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DESIGN AND CONSTRUCTION OF A 500W WIND GENERATOR USING THIN SHEET METAL TURBINE BLADES

Auwal Ibrahim[1], B. I. Kunya[1], Abdullahi Ahmad[1], Kayode P. Dare[2], Sani Isah[3], Ahonye O. Job[4], S. T. Auwal[1], M. S. Dambatta[1], A. Y. Gidado[1] and N. Muaz[1]

[1]*Department of Mechanical Engineering, Kano University of Science and Technology Wudil, 3244 Kano, Nigeria*

[2]*Nexans Nocaco Company, No. 12, Maichibi Road Kakuri Industrial Estate, Kaduna, Nigeria*

[3]*Mechanical Engineering Section, Yobe State Ministry of Works, Geidam/Gashua Road, Damaturu, Nigeria*

[4]*Jobest Water Solutions, No. 76 Kaledi Street, Uke Bus Stop, Nasarawa State, Nigeria*

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ABSTRACT

Generally small wind turbines have a big role to play in rural and remote areas, especially for domestic application in areas where connection to the grid is not available. This paper study the average wind speed available in Kano Nigeria, to design and construct a 500 watts wind generator using thin sheet metal blades, for horizontal-axis wind turbine. The machine is designed to work at 12m height, and therefore the research determines the material selected for the blades, and calculated the required height of the blade to drive the selected DC generator incorporated in the wind generator. The research work also designed the tail vain for direction control, the driving shaft and other components attached to it such as; the shaft key, the hub, the bearings and the gear box. This research work is meant to be used on top of the house roofing gutter or wall pillars; it is intended to encourage the use of available renewable energy sources for domestic applications.

Corresponding Author: auwalwdl@gmail.com

1. Introduction

Wind turbine blades are made in air foil shape, designed to harness the wind energy and drive the rotor of a wind turbine. The air foil shape is designed to provide the lift when the kinetic energy of wind perpendicularly strikes the blades. Wind energy has been used for similar purposes for thousands of years to propel boats and ships, and to provide rotary windmill power to reduce the physical burden of man. The wind turbine is an intervention that dates from the earliest time of an intervention history.

The first written evidence for the use of wind turbines are that of Hero of Alexandria who in the third or second century BC described as given power to an organ, but it has been discussed whether it was of any practical use apart from being a kind of toy [1]. In another research conducted in 20's, by German professor Albert Bertz of the German Aerodynamics Research Centre in Gottingen, who made some path-breaking theoretical studies on wind turbines, he calculated that, "no wind turbine (even if 100% efficient) could convert more than 59.3% of the kinetic energy of the wind in to mechanical energy turning a rotor" [2]. This is known as the Bertz limit, and is the theoretical maximum coefficient of power for any wind turbine. However, practically good wind turbines generally convert 35-45% of the available wind energy to electricity [4]. The amount of energy generated by a wind turbine is dependent upon the height of a tower to which the wind turbine is installed.

2. Methodology

This research work considers two types of forces acting on the rotor blades of a propeller-type wind turbine. They are circumferential forces in the direction of wheel rotation that provide the torque, and the axial forces in the direction of the wind stream that provide an axial thrust that must be counteracted by proper mechanical design.

2.1 Data Collection of Average Wind Speed in Kano Nigeria

This wind speed data is obtained from the Nigeria Metrological Agency (NIMET) at Aminu Kano Intentional Airport, Kano, as shown in table below;

Table 2.1 Wind Speed Data (m/s)

Yrs/Mth	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
2006	7.5	7.5	7.5	7.7	5.1	7.7	5.1	7.7	7.2	7.5	7.7	8.0
2007	5.3	6.9	6.9	5.1	8.0	7.7	7.7	7.5	6.7	6.7	5.1	8.0
2008	5.3	5.0	5.3	4.6	5.3	5.1	5.0	4.6	7.2	6.9	5.1	7.5
2009	7.7	6.9	7.7	7.7	8.0	5.1	8.0	5.1	7.2	8.0	7.7	8.0
2010	8.0	7.2	5.3	7.5	8.0	7.7	5.3	5.1	6.9	×	×	×
Average	6.8	6.7	6.5	6.5	6.9	6.7	6.2	6.0	7.0	7.3	6.4	7.9

Source: Nigerian Metrological Agency (NIMET) [6]
Where; x = wind speed not available

2.2 Selection of Blades Material

When a stream of air strikes the blades, the air loses its kinetic energy at the surface of the blades and the pressure increases at the surface of the blades, thus rotate the shaft connected. Materials of the blade includes aluminium which is very expensive, polyvinyl chloride (PVC) although it is lightweight and flexible, but PVC quickly degrades when subjected to direct sun light, another material of the blade is wood, but its greatest disadvantage is, it needs sophisticated lamination that perfectly protects the wood from rain and attack by insects, this treatment invites high cost, and only treatment of experts in that field should be reliable. Thin steel sheet is another alternative for materials of blades, it is cheaper, easy to shape and strong enough especially for small wind turbines mean for home application, it does not require any non-engineering treatment. Hence this research chooses thin steel sheet metal for material of blade.

2.3 Design Considerations

The design of any engineering component begins as a mere imagination which then transforms into design before its actualization. In carrying out the design, certain criteria have to be considered, and a number of factors also have to be taken into consideration. These include the following [7];

i. Availability of materials;
ii. Mechanical properties e.g. Strength, toughness, rigidity etc.;
iii. Machineability and Formability;
iv. Corrosion resistance of materials; and
v. Cost of Production.

Although there are so many factors that play a role in designing a blade for a wind turbine, but the most important one is the aerodynamics, which causes the blades to lift. Modern wind turbines mainly operate on three blades, and it's the most widely used standard. This is because of the unbalances in using one blade, two blades would offer greater yield but could results in wobbling, which would further create another stability issue that may cause wear to the machine components. While any number of turbine blades, that are greater than three would also create a wind resistance. Therefore, this paper selected three blades for the design of this research work.

3. Design Analysis

Average velocity of wind in kano Nigeria is given by;

$$\sum_{i=1}^{n} \frac{V_i}{n} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(3.1)$$

$$V_{av} = \frac{6.8+6.7+6.5+6.5+6.9+6.7+6.2+6.0+7.0+7.3+6.4+7.9}{12}$$

= 6.74m/s (at 10m height)

But this research work is designing for 12m height. Therefore, to obtain the wind speed at 12m height; Wind speed at a given height can be obtained from the wind speed of the reference height using the relation given by Rai (2011):

$$\frac{V_2}{V_1} = \left(\frac{H_2}{H_1}\right)^{\alpha} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.(3.2)\ [3]$$

Where; $\alpha = \alpha_0 \left(1 - \frac{LogV_r}{LogV_0}\right)$

Where also; V_r = reference velocity = 6.74m/s at 10m height.

V_0 = a fixed velocity = 67.1m/s

$$\alpha_0 = \left(\frac{Z_0}{H_r}\right)^{0.2}$$

in which; Z_0 = Surface roughness length, varying widely but taken as 0.4m for evaluation purposes.

H_r = Reference height (i.e. 10m)

$$\therefore \alpha_0 = \left(\frac{0.4}{10}\right)^{0.2} = 0.525$$

$$\alpha = 0.525\left(1 - \frac{Log6.74}{Log67.1}\right) = 0.286 = \frac{2}{7}$$

$$\therefore V_2 = 6.74\left(\frac{12}{10}\right)^{2/7} = 6.92m/s$$

∴ Average wind speed = 6.92m/s

3.1 Rated Wind Speed

To get the rated wind speed, we use the equation of maximum extractable power [2].

$$P_{rated} = \frac{1}{2}\rho A V^3 \eta \times \frac{8}{27} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.3)$$

Where; A = $\frac{\pi D^2}{4} = \frac{\pi \times 3^2}{4} = 7.069m^2$

$\eta = 30\%$ (since modern wind turbines has an efficiency between 30 to 40%)

p = rated power = 770W (since we are looking for a rated speed)

$$\therefore V = \sqrt[3]{\frac{2\times27\times770}{1.201\times7.069\times0.3\times8}} = 12.68m/s$$

∴ Rated wind speed = 12.68m/s

3.2 Rotor Diameter

The theoretical power in the wind is given by [6];

$$P = \frac{1}{2}\rho A V^3\ (Watt) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(3.4)$$

Where; ρ = density of air (1.201 kg/m^3 at NTP)

V = mean air velocity (m/s)

A = swept area of blades (m^2)

It will shortly be apparent that the total power in the wind cannot all be converted to mechanical power.
Therefore, maximum extractable power is given by the equation;

$$P_{max} = \frac{8}{27}\rho A V^3 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.(3.5)$$

Where; A $= \frac{\pi D^2}{4} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3.6)$

In which D is the rotor diameter of the wind turbine.

To achieve a power of 500Watt by a generator working on reasonable efficiency of 75%, and by adopting a factor of safety of 1.5 as for some machine design [8], The power compute as;

$$P = \frac{1}{2}\rho V^3 \times \frac{\pi D^2}{4} \times \frac{8}{27} \times 1.5 \times 0.75 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots. (3.7)$$

Where V; is the average wind velocity = 6.92m/s

$$\therefore\ 500 = \frac{1}{2} \times 1.201 \times 6.92^3 \times \frac{\pi D^2}{4} \times \frac{8}{27} \times 1.5 \times 0.75$$

$$500 = 52.1D^2$$

$$D = \sqrt{\frac{500}{52.1}} = \sqrt{9.597} = 3.098m$$

∴ Rotor diameter = 3m = 300cm

3.3 Length (height) of the Blades

In determining the height of the blades of a wind turbine, the most important parameter supposes to know is the rotor diameter of the blades, and has been designed above to be equal to 3m.

i.e. Rotor diameter = 3m = 300cm

Hence; Radius of the Rotor = $\frac{3m}{2} = 1.5m = 150cm$

3.4 Rotational Speed of Blades

The rotational speed of a blade of wind turbine is indicating as N. But tip speed ratio for Dutch four arm type started from 2, and that of high speed two blades type started from 4, and it's given by [5];

$$\text{Tip speed ratio} = \frac{speed\ of\ rotation\ of\ blade\ tip}{wind\ speed}$$

i.e. $V_{TSR} = \frac{2\pi RN}{V_w} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(3.8)$

Adopting a minimum tip speed ratio of 2 for this three blades design

$$\therefore\ N = \frac{\mathrm{V_w \times V_{TSR}}}{2\pi \mathrm{R}} = \frac{6.92 \times 2}{2 \times \pi \times 1.5} = 1.5\ \text{rev/sec}$$

$\therefore speed\ of\ rotation\ (N)$ = 90 rpm

3.5 The circumferential force or Torque on blades

$$T = \frac{p}{\omega} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.9)\ [5]$$

Where; P = power on blades

$= \frac{1}{2} \times 1.201 \times \frac{\pi}{4} \times 3 \times 6.92^3 = 468.86W$

ω = Angular Speed of blade = 4.61rad/sec

∴ Torque = $\frac{468.86}{4.61} = 101.7Nm$

3.6 Axial force on a turbine wheel (f_p)

$$f_p = \frac{4}{9}\rho A V_f^2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots. (3.10)[3]$$

$$f_p = \frac{4}{9} \times 1.201 \times \frac{\pi}{4} \times 3^2 \times 6.92^2$$

$$\therefore \text{Axial force} = 180.68N$$

3.7 Hub Design

The hub of a wind turbine is designed in an aerodynamic shape for a good mechanical designed as it always pointed toward the direction of the wind stream where axial thrust react with the rotor, and which must be counteracted by a proper mechanical design.

This paper designed hub in two parts, an aerodynamic shaped thin curve of negligible weight and a cylinder that will be attached to the shaft on which the hub and blades were attached to. For uniformity, the cylinder diameter is designed to be the same as gear wheel diameter (calculated in gear ratio).

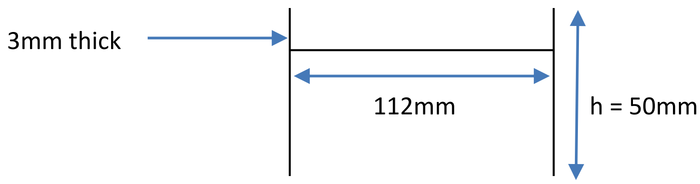


Fig. 3.1 Rotor Hub Base Cylinder

$$\text{Surface area} = \frac{\pi d^2}{4} h \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.11)$$

$$= \pi \times \frac{0.112^2}{4} \times 0.05 = 4.9\text{x}10^{-4}\ \text{m}^2$$

3.7.1 Hub Cylinder volume

$$\text{Volume = area x Thickness} = 4.9\text{x}10^{-4} \text{ x } 0.003 = 1.5 \times 10^{-6} m^3$$

3.7.2 Hub Cylinder Weight

Weight = Mass x gravity = (Volume x density) x gravity

$= 1.5\text{x}10^{-6}$ x 7861 x 9.81 = 0.116N

3.8 Shaft Design

The shaft is a part that connects the turbine wheel with the gear box, and to design the shaft, there are some factors to be considered.

i. Material must have high strength.
ii. Good machineability
iii. Low heat treatment properties.
iv. High wear resistance
v. Available at affordable cost.

However, to design a shaft of a wind turbine, various forces acting on a shaft has to be considered in determining the diameter of the shaft, these include axial load, bending moments and tensional load.

3.8.1 Forces Acting on the Shaft

i. **Axial force:** Since shaft is directly attached to rotor, this implies that the same axial force on the rotor acts on the shaft, and is already calculated in blade design section.

∴ Axial force = 180.68N

ii. **Torque:** It is also the same case as above, since rotor rotate together with the shaft attached to it, and therefore they have the same torque.

∴ Torque = 101.7Nm

iii. **Weight of the rotor:** Act on the shaft downward. This design adopts a blade of 140mm width at the inner and 100mm width at the outer end, and its $\frac{3}{2}$ m long, that is 1500mm long, and is 1mm thick.

$$\text{Area of blade} = \frac{1}{2}(a+b)h = \frac{1}{2}(0.14 + 0.10) \times 1.5 = 0.18 m^2$$

Volume of blade = Area x thickness = 0.18 x 0.001 = 0.00018m^3

Mass of the blade = density x Volume

= $\rho \times V$ (but ρ = 7861 Kg/m^3 for steel)

= 7861 x 0.00018 = 1.415Kg

⇒ Weight of one blade = mass x gravity = m x g

= 1.415 x 9.81 = 13.88N

⇒ Weight of 3 blades = 3 x 13.88 = 41.64N

∴ **Total weight of the rotor** = weight of blades + weight of the hub

= 41.64 + 0.116 = 41.756N

iv. **Weight of the Gear:** = surface area of the gear x thickness x Density of the material x gravity

$$\text{But gear ratio} = \frac{T_g}{T_p} = \frac{D_g}{D_p} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.12)$$

Where; D_g = pitch cycle diameter of gear (Driver)

$$\therefore D_g = \frac{T_g \times D_p}{T_p} = \frac{67 \times 30}{18} = 112mm$$

$$\therefore \text{Weight of gear} = \pi \frac{D_g^2}{4} \times t \times \rho \times g \ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.13)$$

$$= \pi \times \frac{0.112^2}{4} \times 0.012 \times 7861 \times 9.81 = 9.1\text{N}$$

3.8.2 Shear Force on the Shaft

The shearing force is the algebraic sum of all forces to one (either) side of the section. Adopting a standard shaft length of 370mm, and having the attachments in positions as in figure below:

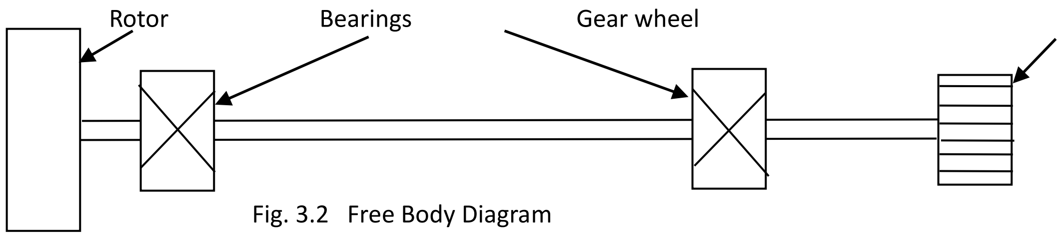
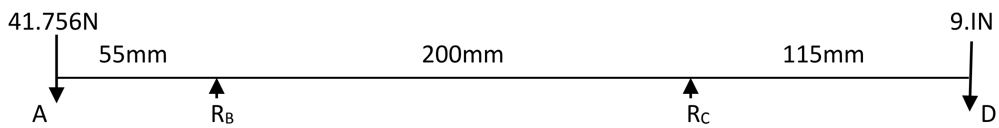


Fig. 3.2 Free Body Diagram



Taking moment about point B

$$\sum M_B = 0 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.14)$$

$$\therefore (-41.756 \times 0.055) + (-R_C \times 0.2) + (9.1 \times 0.315) \text{ -2.2966 - } 0.2R_C + 2.8665 = 0$$

$$\Rightarrow -0.2R_C = 2.2966 - 2.8665$$

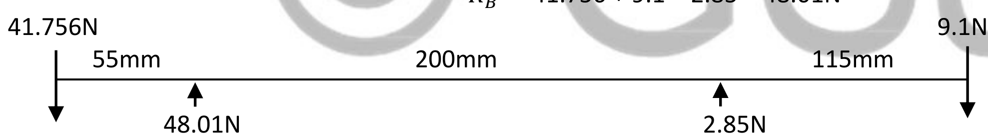
$$\therefore R_C = \frac{-0.5699}{-0.2} = 2.85\text{N}$$

Then for equilibrium; Sum of vertical upward and downward forces = 0

$$\sum f_y = 0 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.. (3.15)$$

$$\therefore -41.756 + R_B + 2.85 - 9.1 = 0$$

$$R_B = 41.756 + 9.1 - 2.85 = 48.01\text{N}$$



∴ 0 < x < 55mm	SF_A = -41.756N
55mm < x < 255mm	SF_B = -41.756 + 48.01 = 6.254N
255mm < x < 370mm	SF_C = 6.254 + 2.85 = 9.1N
x = 370mm	SF_D = 9.1- 9.1 = 0

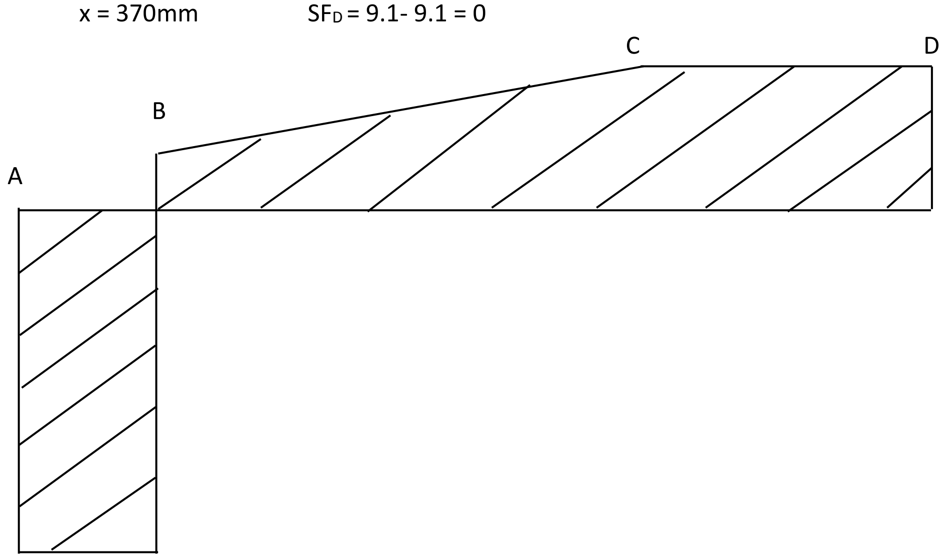
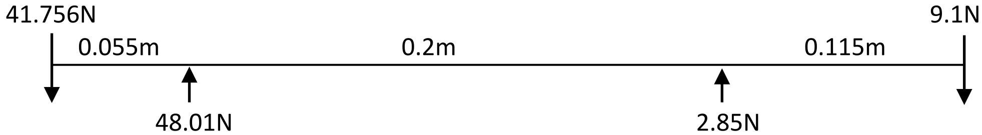


Fig.3.3 Shear Force Diagram

3.8.3 Bending Moment on the Shaft

The bending moment is the algebraic sum of all the moment of forces and concentrated moments to one side of the section.



Taking moment about point D

$$\therefore \sum M_D = 0 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.16)$$

0 < x < 0.055m	$BM_A = -41.756x$
If x = 0	$BM_A = 0$
If x = 0.055m	$BM_A = -2.297$Nm
0.055m < x < 0.255m	$BM_B = -41.756x + 48.01(x-0.055)$
If x = 0.055m	$BM_B = -2.297$Nm
If x = 0.255m	$BM_B = -1.046$Nm
0.255 < x < 0.37m	$BM_C = -41.756x + 48.01(x-0.255)$
If x = 0.255m	$BM_C = -1.046$Nm
If x = 0.375m	$BM_D = 0$

Fig. 3.4 Bending Moment Diagram

Hence, the highest bending moment occurred at B, with 2.297Nm

3.8.4 Shaft diameter

When shaft is subjected to combine loads, its size can be determined using maximum shear stress theory given by the equation:

$$\tau_{max} = \frac{16}{\pi\, d^3}\sqrt{(K_m M^2 + K_T\, T^2)} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.17)$$

$$d = \left(\frac{16}{\pi\, \tau_{max}}\sqrt{(K_m M^2 + K_T\, T^2)}\right)^{\frac{1}{3}} \ldots\ldots\ldots\ldots\ldots\ldots (3.18)$$

Where; $\tau_{max} = \tau_p$ = 42MPa for commercial shaft without specification and with key way.

K_m = 1.5 for gradually applied load, and wind increases gradually

K_T = 1.0 for the same reason as Km

T = toque on blades = 101.7Nm

M_B = bending moment = resultant at B =2.297Nm.

$$\therefore d = \left(\frac{16}{\pi \times 42 \times 10^6}\sqrt{(1.5 \times 2.297^2 + 1 \times 101.7^2)}\right)^{1/3} = 0.023\text{m} = 23\text{mm}$$

But to coincide with the standard shaft, this design adopted to select 25mm diameter from the standard available shaft in the markets. ∴ Shaft diameter = 25mm

For a small torque apply on this shaft in transmission a small power, this paper uses a plain medium carbon steel for shaft material that is meant for general purpose shafts.

3.8.5 Design of Key

Square key is selected for this shaft design, as it can be used for general purpose. For the dimension of key;

$$b = h = \frac{d}{4} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.19)$$

where; d = shaft diameter

b = width of the key
h = height of the key

$$\therefore b = h = \frac{25}{4} = 6.25mm$$

3.9 Design of Bearings

Though bearings may be classified in many ways, yet "sliding contact bearing and rolling contact bearings" are the classes of bearings depending upon the nature of contact, and are very important bearing from the point of view of this paper.

Rolling contact radial bearing is the best type of bearing fit for the application of this paper, as the load carried by shaft acts perpendicular to the direction of motion of the moving elements of the bearings, which means radial bearings are best for this load. While the light weight on the shaft rotate continuously, as far as the wind blows, which means rolling contact bearings also fit for this kind of application, and for its low starting friction, the selection of rolling contact radial bearing becomes inevitable.

3.9.1 Internal (Bore) Diameter of the Bearings

The internal ("i.e. bore) diameter of the front and rear bearings does not need any further analysis, rather than to use a bearing of the same bore diameter as the shaft diameter, as its designed to fit with the bearings.

Since the shaft is designed to be 25mm diameter

∴ Bearings bore diameter = 25mm

3.9.2 Outside Diameter and Width of the Bearings

From "Appendix 1" on Rolling contact bearings, of Principal dimensions for radial ball bearings. For 25mm Bore diameter bearing, the standard outside diameter = 52m. The standard bearing width = 15mm.

3.9.3 Rating Life of the Bearing in Revolution

Is given by $L = \left(\frac{C}{W}\right)^k \times 10^6 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots. (3.20)$

Where; W = load on the bearing = 525N
C = basic dynamic capacity rating
= 11kN (for 25mm bore diameter radial ball bearing)
k = 3 for ball bearing

$$\therefore L = \left(\frac{11000}{525}\right)^3 \times 10^6 = 9.198 \times 10^9$$

$$\therefore Rating\ life\ of\ the\ bearing = 9198 \times 10^6 rev$$

3.10 Design of Gears

The type of gear that is best to use for this design is a spur gear, because the shaft of the rotor and that of the generator are parallel to each other, and in spur gearing, the gears have teeth parallel to the axis of the wheel. A pair of spur gears is kinematically equivalent to a pair of cylindrical discs, keyed to a parallel shaft having line contact, therefore spur gear for external gearing will be used for this research work. The nacelle sits on top of the short tower to be on top of a building, contain the engine components of the horizontal-axis wind turbine.

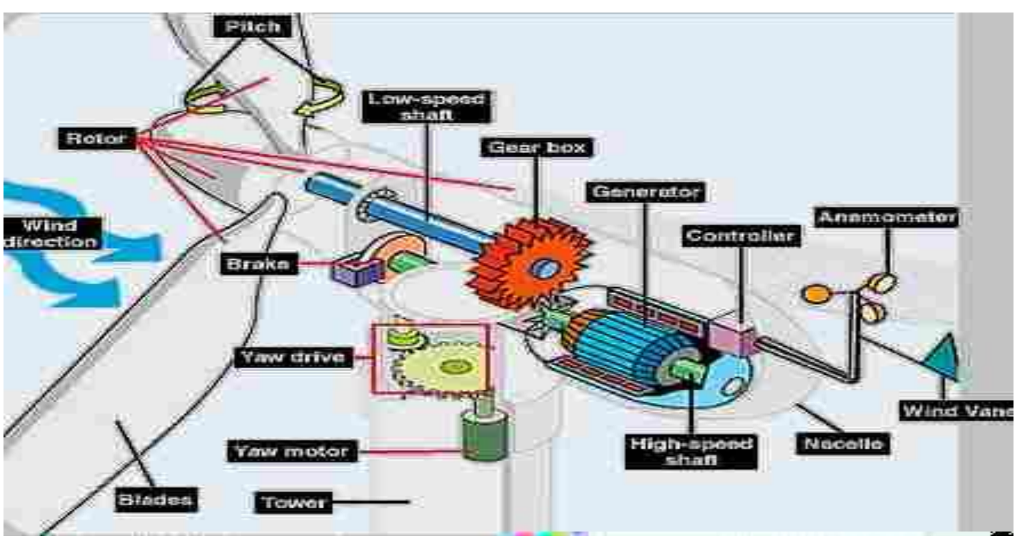


Fig.3.5 Nacelle layout and components (showing rotor and blades (in part) on the left, electrical generator on the right, and yaw drive at the bottom)

3.10.1 System of Gear Teeth Selection

The Commonly used system of gear teeth are 14½^0 composite system which, has no interchangeability, 14½^0 full depth involutes system which is mainly for use with gear hubs, 20^0 stub involutes system which is purposely for taking heavy loads, and the 20^0 full depth involutes system, which has stronger tooth for their increase in pressure angle from 14½^0 to 20^0. Hence this type (20^0 full depth involutes system) will be used for this paper.

3.10.2 Teeth on pinion (driven)

From "Appendix 2" on standard gears, for minimum number of teeth on pinion, the standard minimum number of teeth on the pinion in order to avoid interference is given;

For 20^0 full depths in involute is 18, ∴ Number of teeth on pinion = 18

3.10.3 Teeth on Gear Wheel (Driver)

Some small wind turbines are even design on direct connection to rotor shaft (i.e. no gear box incorporated), but this design incorporate a gear box to boost the rotational speed to a velocity ratio ≥ 3.5 to improve the performance of the machine. Thus; Number of teeth on driver = teeth on pinion x gear ratio, ∴ Number of teeth on driver = 18 x 3.5 = 63

But considering the selection from standard available commercial gears that has no 63, but 67 as the most closer range available for the same class of pinion;

∴ Number of teeth on driver = 67

Then Gear ratio Is given by; $\frac{Teeth\ on\ Driver}{Teeth\ on\ Driven} = \frac{T_g}{T_p} = \frac{67}{18} = 3.72$

3.10.4 Torque acting on pinion

$$\text{Torque (T)} = \frac{\text{P} \times 60}{2\pi\text{N}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(3.21)$$

Where; P = 468.86W
N = 90 x 3.72

$$\text{T} = \frac{468.86 \times 60}{2 \times \pi \times 90 \times 3.72} = 13.37\text{Nm}$$

∴ Torque on pinion = 13.37 Nm

Note**;** since gear wheel (driver) is directly attached to the rotor shaft, therefore forces acting on it, are the same as that act on the rotor.

3.10.5 Tangential Load on Pinion

$$W_T = \frac{\text{T}}{\text{D}_\text{P}/2} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(3.22)$$

Where; D_P = pitch circle diameter of pinion
= 30mm (for 18 teeth 20^0 full depth involutes)
T = torque = 13.37Nm

$$\therefore Load\ on\ pinion\ (W_T) = \frac{2 \times 13.37}{0.03} = 891.33\text{N}$$

3.10.6 Normal load on the tooth

$$W_N = \frac{W_T}{\cos\emptyset} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(3.23)$$

Where Ø is the pressure angle = 20^0

$$\therefore W_N = \frac{891.33}{\cos 20} = 2183N$$

3.11 Design of Tail Vain for Direction Control

The wind machine is design to rotate over a tower in the direction of wind flow to machine, which is achieve automatically by the help of a tail vain.

The tail aligns the rotor blade to face the direction of wind velocity. A tail vain is design to certify the condition given by the inequality;

$$\text{A} > \frac{D^2}{40} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.24)$$

Where; A = surface area of a tail
D = rotor diameter

∴ For this paper, equation 3.27 yield; $\frac{D^2}{40} = \frac{3^2}{40} = 0.225m^2$

This paper selects a sector shape in designing a tail, and with dimension as in figure below;

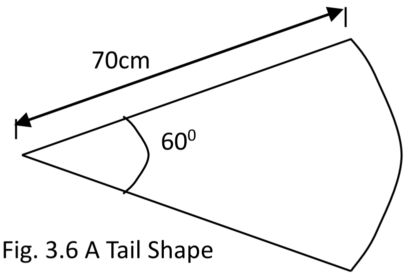


Fig. 3.6 A Tail Shape

$$\therefore \text{surface area of tail} = \frac{\theta}{360} \times \pi r^2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.25)$$

$$= \frac{60}{360} \times \pi \times 0.7^2 = 0.68m^2$$

Since $0.68m^2 > 0.225m^2$; hence the condition of tail design certified

3.12 Selection of Tower

The length of the tower on top of the 8m roofing gutter or wall pillar is selected to be 4m (making a total of 12m height), to allow the rotor blades rotate freely on top of the tower base. And since most of the forces acting on tower will be absorbed by the base building due to the short tower carrying the machine, a steel pipe of considerable size and thickness will used for this construction.

4. Results

Table 4.1 Calculated results

Component	**Calculated value**
Average wind speed at the reference height	6.74 m/s
Average wind speed at the design height	6.92 m/s
Rated wind speed	12.68 m/s
Number of blades in the rotor	3
Rotor diameter	300cm
Blade height	150 cm
Factor of Safety	1.5
Rotational speed	90rpm
Torque on blades	101.7Nm
Axial force on turbine	180.68N
Shaft Diameter	25mm
Gear ratio	3.72

Table 2 above shows some important calculated results for this design. However, components to be selected from the standard available components in the markets, were adopted by the higher minimum size available compared to the calculated size.

4.1 Test Results

Table 4.2 The Test Results

Power (W)	114	142	177	208	232	264	283	301	331	368
Wind Speed (m/s)	2.05	2.31	2.50	2.78	3.02	3.36	3.81	3.97	4.22	4.50

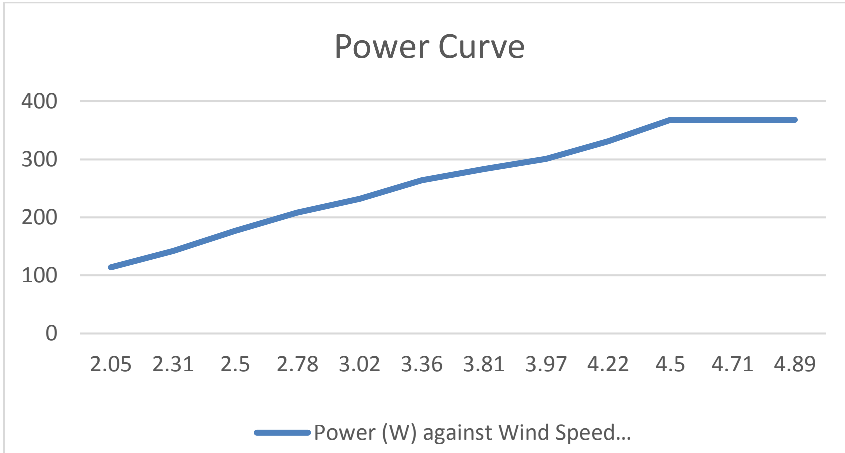


Fig. 4.1 Power Curve of the Machine

It can be seen from the graph above, the machine starts to generate power at about 2m/s, and power generated increases with increase in wind speed until it reaches 4.5m/s when the machine seize to yield more increase in power generated. However, the machine may produce more increase in the output with more increase in powerful wind speed, since the design wind speed limit could not be reach at the test point. Also, the swarm of points along the power curve, is as a result of the fluctuations of wind speed as it always happens in nature.

4.2 Construction Work

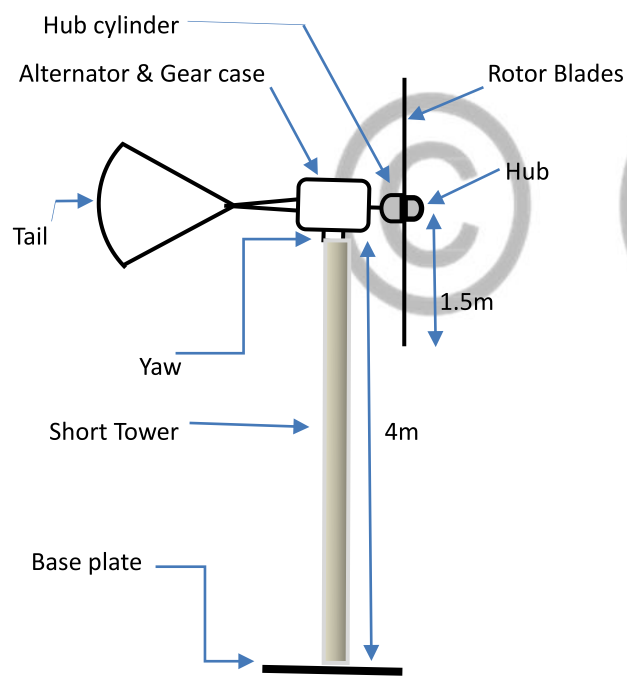


Fig. 4.2 Assembly Drawing of the Wind Power Machine

Plate 4.3 Picture of the Constructed Machine

5. Conclusion

The design obtained a rotor of three blades for horizontal axis wind turbine, with a length of 1.5m, on a 12m height tower, and with a circumferential force of 101.7Nm. The construction was done using the available materials in Kano markets. The machine was tested and functioned, an average wind speed of 4.50m/s generated an average power of 368W, which amount to 73.6% efficiency.

5.1 Recommendations

More research should be encouraged, and some of the possible improvement on this research work includes;

i. The material of blades could be improved toward lighter weight.
ii. The height of the tower could be increase above 12m, as this will increase the efficiency of the machine.

Acknowledgment

I wish to thank all those that have contributed in one way or the other for the success of this reseach work.

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Appendices

Appendix 1. Principal Dimensions for Radial Ball Bearings

Bearing no.	Bore (mm)	Outside diameter	Width (mm)
200 300	10	30 35	9 11
201 301	12	32 37	10 12
202 302	15	35 42	11 13
203 303 403	17	40 47 62	12 14 17
204 304 404	20	47 52 72	14 14 19
205 305 405	25	52 62 80	15 17 21
206 306 406	30	62 72 90	16 19 23
207 307 407	35	72 80 100	17 21 25
208 308 408	40	80 90 110	18 23 27
209 309 409	45	85 100 120	19 25 29
210 310 410	50	90 110 130	20 27 31
211 311 411	55	100 120 140	21 29 33
212 312 412	60	110 130 150	22 31 35

213		120	23
313	65	140	33
413		160	37
214		125	24
314	70	150	35
414		180	42
215		130	25
315	75	160	37
415		190	45
216		140	26
316	80	170	39
416		200	48
217		150	28
317	85	180	41
417		210	52
218		160	30
318	90	190	43
418		225	54

Source: Kurmi R. S and Gupta J. K (2005). A Text Book of Machine Design

Appendix 2. Minimum Number of Teeth on the Pinion in order to Avoid Interference

S/N	System of gear teeth	Minimum number of teeth on the pinion
1	$14\frac{1}{2}^{0}$ Composite	12
2	$14\frac{1}{2}^{0}$ Full depth involutes	32
3	20^{0} Full depth involutes	18
4	20^{0} Stub involutes	14

Source: Kurmi R. S and Gupta J. K (2005). A Text Book of Machine Design