



Final Design Report of a 400W Portable Wind Turbine For Submission to the First NREL National Collegiate Wind Competition



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Acknowledgments

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1 Introduction

1.1 Purpose and Vision

The purpose of this report is to provide detailed engineering analysis and design of a 400W wind turbine and 10W prototype to compete in the Inaugural Collegiate Wind Competition hosted by the DOE and NREL. The vision of Jayhawk Windustries is to develop innovative solutions to the challenges of small-scale wind energy by cultivating a diverse and multi-disciplinary team of forward thinking, aspiring student engineers in the belief that we, as the next generation of engineers will, through education and application, develop into the technical leaders who will solve the greatest challenges of the future.

1.2 Design Team

The following figure introduces the entire company structure of Jayhawk Windustries.



Figure 1.1: Jayhawk Windustries Organization Chart





Team Member	Background	Responsibilities
Katie Constant, AE,	Coursework in Wind Turbine Design	Research and application of turbine in specific markets
Market Research Lead	Extensive research capabilities	Market issues research and presentation
Alejandra Escalera, AE,	 International student from Bolivia 	Review of all designed components
Chief Technical Officer	 Coursework in business concepts 	Communication facilitation with manufacturing team
		Technical advisor for design team
Andrew Lichter, ME,	Internship with Flint Hills Resources	Emergency shutdown system
Designer	Extensive coursework in mechanical design	 Incorporation of disc crake into gearbox
		Designed brake actuation components and layout
Julian McCafferty, AE,	Air Force Cadet Leadership Experience	Direction and vision of the company
Chief Executive Officer	Leadership Studies	Employee welfare and team moral
	 Coursework in Wind Turbine Design 	 Meeting company objectives and deadlines
	Composites and Manufacturing Experience	Final Design reviews and decision making
Evan Reznicek, ME,	Extensive coursework in mechanical design &	Lead team in designing power train
Mechanical Team Lead	alternative energy	Designed gear drive and power electronics
	Research experience in small wind energy technology	
James Sellers, AE,	Air Force Cadet Leadership Experience	Concept design using CAD
Testing Team Lead	• Extensive experience in CAD and graphic illustration	Prototype development
	 Coursework in Wind Turbine Design 	Product test and evaluation
	Interest in test and evaluation	Company graphics development
Alex Sizemore, AE,	Coursework in structural analysis and FEA	Business plan development
Designer		Blade Structure and Design
Emily Thompson, AE,	Air Force Cadet Leadership Experience	Prototype manufacturing
Chief Operating Officer	 Coursework in Wind Turbine Design 	Quality control and safety
Mary Pat Whittaker,	Multiple years experience in assistant positions	Travel arrangements
AE, Executive Assistant	• Lab technician for AE department, specifically running	Competition submission materials
	wind tunnels	Communication facilitation
		Aid to all departments in reaching overall goal
		Preliminary wind tunnel testing



1.3 **Design Objective**

This report details the design of the prototype wind turbine to be tested in the wind tunnel at the competition, and the market turbine in accordance with the Jayhawk Windustries Business Plan. The design objective of the prototype wind turbine is to provide maximum commonality with the market turbine, while meeting all performance requirements outlined in the 2014 Collegiate Wind Competition Rules¹. The objective of the market design is to provide a safe, high performance, durable, affordable, and portable turbine to power provisional project sites and electronic equipment of global Non-Governmental Organizations (NGOs) with the potential to expand to additional markets.

The following merit criteria were developed to determine the design approach, materials, and processes for the wind turbine with a respective weighting for its influence on the design.

Weighting System	Percentage		
Safety	25%		
Performance	20%		
Maintainability	20%		
Procurement Cost	15%		
Life Cycle Cost	10%		
Portability	10%		
TOTAL	100%		

Table 1.2: Market Merit Criteria

Safety considerations include labor safety, structural margins, and most importantly risk reduction of injury to the customer. Performance includes providing a competitive \$/Watt to the competition and power generation to meet the needs of the customer. Maintainability is defined as the turbine's repairability, ease of access, and simplicity of the design. Minimizing the procurement cost requires appropriate and efficient manufacturing, processing, and facilities. Minimizing life cycle cost entails providing durable and lasting components with simple and low cost parts replacement. Finally, portability affects parts count, weight, size, breakdown and assembly, and is considered as a necessary feature for NGOs and other customers with operating sites that move at least once per year.



This wind turbine is differentiated from the market by featuring a shrouded rotor for safety and added performance, a hub-less rotor for low parts count, and a collapsible design to maximize portability. This assemblage of characteristics is currently unavailable on the market.

2 Market Wind Turbine

Table 2.1. Warket Turbine Design Specifications					
Rated Power	Configuration	Rated Speed	Rotor Diameter	Speed Configuration	
400 W	Horizontal Axis	12 m/s	1 meter	Variable Speed	

The Market Turbine is a power regulated, geared, variable speed, 3-bladed horizontal axis upwind turbine (HAWT). Since this is a small turbine, a pitch regulation system would be too costly and complex, but a variable speed rotor is necessary to track optimal TSR for maximum energy capture of IEC Class III wind speeds. A 3-bladed rotor was chosen to minimize tower top oscillation and blade length for portability. An upwind HAWT configuration was chosen to provide the blades clean air and because it generally has greater efficiency over Vertical Axis Turbines which is critical for low wind speeds. The only disadvantage is the necessity to account for blade strike. The scale of the market turbine is constrained to maintain portability while providing adequate power to the customer.

2.1 Airfoil Selection

The market turbine airfoil is the SG6040 selected from the UIUC Airfoil Database². The SG6040 Airfoil has been wind tunnel tested and is selected because it does not exhibit laminar separation for lower Reynolds numbers of small wind turbines, and because it provides a larger thickness to chord ratio for improved structural margins.











2.2 Blade Design

The market turbine was designed using a programmable excel spreadsheet provided in Dr. Wetzel's design course in which chord and twist is iterated to maximize power output using designed and assumed parameters such as TSR, rotor diameter, and efficiencies. The geometry and stall regulated idealized power curve are given below assuming an electromechanical efficiency of 0.9. The blade design evolution is adopted from the prototype blades detailed in the prototype design section.

Table 2.2: Market Turbine Blade Geometry						
Z node (m)	Chord (m)	Twist (deg)				
0.075	0.070	11.00				
0.125	0.050	10.00				
0.175	0.040	8.00				
0.225	0.030	5.00				
0.275	0.025	3.00				
0.325	0.021	1.00				
0.375	0.020	0.00				
0.425	0.016	-1.00				
0.475	0.010	-2.00				
0.500	0.010	-3.00				



Figure 2.3: Market Turbine Power Output, Regulated at 400 W

2.3 Materials

In order to determine the optimal materials and processes for the manufacturing of the market turbine, Jayhawk Windustries contracted a series of undergraduate consultant tiger teams representing distinct companies but comprising of students from the main team to provide cost benefit analyses and structural engineering work for 300, 3,000, and 30,000 turbines per year production rates. Jayhawk Windustries provided the merit criteria from Table 1.2, reviewed each report and presentation, and selected the following recommendations from *Sirius Consulting*. The estimated cost per blade includes materials, tooling, labor, overhead, machine procurement, and operational costs.

Table 2.3: Sirius Consulting	g Recommendation to Ja	yhawk Windustries ⁴
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	Blade (3 per Turbine)
Manufacturing Process	Injection Molding, In-house
Material	RTP 2289 HM Plastic (50% Carbon Fiber)
Estimated Cost/Part	\$13.77 (900/year), \$10.43 (9,000/year), \$10.30 (90,000/year)



2.4 <u>Loads</u>

The following loading simulations of IEC 61400-1 DLCs were performed using FAST v7.02 and AeroDyn v13.00 and TurbSim analysis software. This simulation software is publically available on the NREL website. The TurbSim software creates a stochastic, full-field, turbulent-wind model using a statistical model and three-component wind speed vectors⁵.

- DLC 1.1 (NTM) for $V_{avg} = V_{out}$
- DLC 1.1 (NTM) for $V_{avg} = V_{rated}$

The FAST/AeroDyn software models the dynamic response of conventional horizontal-axis wind turbines⁶. This software was used to generate blade loads and analyze the turbine's dynamic response to different wind case scenarios. Table 2.4 summarizes the simulated



Figure 2.4: TurbSim Flow-Field Model Example⁵

results and analyzed structural margins of safety. A DLC 6.1 of extreme winds at 37.5 m/s is not analyzed because the turbine is not designed to withstand those loads. The user will be instructed to disassemble the turbine before subjecting it to extreme winds above 20 m/s until further maximum loads testing is completed. RTP 2289 HM Plastic Material Properties are found in Reference 7.

		Blade Root	Tauran Tau	Tower- Top	Design Load, M-y/I (Pa)		Failure Load (Pa)		Margin of Safety	
Tria	I	Bending Moment (N-m)	Thrust (N)	Bending Moment (N-m)	Edgewise	Flapwise	Edgewise	Flapwise	Edgewise	Flapwise
DLC 1.1	Max	18.9	95.04	9.82	9.92E+05	2.72E+06	4.60E+06	1.01E+07	3.64	2.8
Vavg= Vout	Min	8.00	43.9	3.41						
DLC 1.1	Max	20.0	101.0	10.5	1.005.00	2.075.07	4.005.00	1.045.07	2.20	25
vavg= Vrated	Min	13.2	72.2	7.51	1.06E+06	2.87E+07	4.00E+06	1.04E+07	3.30	2.5

Table 2.4 only includes the bending moments seen at the root of the blade because they are known to be the critical load. The flapwise and edgewise results were used to calculate the margin of safety for each condition based upon the geometry and material properties for the market turbine; each blade is predicted to safely meet material design allowables as seen by the positive margins of safety.



2.5 Drivetrain Components

The gear drive is necessary to increase generator shaft speed in order to increase generator voltage output. To produce 400 W, the generator must produce at least 17 V RMS, meaning it must spin

at approximately 34000 rpm. With a rotor speed of 1770 rpm, this requires at least a 19.2:1 gear ratio. To achieve this, the 24-28 mm Ammo Motor Great Planes Planetary Gear Drive was combined with a custom designed parallel shaft gear drive. The planetary gear drive has a gear ratio of 4.3:1 and the parallel shaft gear drive has a





gear ratio of 4.5:1. Thus, their combined gear ratio (generator speed/rotor speed) is 19.35:1. A 0.375 inch rotor shaft was selected, along with two flange mounted bearings. The rear bearing is a radial ball bearing that has an extended collar to allow axial constraint of the shaft and take axial thrust load. The spur gear is constrained to the rotor shaft with a locking set screw, and the parallel shaft gear drive pinion is constrained to the planetary gear drive shaft with a second locking set screw. Two 6061 aluminum plates spaced by 0.75 inch standoffs house the parallel shaft gear drive. Drivetrain components and layout are shown in Figure 2.5.

Calculations were performed to ensure that the shaft and bearings would be sufficient to handle loads created by the rotor. Shaft calculations using the Modified Goodman Equation yielded a factor of safety of approximately 18.7, indicating that even at rated power and speed, the shaft will experience infinite life. The bearings have a dynamic load rating of 580 lbs or 2580 N, and a calculated maximum reaction force on the rear bearing of 380 N. This results in bearing life of 3.205 x 10⁸ cycles, and approximately 5550 operating hours using a Rayleigh Distribution of IEC Class III Winds. These calculations show that the shaft and bearings selected are sufficient to handle rotor loading; however,

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designed lifespan should be increased by improving bearing quality and relieving loading to achieve a product lifespan of 10 years assuming continuous annual operation.

The purpose of the mechanical emergency brake is to ensure that the turbine has a fail-safe method of braking. This brake allows a user to physically lock the rotor from turning. A hydraulic disc brake system was designed using parts from a GTB hydraulic system disc brake kit. Brackets were fashioned to allow the brake caliper to mount in front of the gear drive. A 2.5 inch 316 stainless steel washer was adapted to serve as the brake disc, as the brake kit discs were too large. The brake kit master cylinder is mounted behind the gear drive, next to the generator. Dot3 brake fluid is pumped from the master cylinder, through a brake line that runs through the gear drive, into the caliper to clamp the brake pads and slow the rotor. This brake is actuated via a lever and bicycle brake cable mounted at the back of the turbine. A user clamps down on a bicycle brake lever, which pulls on the brake cable, pulling one side of the lever back, and pushing the other side of the lever forward, compressing the piston inside of the master cylinder and pumping fluid to the caliper. Reference the supplemental computer drawings document for the turbine drivetrain.



gear-drive optimizes generator speed for rated power output, it is not sufficient to step up the generator voltage to overcome the potential of a 12 V DC battery for wind

speeds lower than 10 m/s. To further step up the voltage at a lower cost, a power transformer is placed in parallel with the gearbox. This transformer operates as long as the voltage produced by the generator is below 14 V RMS. Once the generator voltage exceeds 14 V RMS, a micro-processor will switch a relay such that the transformer is bypassed. This approach is used because at voltages above 14 V RMS the transformer is not necessary. In addition, if the transformer is selected such that it can also handle the

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high currents seen at voltages about 14 V RMS, transformer cost may become prohibitive. By using a micro-processor and relay to selectively employ the transformer, utility and cost are enhanced.

Because the generator produces alternating current, it is necessary to employ a rectifier to produce a clean DC signal, so the power produced by the generator can be used to charge a 12 V DC battery or power a 12 V DC load. A bridge diode rectifier was selected for its simplicity and low cost.

Shaft speed is controlled by controlling the duty cycle of an insulated gate bipolar transistor (IGBT) placed between the rectifier and the power sink. A propeller microprocessor is used to control this duty cycle. The microprocessor is programmed to adjust duty cycle in order to maximize power produced, until the turbine reaches its rated power. Figure 2.7 shows a logic diagram of the shaft speed control algorithm.

The controller can also be used to slow the turbine when the load is disconnected. A relay is placed in parallel between the generator and the transformer, but in series with a dump load. During normal operation the relay remains open all the time. When





the load is disconnected, the controller will sense this by measuring a current of zero at the point of common coupling. It will then command the relay to close, short-circuiting the generator. The current seen by the generator will be large enough to slow the rotor to a safe speed.

2.6 Nacelle, Tower, and Furling

The nacelle and tail is injection molded from the same RTP 2289 HM Plastic as the rotor to maintain high production runs and drivetrain protection. The market turbine will incorporate a variable height tower by using commercial 3 ft long, 3 inch diameter, partially threaded 316 stainless steel rods and tie downs. The customer can achieve the desired hub height by assembling up to 5 rods, or 15 ft





additional height to operational elevation. Preliminary vibrations analysis of the blade passage through the tower, not shown here, sized the tower diameter to 3 inches.

Figure 2.8: Market Turbine Tower sized by vibration analysis

The market turbine will include a passive tail vane for tower top yawing to track the wind. This is often the most effective yawing mechanism for small wind turbines due to the low torque required. Final tower top weight is



Figure 2.9: Nacelle and Tail Vane

not yet determined, so the tail preliminary size is 1 sq ft surface area with a center of pressure 1.5 feet from the rotation point. The turbine tail is also an opportunity to implement a profile evocative of the Jayhawk Windustries logo in order to build company recognition.

2.7 Shroud

The market turbine will have a shrouded rotor to address the safety and performance of the turbine for a marginal increase in cost. Diffuser Augmented Wind Turbines (DAWTs) have been tested



and implemented on small commercial wind turbines very similar to this 1 meter diameter design. This is accomplished by capturing and accelerating the air through the rotor as seen in Figure 2.10. Debate exists over actual augmentation

Figure 2.10: DAWT Flow Field⁹

achieved outside of controlled testing, but a thesis paper

from the University of Auckland claims to have addressed these concerns and still achieved a 2.02 augmentation⁸. As this is a preliminary design report, the engineering of the shroud is not complete and will require further testing, analysis, and potential redesign of the blades. The turbine design will not claim power augmentation until this detailed work is accomplished.



3 Prototype Wind Turbine

Table 3.1: Prototype Turbine Design Specifications					
Rated Power Configuration Rated Speed Rotor Diameter Speed Configuration					
18 W	Horizontal Axis	10 m/s	17 in	Variable Speed	

The electromechanical and turbine configuration remains consistent between the prototype and market turbine to maintain maximal commonality with size and features. The prototype turbine includes a custom built Gearbox with a gear ratio of 4.55: 1.

3.1 Airfoil Selection

The prototype turbine airfoil is the NeoEnergy NE203, a custom designed airfoil for 10 – 200k Reynold's numbers. Data for this airfoil was provided by Dr. Wetzel from wind tunnel testing for a 100W turbine. The prototype turbine will operate well under Re = 100k, so this airfoil is much more appropriate for the small scale turbine than the SG6040 airfoil.

3.2 Blade Design

A blade optimization tool was used for the





Figure 2.3: First Iteration Blade Design from Optimization Tool





Figure 3.1: NE203 Wind Tunnel Data

In order to improve the structural quality and tip deflections, the chord lengths were increased along the span and root for increased structural margins and the trailing edge wall thickness was increased for improved layering in 3-D printing. These modifications affect ideal theoretical aerodynamic performance, but are exchanged for safety, manufacturability, durability, and life cycle cost. The first iteration design required



Figure 3.3: First iteration design in small wind Tunnel Testing provided preliminary aerodynamic results

nine 2mm M2 screws, M2 washers, hex nuts, and significant patience for assembly. This did not comply with the concept of portability and ease of operation, so the design team minimized parts count and assembly time by implementing a hub-less design with just 3 larger socket cap screws shown below.



Figure 3.4: Second Iteration Blade Design; Chords and thickness adjusted to increase stiffness



Figure 3.5: Integrated hub and blade design to reduce parts count and assembly time for portability

The optimized geometry for the prototype turbine was used to predict the rotor performance.



The designed turbine cut-in speed is 2 m/s and is optimized for 16-18 W at 9-17 m/s as shown below.





3.3 <u>Loads</u>

The prototype blades are selective laser sintered from Nylon 12 Glass Fiber purchased from Solid Concepts. This was chosen for its relatively high Young's modulus and aerodynamic finish. The manufacturer reported material properties are found in Reference 3. The edgewise and flapwise forces and moments are the critical loads acting on the blade as determined by the blade optimization code. A 1.42N, or 0.319 lb_f, flapwise force is calculated using a 2-dimensional flat plate in 17 m/s wind and a 2.0 drag coefficient. Both hand calculations and an FEM model using PATRAN 2012.2 were used to verify blade deflections.





For a distributed load over a cantilevered beam:

$$\delta = \frac{qL^4}{8EI}$$
 $q = \frac{1.42N}{0.20m} = \frac{7.08N}{m}$

Average blade dimensions listed below were used to calculate deflection assuming a flat beam.

- Average thickness = 0.0025 m, 0.0082 ft
- Average chord = 0.02 m, 0.066 ft
- Nylon 12 GF Flexural Modulus = 2.24 GPa, 324.88 ksi

The design team assumed that the distributed load acts directly on the face of the blade.

$$I = \frac{1}{12} (.02) (.0025)^3 = 2.60 * 10^{-11} m^4 \quad \delta = \frac{(7.08) (0.20)^4}{8(2.24 * 10^9)(2.60 * 10^{-11})} = 0.024 m, 0.94 in$$

A PATRAN model was developed to measure the static and dynamic blade deflections. The blade was

approximated as seven flat plates with the following dimensions.

Section	Thickness (in)	Avg Chord (in)
1	0.275	1.325
2	0.125	0.958
3	0.090	0.808
4	0.080	0.712
5	0.073	0.565
6	0.068	0.511
7	0.050	0.429

Table 3.3: FEM Blade Geometric Specifications



Figure 3.9: FEM Blade Sections

The geometry was then meshed into 136 rectangular elements and a pressure load of 0.0513

psi at 17 m/s wind speed was applied along the entire span with the root fixed at the location of the

bolts. Next, an inertial load at 2000 rpm was added to the model at the axis of rotation.





Figure 3.10: FEM Pressure Load Blade Deflection



Figure 3.11: FEM Pressure and Inertial Load Blade Deflection

Tip Deflections (in)		
X 0.320		
Y	0.666	
Z	0.528	
Magnitude 0.909		

Table 3.4: Pressure and	Inertial Load Deflections
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The pressure load model resulted in a flapwise displacement of 0.528 inches, or 1.34 cm. The rotor to tower distance is 6.03 inches, leaving a 5.50 inch clearance at 17 m/s. This result is lower than calculations, further validating adequate blade deflections. The second model includes the inertial load



at the maximum 2,000 rpm with moderate deflections shown in Table 3.4. These deflections appear large but are still within margins. Furthermore, the rotor has been tested as high as 4,000 rpm and have not exhibited failure or flapwise and edgewise deflections greater than 0.5 inches by inspection.

Using the previously derived 1.42 N, or 0.319 lb_f, pressure load, the flapwise bending moment is 0.16 N-m, or 0.12 ft-lb_f. This results in a bending stress of 7.48 MPa, or 1.085 ksi, with a material flexural strength of 61 MPa, or 8.85 ksi, and Margin of Safety of 7.16. The edgewise bending moment is calculated to be 0.44 N-m, or 0.32 ft-lb_f, for the 18 gram blade spinning at 2000 rpm to a complete stop in half of a second. The loaded stress is 0.249 MPa, or 0.036 ksi, and the calculated Margin of Safety is 23.52. These calculations assume a simple constant area cantilever beam and will be validated with testing.

3.4 Drivetrain Components

The drivetrain components of the prototype turbine were designed based off those of the market turbine. Because the power required for the prototype turbine is significantly lower, only a single step parallel shaft gear drive was employed. This gear drive employs an 11 tooth pinion and a 50 tooth spur gear equipped with a 5 mm bore hub and locking set screw. These gears are again housed by two 6061 aluminum plates spaced by 0.75 inch standoffs. 5 mm flanged roller ball bearings are mounted in these plates to support a 5 mm rotor shaft. These bearings support radial and axial thrust load. The shaft is constrained axially by the placement of Belleville disc washers in between the spur gear and the bearings. The washers essentially constrain the spur gear between the bearings, and because the spur gear is constrained to the shaft via a locking set screw, this also constrains the shaft axially. Results of employing the Modified Goodman Equation indicate a factor of safety of approximately 30. The bearings have a dynamic load rating of 97 lbs or 431 N, and a calculated maximum reaction force on the rear bearing of 23.4 N. This results in bearing life of 6.25 x 10⁹ cycles, or roughly 52,000 hours (slightly less than six years) operating at a constant 2000 revolutions per



minute. These calculations show that the shaft and bearings selected are sufficient to handle rotor loading. The mechanical brake system for the competition turbine is identical to that of the market turbine, except for minor adjustments in placement of components, and use of a 2 inch diameter washer to reduce nacelle dimensions. Computer aided drawings of the drivetrain are provided in the supplemental document and shown in Figure 2.5.

The power electronics of the prototype turbine are also very similar to those of the market turbine; the primary differences are of size and capacity. Because the prototype turbine will not produce as much current, even at high power outputs, the transformer is allowed to remain in use throughout the entirety of the turbine's power regime. In addition, the voltage of the DC Power Sink is 5 VDC, rather than 12 VDC. This difference necessitates different sized resistors, transistors, and transformer. The controller performs the same functions as those of the market turbine. Figure 2.6 shows a schematic diagram of the competition turbine electronics.

4 **Power Electronics and Drivetrain Testing**

4.1 Generator Testing

Testing was performed frequently to identify design characteristics and parameters. The zero load voltage-shaft speed relationship of the generator was determined by spinning the generator with a cordless drill, and measuring voltage output with a digital multimeter. This relationship was used to determine the gear drive gear ratio and transformer ratio necessary to increase the voltage to a value high enough to allow the turbine to inject current out of the rectifier and into the load.

4.2 <u>Wind Tunnel/Power Electronics Testing</u>

Early wind tunnel testing focused on identifying the performance characteristics of the rotor by varying the load resistance at each wind speed with a rheostat. Preliminary tests were performed in a small wind tunnel, but this presented problems, as tunnel wall boundary effects prevented accurate measurement of wind speed in the rotor plane. Subsequent tests in a larger wind tunnel were more useful. A 5 V DC Dell computer power supply capable of sourcing 35 amps was employed with two 0.1



ohm 100 W resistors to replicate the competition power sink. At first the team tested the turbine without any control, to determine if an uncontrolled turbine would meet any of the competition requirements. Because the low resistance of the load restricted shaft speed (and thus power) at all wind speeds, a rheostat was placed in series with the rectifier and load, in order to vary load resistance



Figure 4.1: Large Wind Tunnel Testing; Used to test power electronics and performance

and imitate the effect of a controlled DC buck converter. Results indicated that varying the load resistance as seen by the generator could allow for power optimization. To minimize resistive loss, an insulated gate bipolar transistor was selected as the switching element for the buck converter. To date, the prototype rotor and drivetrain has been wind tunnel tested for a total of 1.5 hours at an average speed of 1200 rpm. No structural degradation or hysteresis has been recorded in the performance data.

5 <u>Conclusions</u>

This report details the process by which a 1 meter diameter 400W HAWT was designed by Jayhawk Windustries. The 400W turbine is sized to address the market needs described in the Jayhawk Windustries Business Plan. A small scale prototype is fabricated for design validation and testing. Multiple iterations of blade and gearbox design are accomplished and the design is supported by structural analysis and wind tunnel testing data. A power control system is designed and determined to be necessary for optimal tracking of TSR and performance. The preliminary design addressed in this research is safe for testing, and demonstrates the market potential for the larger market turbine, though the latter requires further testing and iterations to address safety, portability, and shroud performance augmentation before a final product is brought to the market.





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