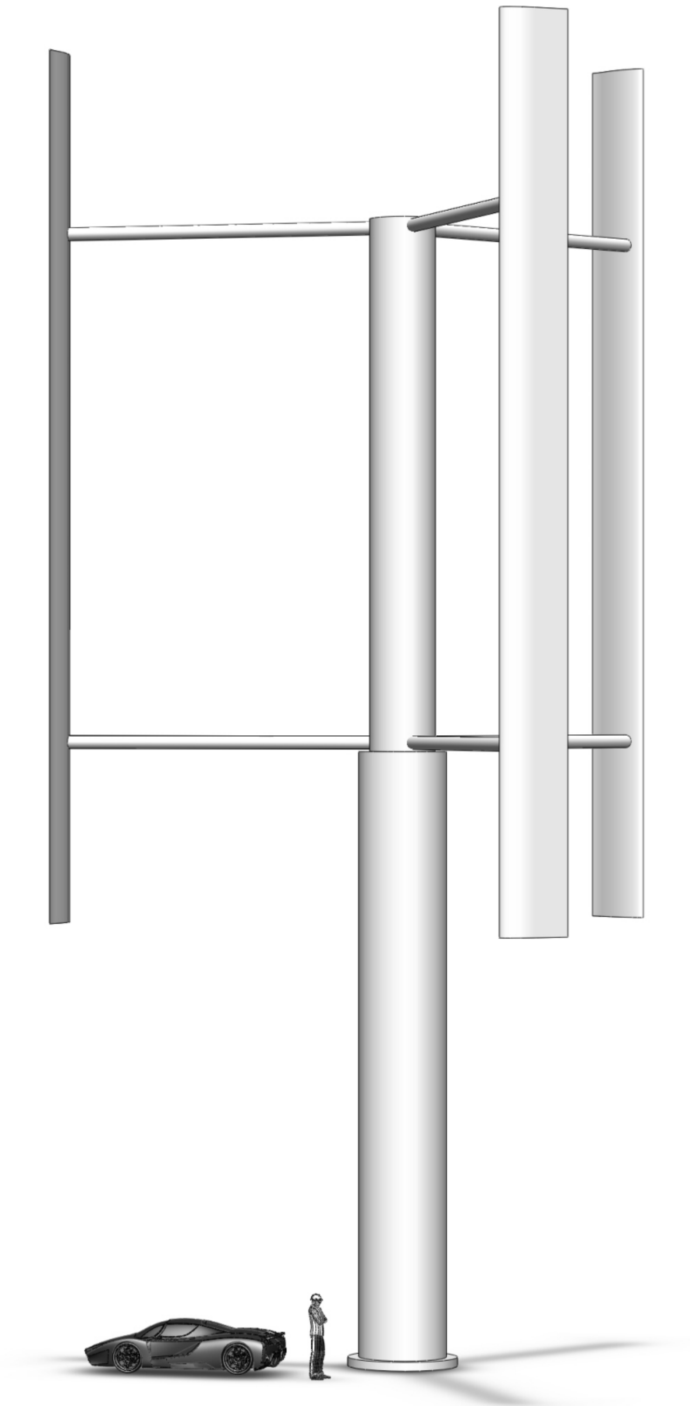
Design of a Vertical–Axis Wind Turbine

Phase II

7 March 2014



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ABSTRACT

This report details Phase II of the vertical-axis wind turbine design conducted by MUN VAWT Design. Included are information about the applicable regulations and standards to be followed in the design process, a discussion of aerodynamic modelling, details on the preliminary structural design, an overview of environmental concerns, and a preliminary economic analysis.

ACKNOWLEDGMENTS

We would like to thank Dr. Sam Nakhla for his help and guidance.

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1 INTRODUCTION

In the province of Newfoundland and Labrador, many remote communities must rely on diesel generators to meet their power requirements. With the rising cost of fuel and transportation, alternatives to total reliance on diesel for electricity generation are becoming attractive. To this end, MUN VAWT Design will design a vertical-axis wind turbine (VAWT) for operation in remote communities in Newfoundland and Labrador.

The design will:

- Work in conjunction with diesel generators, not replace them
- Be robust and simplistic
- Produce sufficient energy to offset diesel fuel costs
- Allow for transportation by boat to remote areas

Based on research conducted for the first design report, a 3-bladed H-rotor configuration with DU06-W200 airfoils was selected as the preliminary VAWT design. This report will elaborate on this design through aerodynamic and structural analyses, relevant regulations, environmental effects, and a cost-benefit analysis.

2 REGULATIONS

2.1 General

After completion of Phase I research into sizing and weather patterns, the standards and regulations that govern wind turbines were consulted. The standard chosen to consult was IEC 61400-1 titled Wind turbines – Part 1: Design Requirements, developed by the International Electrotechnical Commission (IEC). The IEC is a worldwide organization for the standardization of all electrical, electronic and related technologies. The goal of the IEC is to "promote international co-operation on all questions concerning standardization in the electrical and electronics fields."

The purpose of this section of the IEC 61400 is to specify the minimum design requirements for wind turbines in order to ensure the engineering integrity of the turbine. It is not intended to be used as a complete specification or instruction manual but to provide the engineering and technical requirements to guarantee the safety of the structural, mechanical, electrical, and control systems. The standard can be used for any size wind turbine but is not intended for the design of offshore turbines; small wind turbines are also subject to IEC 61400-2.

This standard covers all subsystems of the turbine including: control systems, protection mechanisms, internal electrical systems, mechanical systems and support structures. Also, the standard should be used in conjunction with other applicable standards where appropriate. Although the standard is concerned with requirements that apply to the design, manufacture, installation and manuals for operation and maintenance of a wind turbine and the associated quality management process the area of concern for the project is the design requirements set out in the standard. [1]

2.2 Classes

A turbine following this standard is required to meet one of two types of safety classes, normal and special. The normal safety class is comprised of nine different cases as seen in Table 1, which are based on wind speed and turbulence parameters. This is because wind conditions are the primary source of external conditions that will affect the structural integrity of a turbine and the multiple cases allow for a wide variety of sites for different conditions without being overly specific. These conditions are not meant for the offshore environment or severe storms such as hurricanes. The class seen in the table is for the special safety factor which is chosen by the designer and/or customer.

Table 1: Basic parameters for wind turbine classes

Wind Turbine Class	I	II	III	S
Vref (m/s)	50	42.5	37.5	Special
A Iref	0.16			
B Iref	0.14			
C Iref	0.12			

The desired class for the project to meet is IA, which represents the highest wind reference wind speed and turbulence intensity categories used. This allows the turbine designed herein to operate in the widest variety of environments. The Vref or reference wind speed refers to an extreme 10 minute average wind speed with a recurrence period of 50 years acting at the turbine hub height. Iref is the expected turbulence intensity squared at 15m/s wind speed, where turbulence is defined as a random deviation in the wind speed from the 10 minute average. By meeting this safety classification the turbine will have a minimum design life of 20 years.

2.3 Design Load Cases

The verification of the design can be completed through calculation or full scale testing of a wind turbine. In order to meet the standard through calculations, the use of a structural dynamics model is required. This model shall be used to analyze a number of design load cases specified in the standard. These are the minimum cases that a turbine must be designed for and are presented in Table 2. The design load cases consist of a combination of normal and extreme external conditions acting at various phases of turbine operation that are likely to result in critical situations. The list of abbreviations used in Table 2 can be found in Appendix A.

Table 2: Design Load Cases

Design Situation	DLC	Wind Conditions	Other Conditions	Type of Analysis	Partial Safety Factors
1) Power Production	1.1	NTM Vin<Vhub<Vout	For extrapolation of extreme events	U	N
	1.2	NTM Vin<Vhub<Vout		F	*
	1.3	ETM Vin<Vhub<Vout		U	N
	1.4	ECD Vhub = Vr-2, Vr, Vr +2		U	N
	1.5	EWS Vin<Vhub<Vout		U	N

2) Power Production plus occurrence of fault	2.1	NTM Vin<Vhub<Vout	Control System fault or loss of electrical network	U	N
	2.2	NTM Vin<Vhub<Vout	Protection System or preceding internal electrical fault	U	A
	2.3	EOG Vhub = Vr ± 2 and Vout	External or internal electrical fault including loss of electrical network	U	A
	2.4	NTM Vin<Vhub<Vout	Control, protection, or electrical system faults including loss of electrical network	F	*
3) Start up	3.1	NWP Vin<Vhub<Vout		F	*
	3.2	EOG Vhub = Vin, Vr ± 2 and Vout		U	N
	3.3	EDC Vhub = Vin, Vr ± 2 and Vout		U	N
4) Normal Shut down	4.1	NWP Vin<Vhub<Vout		F	*
	4.2	EOG Vhub = Vr ± 2 and Vout		U	N
5) Emergency shutdown	5.1	NTM Vhub = Vr ± 2 and Vout		U	N
6) Parked (Standing Still or idling)	6.1	EWM 50 year recurrence period		U	N
	6.2	EWM 50 year recurrence period	Loss of electrical network connection	U	N
	6.3	EWM 1 year recurrence period	Extreme yaw misalignment	U	N
	6.4	NTM Vhub < 0.7Vref		F	*
7) Parked and fault conditions	7.1	EWM 50 year recurrence period		U	A
8) Transport, assembly maintenance and repair	8.1	NTM Vmaint to be stated by manufacturer		U	T
	8.2	EWM 50 year recurrence period		U	A

Due to the nature of the project not all the design load cases listed are relevant to the design of the turbine herein. The following cases will be neglected: 2.1 thru 2.4, 6.2, 6.3, 7.1, 8.1 and 8.2. The majority of these are due to the fact that the electrical network and control systems, the transport, assembly, maintenance and repair fall outside of the scope of the project. Case 6.3 is neglected because a vertical-axis wind turbine does not feature a yaw system.

Brief definitions of the applicable cases are presented below:

- DLC 1.1 and 1.2 are for normal turbulence during normal operation.
- DLC 1.3 is for ultimate loading resulting from extreme turbulence conditions.
- DLC 1.4 and 1.5 are for transient cases that have been found to be critical events.
- DLC 3.1 thru 3.3 specify transient events for the period the turbine moves from a standstill to normal operation.
- DLC 4.1 and 4.2 are the opposite cases of DLC 3.1 and 3.2.
- DLC 5.1 is for the same operation as DLC 4.1 and 4.2 but under an emergency situation.
- DLC 6.1 for standstill or idling is evaluated under extreme wind and 6.4 is for normal turbulence.

The simulations of the design load cases are required to be run at least six times for a 10-min stochastic realization. However, case 5.1 requires a minimum of 12 simulations. Another requirement is that the first 5 seconds of data be neglected for turbulent studies because of the effect that initial conditions have on dynamic analyses.

3 AERODYNAMICS

3.1 Preliminary Sizing

Preliminary sizing was conducted using the wind power density formula:

$$W/_{m^2} = 0.5 * \rho_{avg} * C_p * V^3$$

This formula allows for the calculation of power generated per square metre of turbine swept area given average air density ρ_{avg}, power coefficient C_p, and wind speed V. Using an estimated power coefficient of 0.4, preliminary sizing indicates that in order to develop the desired maximum output power of 100kW at a wind speed of 10.5 m/s, the swept area of the turbine will need to be approximately 320 m^2.

3.2 Preliminary Analysis

Preliminary aerodynamic analysis was conducted using QBlade, an open-source wind turbine simulation software developed by a team at the Technical University of Berlin, Germany. The software is integrated with the airfoil simulation code XFoil (developed by Mark Drela at the Massachusetts Institute of Technology) to provide lift and drag polars for standard NACA airfoils and extrapolate them to 360° angle-of-attack. Using these polars, QBlade can evaluate the performance of a vertical-axis wind turbine using the double-multiple streamtube model.

3.3 The Double-Multiple Streamtube Model

The double-multiple streamtube model (DMS model) for a vertical-axis wind turbine is an algorithm that computes aerodynamic forces on the turbine blades, taking into account that the blades pass through the flow stream twice. The blades are considered to absorb energy from the flow once in the upstream side of turbine, and once in the downstream side.

Similar to the actuator disk theory, the DMS method applies conservation of momentum to the flow impinging on the turbine. The swept area of the turbine is divided into two sets of elements (streamtubes). For each of these streamtubes, momentum conservation is compared with the blade forces computed from lift and drag coefficients at all azimuth angles. The upstream side is evaluated separately from the downstream side, with the induced flow velocity at the exit of the upwind streamtubes being used as the inflow for the downwind streamtubes. [2]

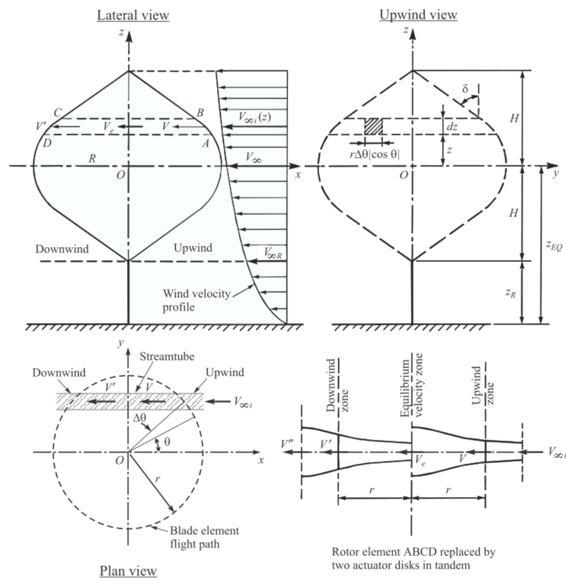


Figure 1: Discretization of VAWT for DMS method

Some limitations of this method identified by the software's developers include convergence issues for very high solidities and high tip-speed ratios. Since the turbine design is to have a moderate solidity of approximately 0.35, the solidity limitation was not anticipated to cause any problems. The limitation on tip-speed ratio was avoided by prescribing the wind speed and turbine rotational speed for each simulation such that excessive tip-speed ratios were avoided. The software also neglects the effects of the wake of the support tower, and cannot account for dynamic stall of the turbine blades. Correction for these limitations is detailed in Section 3.5.

3.4 Turbine Evaluation

To perform preliminary evaluation of the turbine, a simple H-rotor-style VAWT was designed in QBlade with the specifications shown in Table 3. A variety of aspect ratios (ratio of blade height to rotor diameter) were evaluated to determine the optimal configuration.

Table 3: Turbine Parameters

Configuration	1	2	3
Airfoil	DU06-W200	DU06-W200	DU06-W200
No. of Blades	3	3	3
Blade Chord	1.6 metres	1.6 metres	1.6 metres
Rotor Height	20 metres	17.9 metres	16 metres
Rotor Diameter	16 metres	17.9 metres	20 metres
Aspect Ratio	1.25 : 1	1 : 1	1 : 1.25

QBlade was used to generate plots of power coefficient vs. tip-speed ratio for the three configurations. From these results, the 1.25 : 1 aspect ratio gave the highest power coefficient and lowest tip-speed ratio at peak power coefficient. A low tip-speed ratio is considered desirable since high tip speeds are correlated to high levels of aerodynamic noise [3]. Higher aspect ratios may give higher power coefficients; however, it was decided to limit the height of the blades to 20 metres for ease of shipping.

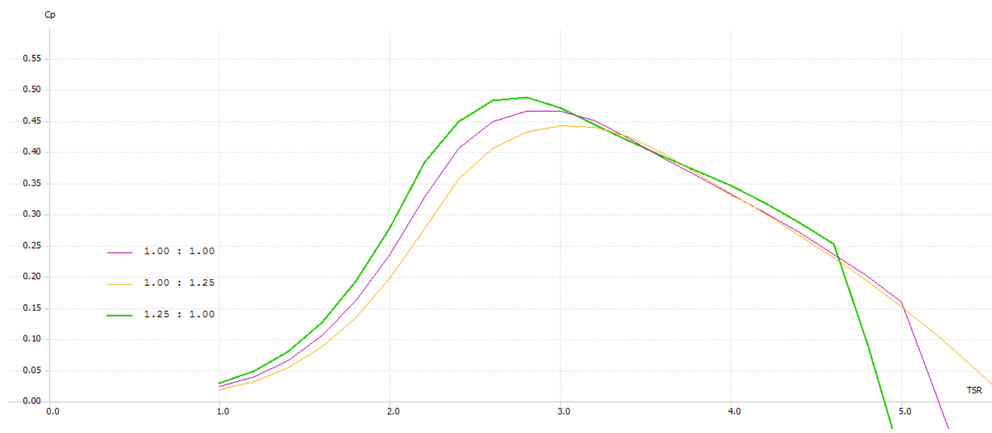


Figure 2: Power Coefficient vs. Tip-Speed Ratio for Various Aspect Ratios

3.5 Aerodynamic Results

To account for software limitations, an existing H-rotor turbine with known performance was modeled in QBlade to allow for comparison between experimental and theoretical data. The turbine used for this analysis was the VAWT 260. Developed in Great Britain in 1989, this experimental H-rotor turbine had a rated output of 100kW, a height of 13.3 metres, and a rotor diameter of 20 metres [4] [5]. By modelling this turbine in QBlade and comparing the results with experimental data, it was found that the software overestimated power coefficient by approximately 20%. Therefore, a correction factor was applied to the calculated power coefficient as follows:

$$C_{p-corrected} = C_{p-raw} * \eta_{correction} * \eta_{generator}$$

Where $\eta_{correction} = 0.8$ and $\eta_{generator}$ is assigned a typical value of 0.96. The corrected power coefficient plot is shown below.

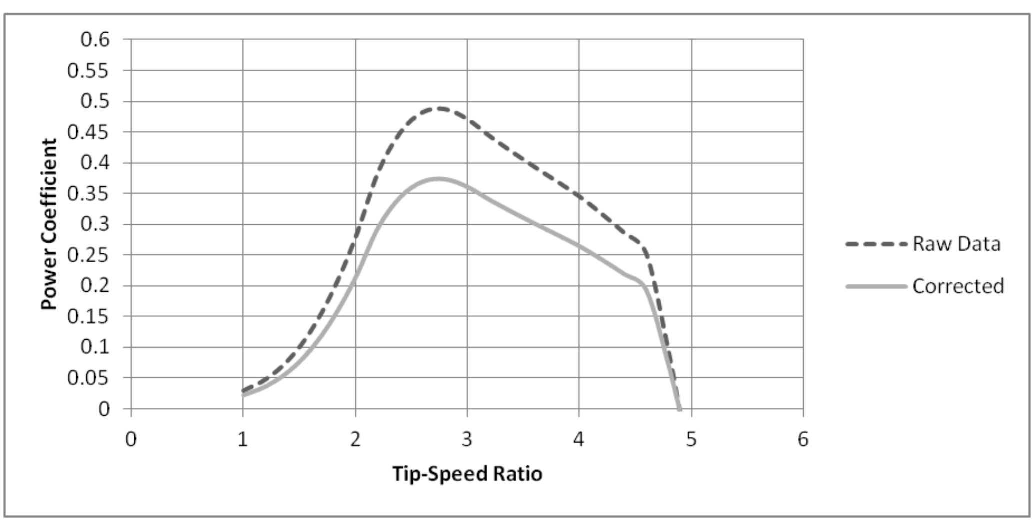


Figure 3: Raw and corrected power coefficient curve

From the power coefficient plot, the best operating point for the turbine was determined to be at a tip-speed ratio of 2.7.

The rotor was then modeled under wind speeds from 2 m/s to 35 m/s (the maximum sustained wind recorded in the target area). The rotational speed of the turbine was allowed to vary to maintain the optimal tip-speed ratio. Output power, rotor torque, and aerodynamic forces on the entire rotor as well as the individual blades were recorded over this range. The rotor was also analysed in a parked condition at a wind speed of 50 m/s to obtain the rotor loads at the IEC 61400-1 Class IA reference state.

While the results show that outputs in the megawatt range are theoretically possible from this turbine, in practice, this will require a larger generator and heavier structure. Given that large power outputs will only occur at high wind speeds that are very rarely recorded, it was deemed beneficial to govern the turbine to 35 RPM and apply a cut-out wind speed of 26 m/s. This greatly reduces aerodynamic loads on the rotor and torque on the driveshaft, allowing these components to be lighter and less costly. While the annual output of the turbine is decreased as a result, the cost of building the turbine structure and components large enough to withstand the increased dynamic forces would cause the system to be prohibitively expensive.

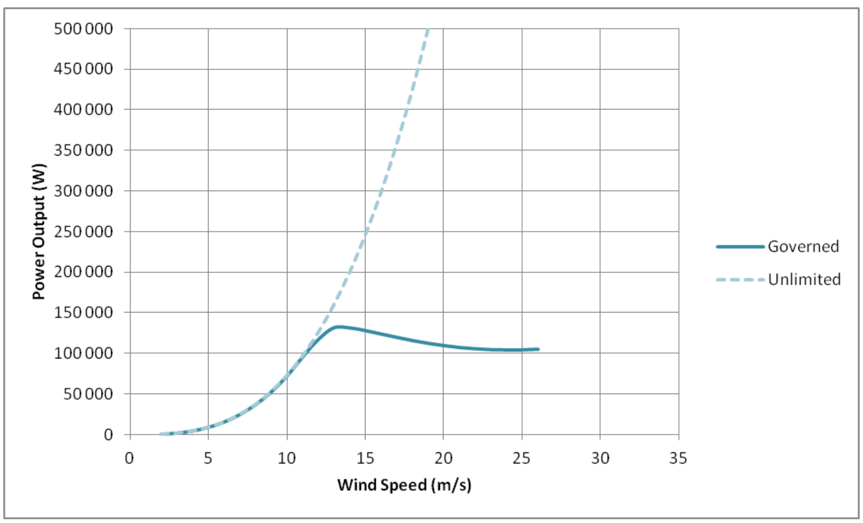


Figure 4: Turbine Power Curve

Estimated required sizes for the shaft and tower are shown below for both the governed and unlimited turbine. Due to the significant increase in component size for the unlimited turbine, it is estimated that the increase in system cost would outweigh the additional power generated. It is expected that governing the turbine will increase financial benefit over the life of the system.

Table 4: Component size estimate for unlimited and governed turbines

Control State	Unlimited	Governed
Required Shaft OD (mm)	600	250
Required Tower OD (mm)	4900	2000
Estimated Annual Output (MWh)	585	375

Methods for governing a wind turbine include: [6]

- Active regulation: the pitch of the turbine blades is altered during operation to reduce torque at high wind speeds. This method allows for higher aerodynamic efficiency, but is more complex.
- Passive regulation: The turbine blades are designed to stall at high wind speeds, reducing lift and lowering the rotational torque. This method is simpler to implement than active regulation, but often results in lower power coefficients.
- Rheostatic braking: Excess generator current is routed through a network of resistors and dissipated as heat, increasing generator torque for a given rotational speed. [7]

In order to brake the turbine above the cut-out wind speed, a combination of aerodynamic braking and mechanical drum brakes are usually used. Mechanical brakes cannot usually be used alone because of the large amount of heat generated in the process of stopping a turbine from full speed. Braking and governing methods will be examined in more detail in the final phase of the project.

With the turbine performance specified, a torque curve for the rotor running at the best operating point was generated. This curve will be used to select a generator.

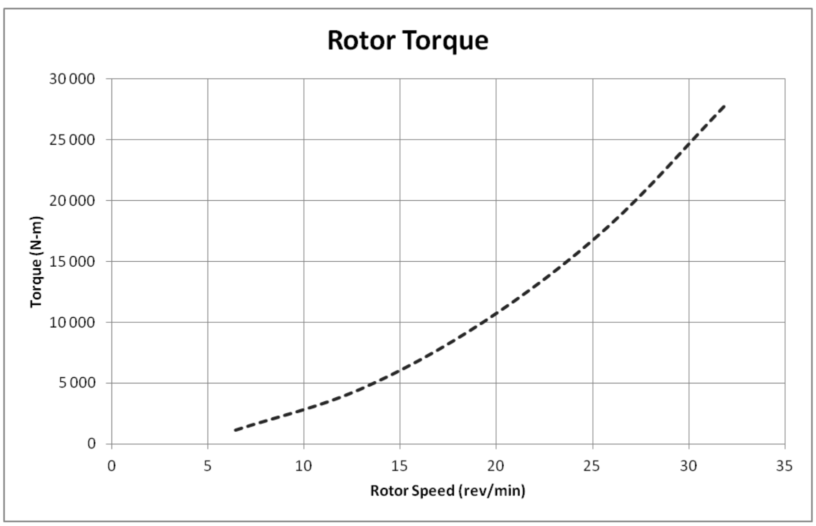


Figure 5: Rotor Torque Curve

4 STRUCTURAL DESIGN

In addition to operating at peak aerodynamic efficiency it is important to design a VAWT that can withstand lift forces, drag forces and wind loads. Structural analysis was performed on individual components of the VAWT since each part was subjected to different loads. The following components were individually analyzed:

- Driveshaft
- Tower
- Blades

For this section of the project, basic solid mechanics as well as bearing selection, manufacturing constraints, maintenance, and transportation concerns were considered. The final report will cover a more comprehensive structural analysis including finite element analysis (FEA), strut sizing, vibration considerations, bolt design and/or weld types. Note that the numerical values calculated in this section of the report are subject to change as the design is refined and more detailed aerodynamic modelling is conducted. The procedure, however, will not change drastically.

The structural analysis was conducted at two operational states. The first case examines the turbine at maximum operating condition at a wind speed of 26 m/s and a rotor speed of 35 rev/min. The second case examines the turbine in a parked condition at a wind speed of 50 m/s, in order to comply with the IEC 61400-1 Class IA reference state.

4.1 Material Selection

For structural design it is important to consider all materials appropriate to the application. For the design of this turbine, the material should be cost effective and capable of providing the required mechanical properties for each application.

For the turbine blades, the material selection is ongoing based on the required stresses that will be applied by lift and drag forces. It was determined that the material should have the following attributes:

- Suitable strength to weight ratio
- Rigid
- Able to withstand calculated shear and normal stresses
- Weather resistant
- Fatigue resistant
- Will not cause galvanic corrosion on struts
- Smooth surface finish for aerodynamic purposes

Materials that were considered were steel, aluminum, and composites. Steel can be immediately rejected due to weight. Aluminum would satisfy the weight requirement, but exhibits poor fatigue resistance. For this reason, the decision was made to use a composite material. Some types of composites that can be neglected immediately are carbon fibres because they have unsuitable galvanic corrosion properties and higher cost. Through research, it was found that glass-fibre composites are commonly used in modern wind turbines and exhibit favourable mechanical properties. Further investigation will be conducted in the final phase of the project.

The cylindrical support structure of the turbine will be made of A35 structural steel, a cost-effective material typically used in larger structures. This steel is subject to change upon contacting manufactures. The properties of A35 steel are listed in the table below:

Table 5: Mechanical properties of A35 structural steel

Density	7 800 kg/m^3
Ultimate Tensile Strength	400-550 MPa
Tensile Yield Strength	250 MPa
Modulus of Elasticity	200 GPa
Shear Modulus	79.3 GPa
Coefficient of Thermal Expansion	13.0 (10^{-6} m/m*K)
Poisson's Ratio	0.260

The driveshaft will be manufactured out of AISI 4340 steel. This steel provides the optimal machinability properties and is typically used in many industrial rotational engineering applications, such as large shafts and axles, for its mechanical properties. These properties are listed below:

Table 6: Mechanical properties of AISI 4340 steel

Density	7 850 kg/m^3
Ultimate Tensile Strength	745 MPa
Yield Tensile Strength	470 MPa
Modulus of Elasticity	190 - 210 GPa
Shear Modulus	80.0 GPa
Coefficient of Thermal Expansion	12.3 (10^{-6} m/m*K)
Poisson's Ratio	0.27 - 0.3

4.2 Turbine Blades

To model the aerodynamic loading on the turbine blades to develop a relationship for the optimal strut configuration, the wind was assumed to be a uniformly distributed load. The straight design of the blade allows for the model to be represented as follows:

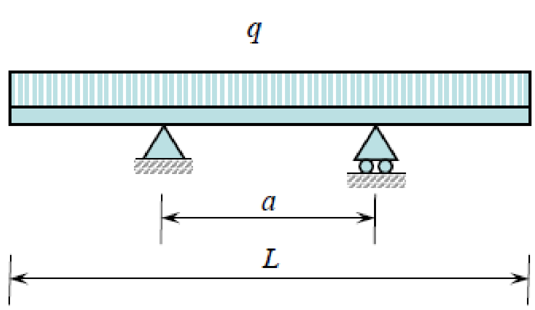


Figure 6: Aerodynamic loading on turbine blades

In Appendix B of the report, calculations were completed to equate the moments at the supports and moments at the middle of the beam. A relationship was developed to determine the optimal strut spacing based on the length of the blade.

$$a/L = 0.586$$

Where a is the spacing between the struts and L is the length or height of the blade. Therefore, the optimal strut spacing is 58.5% of the turbine blade length.

In order to determine the proper composite material for the turbine blades, the Von Mises yield criterion must be applied to determine the required strength of the material. The Von Mises stresses were computed at several points on the airfoil over various attack and azimuth angles. The points in question are plotted below:

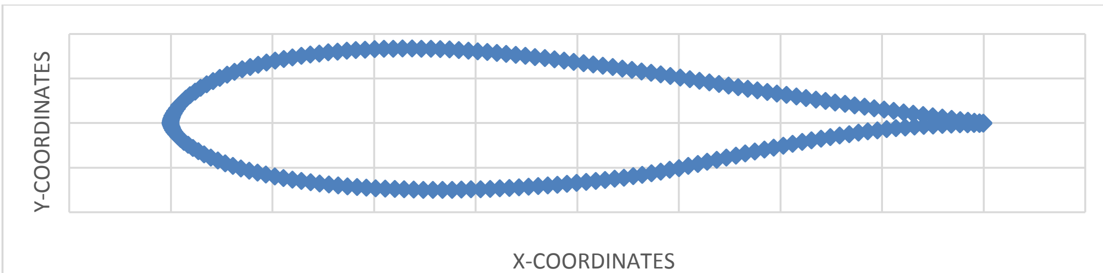


Figure 7: Plot of DU06-W200 airfoil

The DU06-W200 airfoil chosen for this turbine is under asymmetric loading due to its geometry. However, the secondary area moment of inertia I_{xy} is approximately equal to zero, and therefore the bending stress on the airfoil could be modeled as symmetric and later added using superposition theory. To start, bending stresses were calculated in the x-plane using the following equations:

$$\sigma_x = -M_x*y/I_{xx} \quad \text{and} \quad \sigma_x = M_y*x/I_{yy}$$

Where σ_x is bending stress, M_x & M_y are moments about respective planes, x and y are the moment arms from the point on the edge of the airfoil to the centroid, and I_{xx} & I_{yy} are the second area moments of inertia for their respective planes. The second area moments were determined from SolidWorks. The x and y values in the equation are the distances from the point of the edge of the airfoil to the centroid. These values varied depending on what point along the edge was being assessed. To find the moments at the midpoint of the blade, where the maximum stress is applied, the following equation was used.

$$M_{Mid} = (q*L^2/4)*(a/L - ½)$$

Where q is defined as the applied force. In this case, the applied force is the resultant of the lift and drag force in the x and y directions. To calculate M_x and M_y, the x and y components of the lift and drag forces based on angle of attack were superimposed. Leading to the following set of equations:

$$M_x = (F_y*L^2/4)*(a/L - ½);\ \text{where: } F_y = F_L*\sin\alpha + F_0*\cos\alpha$$

$$M_y = (F_x*L^2/4)*(a/L - ½);\ \text{where } F_x = F_L*\cos\alpha + F_D*\sin\alpha$$

Once the bending stresses were calculated, they were added using the superposition theory.

$$\sigma_z = [-M_x*y/I_{xx}] + [M_y*x/I_{yy}]$$

The next step in finding Von Mises stresses was to determine the torsional shear stress that corresponds to its respective bending stress. The following equation was used:

$$\tau = T*r/I_P$$

Where T is torque caused by the lift and drag forces, r is the moment arm (typically the radius if the torque is applied to a cylindrical shaft), and I_p is the polar moment of inertia. To generalize the airfoil, the perimeter was found and translated into the circumference of a circle. This method will give a rough approximation of the shear stress by means of shape factoring, using the circle to find a proportional radius for the calculation. I_P was determined through the use of SolidWorks and the torque was determine by finding the moment caused by lift and drag about the centroid. The lift and drag forces are applied at the centre of lift, which is 0.25*chord from the leading edge. Due to the orientation of the airfoil the x components of the lift and drag are applied through the same plane as the centroid and therefore do not cause moments. However, the y components cause a moment that is modeled as follows:

$$T_y = [F_L*\sin\alpha + F_D*\cos\alpha]*\beta$$

Where T_y is the torque caused by the y components of the lift and drag forces and β is the horizontal distance between the centroid and the centre of lift. This torque with the radius of the translated circle and I_P can be used to calculate an approximate shear stress.

The equation for Von Mises yield criterion includes the bending and shear stresses calculated above. The stress calculated using Von Mises will be used to select a composite material in the next phase of the project. The Von Mises stress formula is as follows:

$$\sigma_{Von\ Mises} = \frac{\sigma}{2} \pm \sqrt{\left[\frac{\sigma}{2}\right]^2 + \tau^2}$$

Using Microsoft Excel, σ_{max} was determine to be 5.3 MPa at an azimuth angle of 27.5 degrees.

4.3 Driveshaft

The design of the turbine driveshaft was primarily governed by the torque generated from the rotating blades.

The yield strength of AISI 4340 is:

$$\sigma_y = 420\ 507\ 000\ Pa$$

Since the maximum principal stress is σ_y and the minimum principal stress is zero, from the Maximum Shear Stress Theory:

$$\tau_{max} = \frac{\sigma_y}{2} = 210\ 253\ 500\ Pa$$

The torsion formula for a solid cylindrical shaft is:

$$\tau_{max} = \frac{TDo}{2J}$$

Where,

$$J = \frac{\pi}{2}\left[(\frac{D_o}{2})^4 - (\frac{D_i}{2})^4\right]$$

*T – torque (N*m)*

Do – outer diameter (m)

Di – inner diameter (m)

4.3.1 Driveshaft – Results

Using the torsion formula, a multitude of values could be selected for the inner and outer diameters of the shaft provided the calculated torsion does not exceed the allowable shear stress. For the initial sizing, inner and outer diameters of 0.26 m and 0.3 m were selected. From the initial aerodynamic analysis using QBlade, the maximum torque on the drive shaft was approximately 100 000 N*m. Thus, using the torsion formula:

$$\tau_{max} = 43\ 279\ 988\ Pa$$

Since the torsion is less than the allowable shear stress, the selected shaft dimensions are appropriate with a safety factor of 4.9.

4.4 Driveshaft Components

There are many types of bearings and couplings to consider when designing a vertical rotational shaft. The VAWT, as noted in the previous section, will contain a driveshaft that consists of two equal length shafts. To support and connect these shafts, the bearings and couplings must be able to satisfy the torsional and axial loads. Other design criteria for bearing and coupling selection are life span, reliability, cost and material.

4.4.1 Bearings

When choosing a bearing, it is important to know what types of bearings are available on the market and how functions are required of the bearing. Types of bearings include ball bearings, roller bearings, and thrust bearings. Typically, bearings are manufactured to withstand radial loads, thrust loads, or a combination of both. For the driveshaft of the vertical axis wind turbine, a combination of radial and thrust loading will be required. This led to a comparison of tapered roller bearings and angular contact ball bearings. Angular contact ball bearings are generally used for high rotational speed and relatively lower loads, in contrast to tapered roller bearings which are used for lower rotational speed and higher loads. With the requirements of a heavy force load at a low rotational speed of approximately 30 rpm, it was decided that the optimal bearing would be a tapered roller bearing.

4.4.2 Mechanical Couplings

The VAWT drive shaft is required to be sectioned in order to meet the shipping criteria set in the project objectives. A mechanical connection is required to join the sectional drive shaft upon installation and compensate for misalignment and thermal expansion in service. To select a coupling, the features required of the coupling must first be determined. Mechanical shaft couplings can be either rigid, flexible/compensating, or clutch type couplings. For the VAWT driveshaft assembly, the coupling must connect two shafts together, transmit a relatively high torque at low rotational speeds, require minimal maintenance, and provide sufficient flexure and axial relief. Due to the constraints, clutch type and rigid couplings can be eliminated immediately from the design, making the coupling of choice flexible or compensating. With the constraints determined to be minimal maintenance, low cost, and high life expectancy, companies that supply various types of couplings were investigated. These companies include:

- Love-Joy
- Ruland
- Hayes
- Renold

Renold has been contacted regarding information on the "Renold Hi-Tec RB" mechanical coupling that specifies shaft to shaft connection for applications such as wind turbines. The features listed for the coupling include the following:

- Intrinsically fail safe
- Control of resonant torsional vibration
- Maintenance free
- Severe shock load protection
- Misalignment capability
- Zero backlash
- Low cost

The driving features of this coupling are the misalignment capability and maintenance free attributes. These features provide significant advantages over common connections such as constant-velocity joints or universal joints, because they do not require any lubrication and can allow for axial as well as angular misalignment.

4.5 Tower

The tower was analyzed as a vertical hollow cylinder subjected to wind loads, rotor forces, and the weight of the VAWT itself. This will cause axial stresses, shear stresses and bending stresses in the structure. The tower supports the upper structure and protects the shaft from the elements (wind, rain, snow, etc.). To reduce the amount of material, it was important to minimize the thickness and withstand the worst possible wind loads.

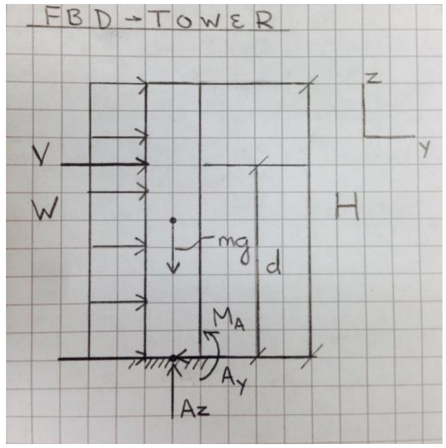


Figure 8: Free-Body Diagram of Tower

The tensile yield strength of A35 structural steel is:

$$\sigma_y = 250{,}000{,}000\ Pa$$

From the figure above,

A_z – reaction force due to axial forces (N)

A_y – reaction force due to shear forces (N)

*M_A – reaction moment (N*m)*

W – distributed wind load on tower (N/m)

V – resultant load on tower from wind load on blades (N)

H – tower height (m)

d – distance from ground to where V acts (m)

The following values were either selected or known for the structural analysis,

ρ (air density) = 1.25 kg/m^3

v (max wind speed) = 50 m/s

C_D (drag coefficient for a vertical cylinder) = 1.17

H (height) = 28 m

d (distance from ground to where V acts) = 22.5 m

M_T (mass of the VAWT) = 43 000 kg (estimated)

4.5.1 Tower – Axial Stress

From basic solid mechanics, the axial stress at the base of the structure is:

$$\sigma_{axial} = \frac{P}{A}$$

Where,

P – axial reaction force (total weight of the VAWT) (N)

A – cross-sectional area of the tower (m^2)

4.5.2 Tower – Shear Stress

The shear stress in the tower structure is caused by wind loads on the tower and the resultant wind loads on the blades (obtained during aerodynamic analysis). The formula for the resultant force caused by wind loading on the tower is:

$$F_W = \frac{1}{2}\rho v^2 C_D (D_O H)$$

Where,

ρ – density of air (kg/m^3)

v – wind velocity (m/s)

C_D – drag coefficient

*Do*H – projected area where the wind load is applied (for a vertical cylinder, the projected area is the outer diameter multiplied by the height) (m^2)*

The following equation was used to convert the single load (F_W) to a distributed load on the tower:

$$W = \frac{1}{2}\rho v^2 C_D D_O$$

Since the wind loading on the blades is constantly varying, the maximum wind force had to be obtained through aerodynamic analysis. Using Excel, the resultant blade wind loads were computed at a variety of azimuth angles (θ). The highest resultant force (F_R) was approximately 60,000 N at an azimuth angle of 32.5 degrees.

From basic solid mechanics, the shear stress at the base of the structure is:

$$\sigma_{shear} = \frac{V}{A}$$

Where,

$$V = WH + F_R$$

A – cross-sectional area of the tower (m^2)

4.5.3 Tower – Bending

As with the shear stresses, the bending stresses are caused by wind loads on the tower and the resultant wind loads on the blades.

The tower was modeled as a vertical beam with a fixed support at one end. Therefore, from solid mechanics the maximum bending moment at the base of the structure is:

$$M_{max} = \frac{WH^2}{2} + F_R d$$

Where,

W – distributed wind load on the tower (N/m)

H – height of the tower (m)

F_R – resultant blade loading (N)

d – distance from the ground to where F_R acts (m)

The maximum bending moment was then used to calculate the bending stress using the following equation:

$$\sigma_{bending} = \frac{M_{max}(\frac{D_o}{2})}{I}$$

Where,

$$I = \pi\left(D_o^4 - D_i^4\right)/64$$

*M_{max} – maximum bending moment (N*m)*

$D_o/2$ – distance from the neutral axis to the point where stress is being calculated (m)

4.5.4 Tower – Von Mises Stresses

The Von Mises yield criterion was applied to the bending, axial, and shear stresses to determine the Von Mises principal stresses. The maximum Von Mises stress will be used to determine if the selected material and dimensions are valid. If the maximum Von Mises stress is lower than the yield strength of the material, and give an appropriate safety factor, the dimensions and material are valid. Conversely, if the maximum Von Mises stress exceeds the yield strength of the material, then larger dimensions and/or an alternate material are required. The Von Mises stresses were computed in Excel using the following formula:

$$\sigma_{von-mises} = \left|\frac{(\sigma_{bending} + \sigma_{axial})}{2}\right| + \sqrt{\left[\frac{(\sigma_{bending} + \sigma_{axial})}{2}\right]^2 + [\sigma_{shear}]^2}$$

4.5.5 Tower – Results

The maximum Von Mises stress was determined to be 25 MPa, assuming an inner diameter of 2 meters, a thickness of 0.0254 meters (1 inch), and A35 structural steel as the material. The inner diameter should be at least 2 meters to allow for access to the interior components (drive shaft, bearings, etc.) for maintenance purposes. Since the yield strength of A35 steel is 250 MPa, the safety factor of the structure is 10. Therefore, the dimensions and material are appropriate for this stage of the design.

5 ENVIRONMENT

Based on the feedback from phase one of the project MUN VAWT Design decided to investigate the impact of wind turbines on the avian population to assuage the fears raised by many people. A 2001 investigation into the causes of avian collision mortality in the United States estimated that wind turbines accounted for only 0.01 to 0.02% of the total fatalities. [8] More recently, a 2013 Canadian study estimated that the average number of avian deaths per turbine was 8.2 ± 1.4. However, it found this was heavily dependent on location has the average ranged from 0 to 26.9 deaths per turbine. [9] With this in mind, Nain, the location previously discussed as an area of interest was investigated for potential avian impacts on a proposed wind turbine project. By reading Environmental Assessment's for two different wind turbine project's proposed for Nain in the past it was found that Nain is designated an Important Bird Area (IBA). However this title carries no legal repercussions and Nain is therefore not a legally protected Ecological Reserve. An IBA is an area that meets the requirement set out by the IBA program for monitoring and designation of important areas for the conservation of birds. [10] It was also found that there are 8 Peregrine Falcon, which are considered a threatened species, nests in the surrounding area of Nain. Due to this a 1 km radius around the nest sites was proposed as a buffer zone. [11] The last consideration found was a condition on the release of the project from Governmental review mandating that an avian monitoring program be developed in line with Environment Canada regulations. [12]

6 PRELIMINARY ECONOMIC ANALYSIS

The cost estimates in this report are extrapolations from the economic analysis of Northwind 100, the 100kW wind turbine used by Nalcor in Ramea, Newfoundland, and the diesel generators in Nain, Labrador. [13] [14] Using engineering economic principles, the monetary values for maintenance costs, fuel consumption and capital costs were translated through time with an interest rate of 3.5% using the equation below:

$$F = P\,(1 + i)^n$$

Where F is the monetary or accumulated value at a future point in time, P is the current or principal value, i is the interest rate and n is the number of years or interest periods between the present and future.

The following findings are estimates only, since the actual cost of the VAWT has not been finalized, and because transportation costs for both diesel fuel and the VAWT have not yet been considered. In the final stage of this project there will be an accurate cost analysis for the VAWT based on quotes and cost estimates from venders and manufactures.

Table 7: Key economic estimates

Estimated Capital Cost	$500 000.00
Maintenance Cost/Year - Turbine	$10 000.00
Maintenance Cost/Year – Diesel Generator	$20 000.00
Fuel Cost for 2015	$3 630 967.00
Payoff Period for a Single Turbine	2 years
Payoff Period for an Installation of 5 Turbines	3 years
Payoff Period for an Installation of 20 Turbines	4 years
Payoff Period for an Installation of 50 Turbines	4 years

An initial installation of five turbines will produce sufficient power to reduce annual fuel costs by approximately $750 000. This corresponds to a saving of 15% of the total cost of power generation in the target location.

7 PROGRESS REPORT

With the closure of Phase II, it is important to highlight the areas where progress met, exceeded, or fell short of expectations. The majority of the goals were met including the preliminary structural design, aerodynamic modelling in Q-Blade and production of the preliminary CAD Model. The design group over-achieved in aerodynamic CFD modelling detailed mechanical design, and the economic analysis. However, the preliminary vibrational analysis and generator selection were pushed to Phase III in favour of a preliminary economic analysis. The selection of the induction generator was delayed due to lack of communication on the vender's behalf. The design group has contacted venders and are awaiting a reply.

Preliminary Structural Design	•On schedule
Aerodynamic Modelling - QBlade	•On schedule
Preliminary Vibrational Analysis	•Delayed •Priority given to economic analysis
Preliminary CAD Model	•On Schedule
Selection of Generator	•Delayed •Waiting on vendors

Figure 9: Status of Phase II tasks

Moving into the final phase of the project, the design group is on schedule for the structural finite element analysis and the detailed CAD model. As previously noted, the CFD modelling, detailed mechanical design and economic analysis is ahead of schedule, which will allow more focus to be placed on the vibrational analysis.

Structural FEA	•On schedule
Aerodynamic Modelling - CFD	•Ahead of schedule
Detailed Vibrational Analysis	•Delayed •Priority given to economic analysis
Detailed Mechanical Design	•Ahead of schedule
Detailed CAD Model	•On schedule
Economic Analysis	•Ahead of schedule

Figure 10: Status of Phase III tasks

The project is projected to proceed and be completed on schedule. The Gantt chart can be found in Appendix C.

8 CONCLUSION

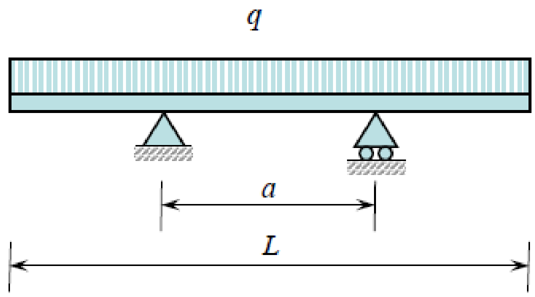
During Phase II of the project, aerodynamic and structural analysis was carried out on the desired turbine design. These analyses concluded that the turbine will be able to produce the target power output, and that the structural and mechanical design of a VAWT of this size that complies to the relevant standard is feasible using conventional materials. Based on the parameters computed from aerodynamic modelling, a preliminary economic analysis was conducted, which shows that a turbine of this design will be economically viable.

APPENDIX A: ABBREVIATIONS USED IN IEC 61400-1

The following abbreviations are used in Table 2:

DLC	Design load case
ECD	Extreme coherent gust with direction change
EDC	Extreme direction change
EOG	Extreme operating gust
EWM	Extreme wind speed model
EWS	Extreme wind shear
NTM	Normal turbulence model
ETM	Extreme turbulence model
NWP	Normal wind profile model
Vr+_2	Sensitivity to all wind speeds in the range shall be analysed
F	Fatigue
U	Ultimate strength
N	Normal
A	Abnormal
T	Transport and erection
*	Partial safety for fatigue

APPENDIX B: AIRFOIL BENDING MOMENT CALCULATIONS



By inspection, the maximum bending moments will occur at the supports and at the mid-span.

Moment at Supports:

$$M_{sup} = \frac{q}{8} L^2 \left(1 - \frac{a}{L}\right)^2$$

Moment at Mid-span:

$$M_{mid} = \frac{q}{4} L^2 \left(\frac{a}{L} - \frac{1}{2}\right)$$

Equating the two:

$$\frac{q}{8} L^2 \left(1 - \frac{a}{L}\right)^2 = \frac{q}{4} L^2 \left(\frac{a}{L} - \frac{1}{2}\right)$$

Negating the coefficients q and L and expanding:

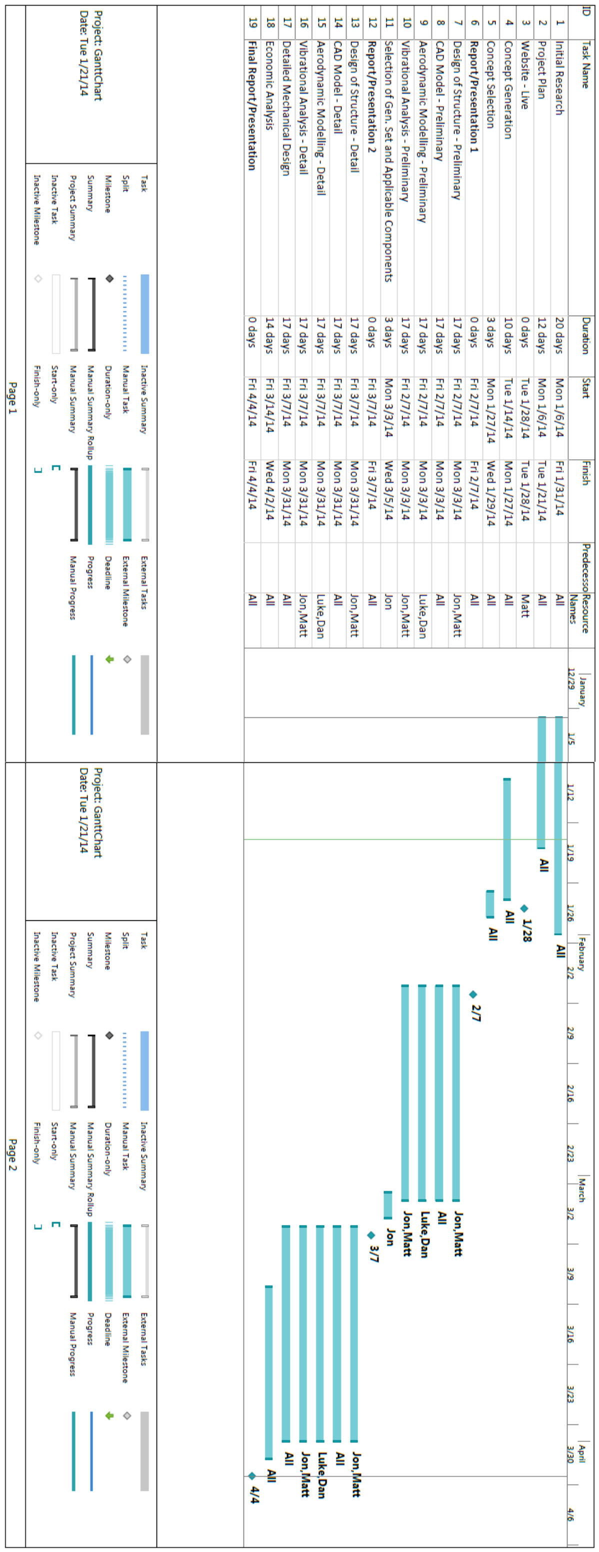
$$\left(\frac{1}{2} - \frac{a}{L} + \frac{a^2}{2L^2}\right) = \left(\frac{a}{L} - \frac{1}{2}\right)$$

$$\left(1 - 2\frac{a}{L} + \frac{a^2}{2L^2}\right) = 0$$

$$\frac{a}{L} = 0.585786 \; or \; \frac{a}{L} = 3.41421$$

Since the supports cannot be spaced farther apart than the blade is long, $\frac{a}{L} = 0.586$ is the optimal support spacing. Either bending moment formula can then be used to calculate maximum bending moment.

APPENDIX C: GANTT CHART



ID	Task Name	Duration	Start	Finish	Predecessors	Resource Names
1	Initial Research	20 days	Mon 1/6/14	Fri 1/31/14		All
2	Project Plan	12 days	Mon 1/6/14	Tue 1/21/14		All
3	Website - Live	0 days	Tue 1/28/14	Tue 1/28/14		Matt
4	Concept Generation	10 days	Tue 1/14/14	Mon 1/27/14		All
5	Concept Selection	3 days	Mon 1/27/14	Wed 1/29/14		All
6	**Report/Presentation 1**	0 days	Fri 2/7/14	Fri 2/7/14		All
7	Design of Structure - Preliminary	17 days	Fri 2/7/14	Mon 3/3/14		Jon,Matt
8	CAD Model - Preliminary	17 days	Fri 2/7/14	Mon 3/3/14		All
9	Aerodynamic Modelling - Preliminary	17 days	Fri 2/7/14	Mon 3/3/14		Luke,Dan
10	Vibrational Analysis - Preliminary	17 days	Fri 2/7/14	Mon 3/3/14		Jon,Matt
11	Selection of Gen. Set and Applicable Components	3 days	Mon 3/3/14	Wed 3/5/14		Jon
12	**Report/Presentation 2**	0 days	Fri 3/7/14	Fri 3/7/14		All
13	Design of Structure - Detail	17 days	Fri 3/7/14	Mon 3/31/14		Jon,Matt
14	CAD Model - Detail	17 days	Fri 3/7/14	Mon 3/31/14		All
15	Aerodynamic Modelling - Detail	17 days	Fri 3/7/14	Mon 3/31/14		Luke,Dan
16	Vibrational Analysis - Detail	17 days	Fri 3/7/14	Mon 3/31/14		Jon,Matt
17	Detailed Mechanical Design	17 days	Fri 3/7/14	Mon 3/31/14		All
18	Economic Analysis	14 days	Fri 3/14/14	Wed 4/2/14		All
19	**Final Report/Presentation**	0 days	Fri 4/4/14	Fri 4/4/14		All

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