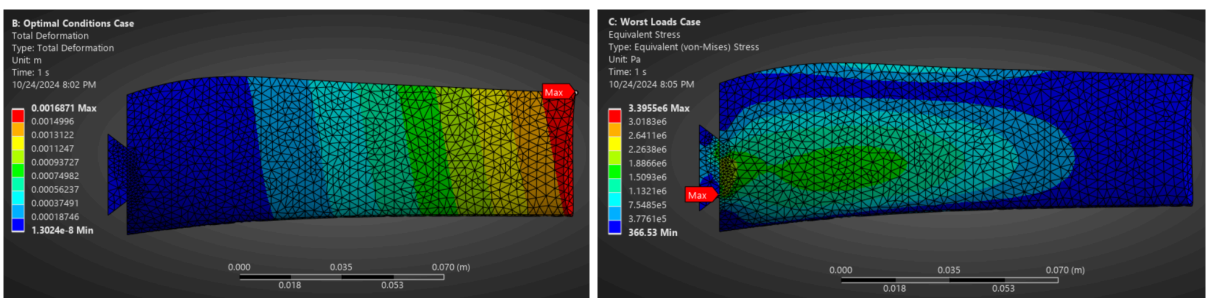

Fig 6. ANSYS results for (Left) deflection at optimal conditions and (Right) equivalent stress at worst-case conditions.

Testing

Our group was allocated 45 minutes to test our blades in Big Blue, aiming to evaluate blade performance while minimizing the risk of failure or damage to lab equipment. To ensure safety, we reviewed the torque brake specification sheet to confirm that expected power outputs were within the operating range of the equipment.

Our testing methodology involved varying wind speeds in the wind tunnel, both below and above the blade's optimal wind speed, and recording power output at discrete rotational rates. We started the machine at the lowest wind speed that could induce blade rotation, approximately 2.7 m/s. At this speed, we tapped the wind tunnel to overcome the static friction in the bearing and initiate rotation. Once the blade was spinning, we gradually increased the wind speed until the rotational rate reached 2000 RPM — a target selected to provide sufficient power data before stall, while staying below the 3000 RPM limit. We then incrementally increased the torque brake current, recording LabVIEW data at each 100 RPM decrease and saving it for further analysis.

This process was repeated for wind speeds of 3 m/s, 3.7 m/s, 4.2 m/s, and 5.4 m/s As measured by the pitot tube. We observed that higher wind speeds produced more data points before stall, consistent with the expectation that increased wind speeds supply more available power, enabling the blade to maintain rotation at lower RPMs.

These speeds measured by the pitot tube do not always correspond to the far field velocity. Since the experiment is taking place inside of a wind tunnel, the flow is slowed by the sides of the wind tunnel and is not the same as wind in an unconfined environment. The wind turbine blade itself also impacts the results of the pitot tube measurement due to reverse wake and the blockage effect of having a large object rotating in the flow. We also recorded the frequencies of the wind tunnel at each recorded wind speed and then used the calibration curve on the side of the wind tunnel to map these frequencies to a wind velocity. According to this mapping, we had been recording data at 4.4 m/s, 5.4 m/s, 5.9 m/s, and 7.4 m/s. We will present our results with both the pitot tube wind speeds and the calibration curve wind speeds separately and comment on each.

A key challenge arose during testing at 5.4 m/s (from pitot tube), when the torque brake mount began spinning within its threading, causing the blade to rotate and misalign with the fluid flow. This issue compromised the accuracy of our results and limited the range of wind speeds we could test within the allotted time.

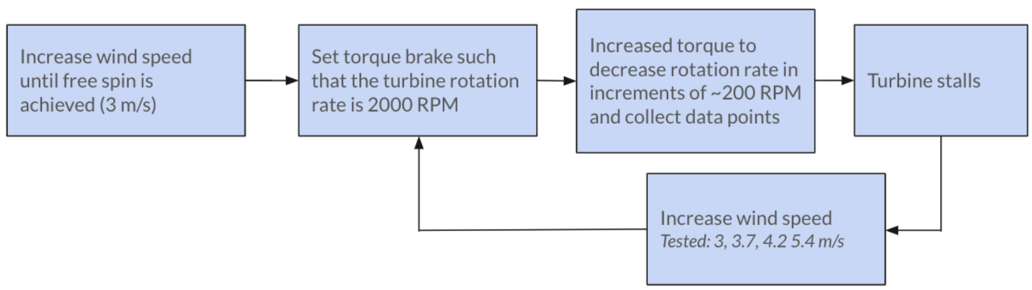

Fig 7. Flow chart outlining the steps for our testing procedure

Results

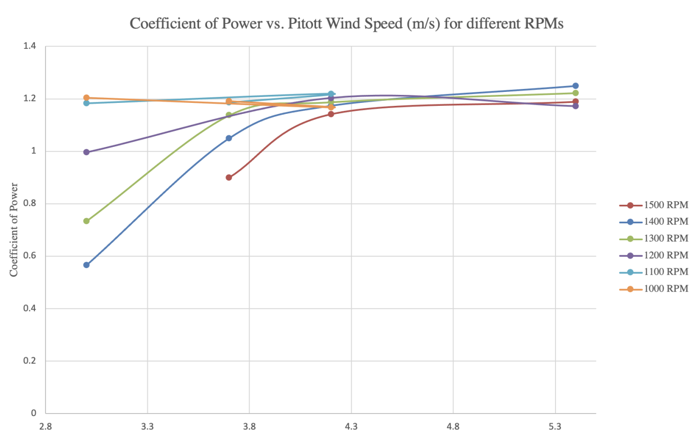

Fig 8. Coefficient of Power vs. Pitot Tube Wind Speed (m/s) for different RPMs

We observed that the RPM affects the coefficient of power more drastically for lower wind velocities, while the effects of RPM diminishes at higher RPMs. The main contributing factor becomes wind velocity more than anything else. This is expected because power is a function of wind velocity to the third power.

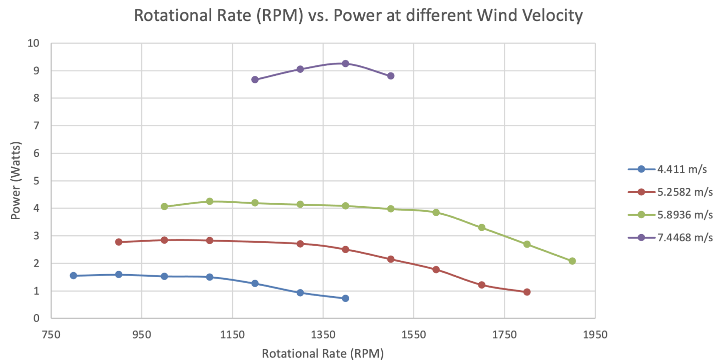

Fig 9. Rotational Rate (RPM) vs. Power (Watts) for different Wind Velocity

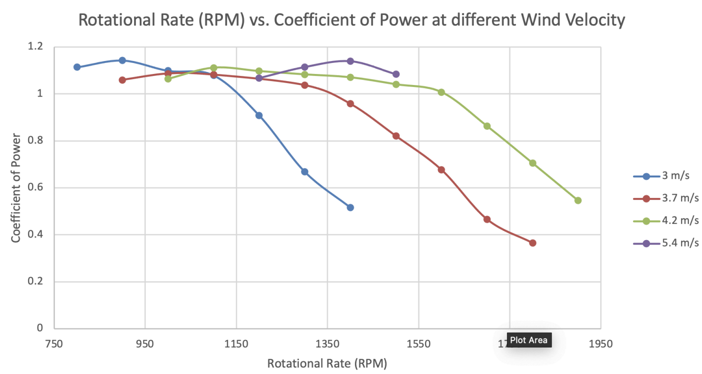

Fig 10. Rotational Rate (RPM) vs. Coefficient of Power for Wind Velocity from Pitot Tube

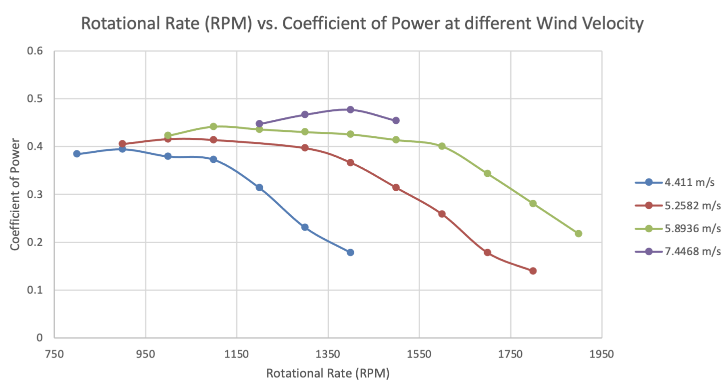

Fig 11. Rotational Rate (RPM) vs. Coefficient of Power for Wind Velocity from Wind Tunnel Calibration Curve

The maximum power achieved is 9.24 W at 1400 RPM and 5.4 m/s (Fig 9). Observing the rotational rate vs. power graph, the curves are relatively uniform, indicating that changes in RPM have a minimal impact on power until a certain threshold RPM is reached, beyond which power begins to decrease. After non-dimensionalizing the power using the pitot tube wind speeds (Fig 10), we found that the power coefficient is maximized to the same value of 1.15 across all measured wind speeds. This suggests that the design is not only optimized for a specific wind velocity but performs well across the Weibull distribution of wind speeds.

The power equation for a wind turbine, $P = 1/2\rho A U^3$, assumes the turbine operates in an open area with uniform wind velocity. In contrast, our wind turbine operates within a confined wind tunnel where the blade's swept area is significant relative to the cross-sectional area of the tunnel. This difference introduces blockage effects, as described in the previous section, which may have significantly impacted our experimental power measurements.

Notably, when using the pitot tube wind speeds we obtained power coefficients that were twice as large as the Betz limit of 0.593. Upon further review, we determined that the discrepancy is not due to a calculation error but rather is due to the assumption that the pitot tube's measure of velocity is sufficiently close to the far field velocity. As stated before, this assumption is incorrect and a more accurate way to find U1 is to use velocities found from the wind tunnel calibration curve. Fig 11 shows that the actual highest power coefficient of .476 is found at 1400 RPM at 7.4 m/s. This contrasts our result from Fig 10 and suggests that our blade performs more optimally at higher wind speeds than our target of 5.4 m/s.

Although using the velocities from the calibration curve is still not a perfect estimate of U1, it is important to note that when we used the calibration curve wind speeds, the power coefficients no longer exceeded the betz limit. This indicates that these wind speeds are the more accurate ones and we should consider the power coefficients obtained using these wind speeds more heavily than the pitot tube power coefficients.

Conclusion of Blade Design

Conclusion

To recap our process, we began by designing an initial geometry using 1D fluid dynamic analysis to optimize for a wind speed of 5.4 m/s and rotation rate of 1250. Then through iterative design and finite element analysis (FEA), we optimized the blade's performance while ensuring structural integrity. Our final design met key criteria for deflection and stress, confirming its ability to operate safely under expected loading conditions

Our wind turbine blade design achieved a maximum power output of 9.24 W at 1400 RPM and 7.4 m/s, showing large differences from our design target. This indicates that although we used matlab analysis and fluid equations, these equations can break down when placed in the confined box of the wind tunnel test section. Though wind tunnels are great for creating and studying laminar flows, the test object should always be much smaller than the cross section of the wind tunnel test section if we are attempting to measure how it performs in an open area. We chose to maximize our wind turbine blade length, but this meant that they almost touched the sides of the wind tunnel. Since the swept area of the turbine was almost equal to the area of the test area cross section, we ended up experiencing a reverse wake from the turbine and a blockage effect. If we were to repeat this design process and experiment again, we would choose to create a blade with a smaller radius, so that when it is placed in the wind tunnel we can find out more accurately how it would perform in an open area.

For future iterations, we recommend collecting data for wind speeds beyond 5.4 m/s to assess performance at velocities closer to the maximum of the Weibull distribution. Additionally, improving far-field velocity measurement techniques by stalling the turbine and recording velocity when the blade is stationary would yield more accurate power coefficient calculations. Given that power coefficient is inversely proportional to the cube of wind speed, even a small increase in measured velocity can lead to significant changes in power coefficient, underscoring the importance of accurate velocity measurement.

Conclusion of group dynamics:

Our group structured our work mostly around out of class meetings which proved to be very productive and helpful in our design process and performance analysis. We were able to communicate effectively over text message and met all deadlines well in advance. Lab experimentation went smoothly and there proved to be little to no group conflict.

Sources

[1] Noronha, Naveen Prakash & Krishna, Munishamaih. *Aerodynamic performance comparison of airfoils suggested for small horizontal axis wind turbines*. ScienceDirect. Retrieved from https://www.sciencedirect.com/science/article/pii/S2214785321004508

[2] Placid Industries, Inc. *magnetic particle BRAKE B2 1.2 to 40 oz.-in.* (Technical data sheet). Retrieved from https://canvas.cornell.edu/courses/66286/files/11712521?module_item_id=2861876

[3] MatWeb. *Overview of materials for Polylactic Acid (PLA) Biopolymer*. (Material data sheet) Retrieved from https://www.matweb.com/search/DataSheet.aspx?MatGUID=ab96a4c0655c4018a8785ac4031b9278