UNIVERSITY OF NAIROBI

DEPARTMENT OF MECHANICAL AND MANUFACTURING ENGINEERING

FINAL YEAR PROJECT REPORT

TITLE: DESIGN OF A WINDMILL FOR PUMPING WATER

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OKERO JULIUS MOKAYA **F18/2412/2009**

PROJECT SUPERVISOR: PROF. NYANG'AYA J.A

DECLARATION

We declare that this is our original work and has not been presented for any degree in any other university or institution of learning.

MACHINI N. MERCYLINE **F18/2444/2009**

……………………………………. **DATE**………………………….

OKERO JULIUS MOKAYA **F18/2412/2009**

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This project is submitted as part of the examination board requirements for the award of the **Bachelor of Science** degree in **Mechanical and Manufacturing Engineering** from the **University of Nairobi.**

Project Supervisor: **Prof. Nyang'aya J.A**

……………………………………….. DATE…………………………….

ABSTRACT

Since time immemorial, the main source of energy has been coal, oil, natural gas, nuclear energy, wood and coal. However, all these sources are limited and are the main cause of pollution and this has led to development and more focus on sustainable energy supply with minimum pollution effects. Hence research and analysis has shown that wind energy, solar energy and biomass are the most prominent solutions to the above problems because they are eco-friendly and readily available in nature.

Wind energy can be generated using windmills that provide mechanical energy that is used directly on machinery e.g. water pump and grinder; or wind turbines that provide electrical energy. The main objective of our project was to design a windmill and therefore our scope will be limited to a windmill for water pumping water.

Windmills are classified into two main types based on the axis about which they rotate. Horizontal axis have the main rotor shaft running horizontally and if the rotor must be oriented in the direction of the wind, a wind vane is coupled with a servomotor. Vertical axis have the main rotor shaft running vertically. The rotor assembly can have two or more blades depending on the desired solidity.

In our design, we used a horizontal axis windmill with **24** blades. Each blade has a radius of **2 m** giving a total surface area of **0.5585 m^2** and this gives a solidity of **0.8**, the minimum theoretical optimum value for windmills.

The torque output of the windmill is **106.4572 Nm** and this is sufficient to sustain the desired flow rate of (**0.1736 X 10^{-3}) m^3 per second** with a maximum head of **30m**, and also overcome other barriers to motion such as friction. The total cost of the entire design is**Ksh. 86,902.95**

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ACKNOWLEDGEMENTS

First of all, we would wish to express our sincere gratitude to the Almighty God for giving us the intellect so used in the course of this project.

Special thanks to our project supervisor Prof. Nyang'aya J.A for being instrumental and supportive throughout the course of our project.

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Lastly, we wish to thank the department of Mechanical and manufacturing Engineering for the support and funding of this project to make it successive.

DEDICATION

We would like to dedicate this project to our families for playing a pivotal role, in not just our project but our education thus far.

LIST OF ABBREVIATIONS

C_p	- Coefficient of power
A_s	- Swept Area
A_B	- Area of one blade
V	- Velocity
U_o	- Speed of Wind
ρ	- Density
D	- Diameter
R	- Radius
λ	- Tip Speed Ratio
N	- Number of Blades
ω	- Angular Velocity
C_t	- Torque Coefficient
$\dot{m}$	- Mass Flux
P_w	- Wind Power
P_T	- Power Extracted From wind
T	- Torque
g	- Acceleration due to gravity
Q	- Flow rate
H	- Head
P_h	- Hydraulic Power
P_d	- Diametral Pitch
m	- Module
α	- Gear Pressure angle
E	- Young's Modulus
ν	- Poisson's Ratio
σ_y	- Yield Strength
σ_{UTS}	- Ultimate Tensile Strength

τ - Shear Stress

J - Polar Moment of Inertia

F - Force

M - Moment

1.0 INTRODUCTION

1.1 DEFINITIONS

Wind

This is air in motion and it is a natural resource that is freely available in space and moves at varying speeds depending on the geographical location.

Wind energy

Wind energy originates from solar energy where the sun heats the atmosphere unevenly causing some parts to be warmer than others. The warmer patches of air rise and other air patches blow in to replace them. Thus alternating air flow which results in wind.

Windmill

Windmills are machines that are used to harness the kinetic energy of the wind which blows over the blades rotor assembly causing it to rotate on a shaft. The resulting shaft poweris then used to provide mechanical work for pumping water.

1.2 BACKGROUND

The increase is costs of fossil fuels has greatly lead to the development of alternative sources of Green Energy that are environmental friendly and cheaper to produce. The major resource in this category is wind energy.

The exact origin of the first use of wind power is not known, however one of the earliest known uses dates back to 3500 B.C where they were used to drive sailboat using aerodynamic lift. The advancements of this design were adapted to China where the first windmill was developed and it was a vertical axis type that used sheet like wings to capture the wind. This setup was the connected to pulleys or other transmission mechanisms to be used for pumping water or grinding.

In the middle ages, wind energy was introduced in Northern Europe where Horizontal axis windmills were used where the sails connected to a horizontal shaft attached to a tower with gears and axles that were used to translate the horizontal motion into rotational motion. This type of windmill used drag forces for similar purposes of grinding and sawing timber.

In the 19th Century, wind energy developed in the United States where horizontal axis windmills were used for farms, ranches and to generate electricity. This is where the first multi-blade was developed for irrigation purposes.

The wind power technology has evolved greatly and this has been motivated by the incredible benefits resulting from wind energy. Very efficient and technologically up-to- date designs have been developed and are used all over the country especially in arid and semi-arid areas to pump water for domestic and irrigation purposes. Viability of windmills is however practical only in areas with free flow of air and therefore site selection is very critical in the initial design process.

The power generated from the wind can be supplied directly to the national grid system or used to drive other mechanical devices such as windmills and grinders. This has greatly reduced the levels of pollution resulting from the use of fossil fuels. This project is mainly based on the design of a windmill for water pumping.

1.3OBJECTIVE

The main objective of the project was the design of a windmill. The following were the key characteristics of the windmill to be designed:

- Improved torque characteristics on the current designs.
- Improved efficiency based on the current designs.
- The design should be a low-cost model.

1.4PROBLEM STATEMENT

Wind is a natural and renewable resource that is freely available all over the world.

Harnessing this power for pumping water would save a lot on power costs that are continuously on the rise. Also, wind power is available in remote regions that have not yet been connected to the national power grid. A windmill provides the best way of harnessing this wind power and using it to pump water that is below the surface or delivering the water to a raised storage tank.

By using a rigorous design approach, an optimum windmill design was developed. It had optimum power and optimum torque characteristics.

1.5SCOPE OF DESIGN

The design of a windmill is a very wide subject and therefore our design is based on data analysis of various components of windmill and their actual drawings. This includes the rotor assembly i.e. the blade and the hub, transmission shafts (both vertical and horizontal) the gear box which houses the gears, actuating pulley and the piston actuating mechanism, tail, turntable and the 2meters additional rig that will be joined to the existing rig.

The design does not cover the design and selection of the pump but a desirable pump that can function with the vertical movement of the actuating rod will be selected.

The design process started off with analysis of the existing Windmill designs and their respective operating condition. This formed a rough idea of the eventual windmill design that would be tailored to the Nairobi region and possibly the rest of Kenya. In order to achieve the optimum design characteristics such as Torque and power produced by the windmill, various calculations and structural analysis had to be done as well.

2.0 LITERATURE REVIEW

2.1 SITE SELECTION

The viability of a windmill is greatly affected by its location. The site must have sufficient wind power to move the windmill and also be away from obstructions that might cause turbulence.

The speed of wind for a given location is not constant and thus the climatic condition of the site should be examined for over on a year and recorded on a wind map which is then used to analyze the suitability of the site.

To avoid distractions, most windmills are located on hilly areas or the rigs are tall enough to ensure the rotor is far above the obstacles. The site of our windmill had been identified and there was no need for selection of another site. However we did an analysis on the site to determine its suitability and the findings were as follows:

- The winds speed of Nairobi varies from 3m/s to 4m/s and these were the maximum and minimum wind speeds recorded from January to December in the year 2013, hence the site is suitable.
- The location of the windmill is strategically away from tall buildings and trees and therefore minimal obstruction of the wind

2.2.1Advantages of Wind Power

- It is the cheapest source of energy. This is because it does not require importation and it is readily available.
- It is environmental friendly i.e. it is a major source of green energy as no greenhouse emissions are released to the atmosphere during its production.
- Require less labour expenses as maintenance is very minimal and few personnel are required at the site.

2.2.2Disadvantages of Wind Power

- Varying wind speeds and directions make it difficult to use wind as a consistent source of power.

- Initial investment on construction and installation of wind power machinery is very costly. In the year 2008-2009 the average investment required was $1300 and $1800 for every kW the windmill was to produce (wind energy).
- They produce so much noise and are therefore limited to areas away from homesteads and also away from wildlife reserves.

2.3 TYPES OF WINDMILLS

Windmills generally consist of two basic types with the classification being based on the orientation of the axis of the rotor .The main classifications are discussed below:

2.3.1Vertical Axis Wind Turbine (VAWT)

This has blades which are arranged on the vertical axis and are rotated by wind and therefore it doesn't require a yaw mechanism since it can harness wind from any direction. It does not rely on the direction of the wind to generate power as in the case of the horizontal axis.

They usually operate closer to the ground which has an advantage of allowing for placement or replacement of heavy equipment. However this is a disadvantage as winds are lower near ground level hence less power output.

There are two main types of the VAWT namely:

2.3.1.1 Savonius

It operates like a water wheel which uses drag forces. It has a simple design and is therefore relatively simple and cheaper to build. It is mostly used in situations that do not require large amounts of power. However, it is less powerful than most HAWT because it uses drag to rotate itself and has a higher power to weight ratio. The total amount of turning torque of the mechanism relies on the drag force on each blade.

2.3.1.2 Darrieus

It uses blades similar to those used in the horizontal axis wind turbine (HAWT). It has two or more curved blades that depend on wind in order to revolve around a central column. It functions by generating a lift using the rotating motion of the blades. The wind acting on the blade creates a rearward momentum change which propels the blade in the direction of rotation. This cannot occur unless the blades are already rotating and therefore they require a separate means of starting i.e. they are not self-starting.

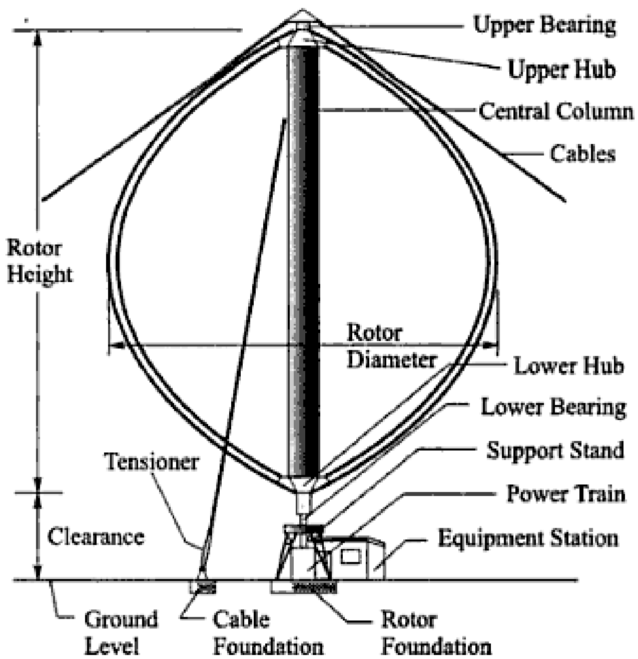


Fig 2.1: Darrieus Wind Turbine (Source: Green energy technology, Wortman, 1983)

2.3.2 Horizontal Axis Wind Turbine (HAWT)

It has blades that are similar in design to aircraft propellers where air flow over the airfoil shaped blades produces a lifting force that turns the rotor. They should be placed on towers to ensure maximum use of the winds at higher levels.

For large scale types, they have an active yaw mechanism with wind direction sensors and motors that will rotate the nacelle. In both upwind and downwind the rotors should be perpendicular to the direction of wind and if the rotor is held in a fixed position, only 21% of the wind energy will be captured (Wortman, 1983).For upwind type, the rotor rotation is accomplished by using a vane to measure the direction of the wind and then the information is communicated to the yaw drive. The yaw drive then drives the rotor so that the turbine is facing the direction of wind for maximum harness. They don’t suffer from wind shade phenomenon as the wind is tapped early enough before obstruction by the tower.

For downwind types, they don't use a yaw drive because the wind itself orients the turbine. The blades are situated on the downwind side and therefore capture the wind and rotate following its direction. These designs are prone to "wind shade" a process in which the wind flow is obstructed by an object e.g. the tower thus reducing amount of wind and therefore a reduction in the power output.

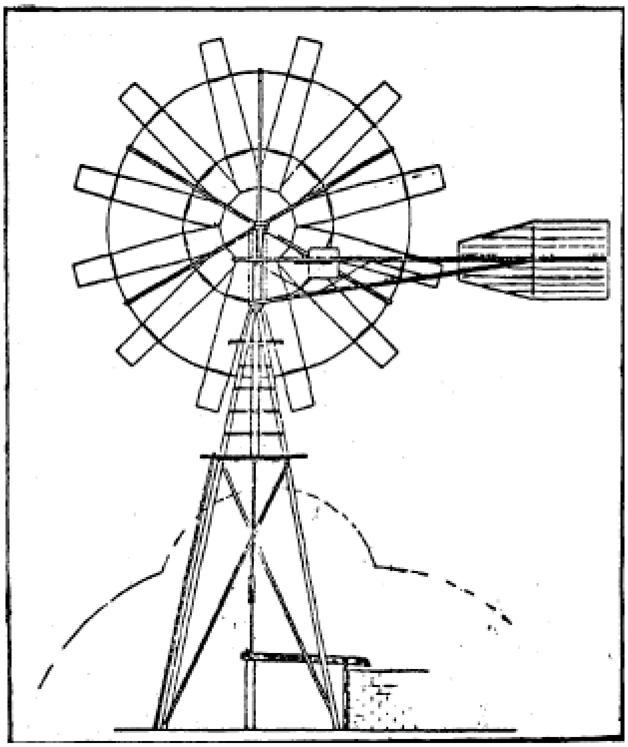


Fig 2.2: Horizontal Axis Wind Turbine (Source: Wind power Workshop, Hugh Piggott, 1982)

3.0THEORY

3.1 DEFINITIONS

3.1.1 Power Coefficient C_p

The power coefficient is the ratio of the actual power output (H_w) to the theoretical power in the wind (H_t).

$$Power = Force \times Velocity$$

$$Force = Rate\ of\ change\ of\ Momentum$$

But;

$$Momentum = Mass \times Velocity$$

For a fluid of density(ρ) , flows through a cross-sectional area of A, the mass flow rate $\dot{m}$ is given by:

$$\dot{m} = \rho AV$$

$$Average\ Force = \frac{1}{2}\rho AV^2$$

$$H_T = \frac{1}{2}\rho AV^3$$

$$C_p = \frac{H_w}{H_t} = \frac{H_w}{0.125\rho\pi D^2 V^3} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.1)$$

3.1.2 Swept Area, A_s

This is the section of air that encloses the wind turbine or windmill in its movement and interacts with the rotors to produce the rotation motion. For a Horizontal Axis Wind Turbine (HAWT), the swept area is circular in shape. On the other hand, for a Vertical Axis Wind Turbine (VAWT) with straight blade, the swept area is rectangular in shape.

The swept area for the HAWT is calculated by:

$$A_s = \frac{1}{4}\pi D^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.. \quad (3.2)$$

Where: A_s – Swept area (m^2)

D – Rotor Diameter (m)

NOTE: The rotor radius is the distance from the tip of one blade to the center. The diameter is twice this length.

3.1.3 Tip Speed Ratio, λ

Tip Speed ratio is ratio of the speed of the windmill rotor tip, at radius R when rotating at ω radians per second to the speed of the wind *V* m/s. It is numerically represented as:

$$\lambda = \frac{\omega R}{U_o} \quad \text{.............................. (3.3)}$$

When the windmill is stationery, the tip speed ratio is zero. This implies that rotor has stalled. This is experienced when the torque produced by the wind is below the level needed to overcome the resistance of the load. With a tip-speed ratio of 1, it implies that the blade tips are moving at the same speed as the wind (the wind angle that is 'seen' by the blades will be 45^0). At a tip speed ratio of 2, the tips are moving at twice the speed of the wind and so on.

From empirical results, the optimal tip speed ratio to ensure maximum power extraction is achieved for a windmill with N blades is:

$$\lambda = \frac{4\pi}{N} \quad \text{.................................. (3.4)}$$

3.1.4 Specific Speed of the Windmill

This is the angular velocity in revolutions per minute at which a turbine will operate if scaled down in geometrical proportion to such a size that it will develop unit power under unit head.

3.1.5 Cut in Speed

This is the speed at which the turbine starts to produce any useful power. It is the lowest speed at which power output of the turbine H_w is greater than zero.

3.1.6 Cut out Speed

This is the wind speed at which the turbine stops to produce any useful power. This is the highest speed at which power developed by the wind turbine is just zero.

3.1.7 Rated Wind Speed

Wind speed at which the rated power is produced, this value defines the shape of the power curve.

3.1.8 Torque Coefficient, C_t

The torque coefficient is the non-dimensional measure of the torque produced by a given size of rotor in a given wind speed. This is given as the ratio of the actual Torque produced to the torque due to the force of the wind on the rotors.

It is represented mathematically by:

$$C_t = \frac{T}{\frac{1}{2}\rho A_s {U_o}^2 R} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3.5)}$$

Where: T – The actual torque produced (Nm)

U_o – Wind speed (m/s)

A_s – Swept Area (m^2)

R – Radius (m)

3.1.9 Rotor Solidity

Solidity of a windmill loosely refers to the proportion of a windmill rotors' swept area that is filled with solid blades. This is the ratio of the sum of the width or 'chords' of all the blades to the circumference of the rotor.

This is represented mathematically by:

$$Solidity = \frac{NA_B}{A_s} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3.6)}$$

Where: σ - Rotor Solidity

N – Number of Blades

A_B – Area of one blade (m^2)

A_s – Swept Area (m^2)

Most of the HAWT Windmills that are used on wind pumps are Multi-bladed rotors. They are usually said to be of high solidity due to the fact that a large proportion of the swept area is 'solid' with blades. High solidity means that the windmill will run at a very low speed and creating high torque in the process. The blades of such high solidity windmills set a rather coarse angle to the plane of rotation just like a screw with a coarse thread. At maximum efficiency, the tip-speed ratio is very low at approximately 1.25. The efficiency of the high solidity windmills is slightly lower than that of the faster types of rotors. These multi-bladed rotors have a much higher torque coefficient at zero tip-speed ratio (i.e. between 0.5 and 0.6) than any other types. The starting torque, which is usually higher than the running torque, together with the low speed of rotation in the given wind regime make the multi-bladed high solidity windmill suitable for driving the reciprocating borehole pumps.

On the other hand, the low solidity rotors have a higher efficiency with the highest values of C_p. They have very high tip speed ratio. To achieve these optimum operating conditions, the rotors have to be set at a slight angle to the plane of rotation. The starting torque is very low implying that they can only start against the loads that require little torque to start them such as centrifugal pumps and electricity generators.

A typical windmill will have a starting torque that is five to twenty times the starting torque of the three bladed low solidity wind turbines.

3.1.10 Thrust Coefficient C_T

This is non-dimensional measure of the force that falls on the windmill. It given as the ratio of the actual thrust force on the windmill to the average force from the wind.

It is represented mathematically as:

$$Force\ of\ the\ Wind = \frac{1}{2}\rho A_s U_o{}^2$$

Thrust Force on the windmill, F_T;

$$F_T = \frac{1}{2} C_T \rho A_s {U_o}^2$$

$\therefore C_T = \frac{F_T}{\frac{1}{2}\rho A_s {U_o}^2}$ …………………………………….. (3.7)

3.2 WIND POWER DERIVATION

3.2.1 Wind Power:

$$Wind\ Power\ (P_w) = Kinetic\ Energy\ per\ Unit\ Time$$

$$= \frac{1}{2}\dot{m}v^2$$ ……………………………………… (3.8)

Where:$(Mass\ Flux)\ \dot{m} = \frac{dm}{dt} = \rho A v$

$$\rho = Density\ of\ air\ (Kg/m^3)$$

Thus Power (P_w) $= \frac{1}{2}\rho A v^3$ Watts …………………………. (3.9)

Where: A$= Swept\ area\ of\ blades\ (m^2)$

$$A = \pi R^2$$

Where: R = $Radius\ of\ the\ Blade\ (m)$

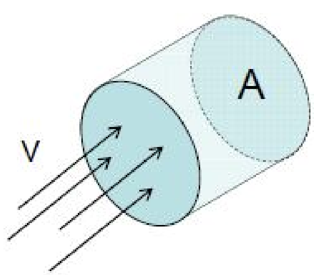


Fig 3.1: Illustration of area and velocity.

3.2.2 Efficiency in Wind Power Extraction

The efficiency in wind Power extraction is a function of Power Coefficient C_p, where C_p is the ratio of power Extracted by the Windmill to the total contained in the wind resource.

$C_p = \frac{P_T}{P_w}$.. (3.10)

Where: Windmill Power Extracted $P_T = P_w \times C_P$

$$= \frac{1}{2}\rho A v^3 \times C_P \text{ (Watts)}$$

The Betz Limit is the maximum possible value for C_p which is equal to $\frac{16}{27}$ but the optimum possible for a multi-blade windmill is 30%.

3.2.3 Torque Extracted

In the windmill used for pumping water, Torque output is key.

Torque is given by the ratio power extracted to rotor speed.

$T = \frac{P_T}{\omega}$ ……………………………………………….. (3.11)

Rotor Speed $\omega = \frac{\lambda v}{R}$

Where: $v = Wind\ Speed\ (m/s)$

$\lambda = Tip\ speed\ ratio$

$R = Length\ of\ blade\ (m)$

Thus: $T = \frac{P_T}{\lambda v / R}$ ………………………………… (3.12)

But $P_T = \frac{1}{2}\rho\pi R^2 v^3 \times C_p$ Watts

Therefore: $T = \frac{\frac{1}{2}\rho\pi R^2 v^3 \times C_p}{\lambda v / R}$

$$T = \frac{1}{2}\frac{\rho\pi R^3}{\lambda} v^2 C_p \text{(Nm)} \text{ ………………………….. (3.13)}$$

3.3 EFFICIENCY, POWER AND TORQUE CHARACTERISTICS

3.3.1 Power – Speed Characteristics

In order for the wind pump to work as desired, the pumping requirements must be matched with the wind speed as well as the rotor shaft power available. The sample figure below indicates how one would match a wind speed of 5 m/s with required pump power requirement.

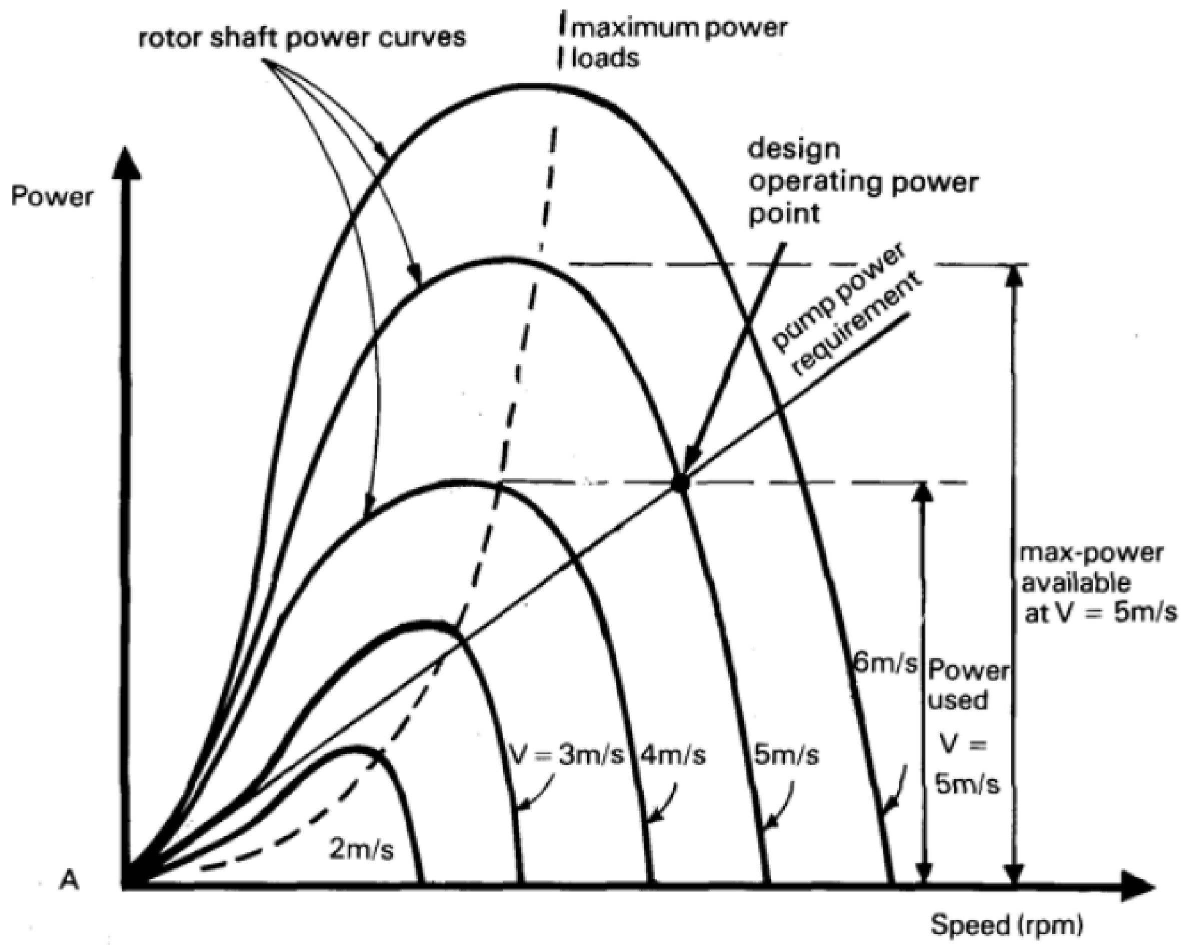


Fig 3.2: Power of a windmill as a function of rotational speed for various wind speeds. Source S.B Kedare (2003)

3.3.2 Torque-Speed Characteristics

In pump selection, the rotor shaft torque has to be matched to the pump torque requirements as well. The pump in this case is the piston pump. The figure below shows how one would match a pump depending on the torque requirements.

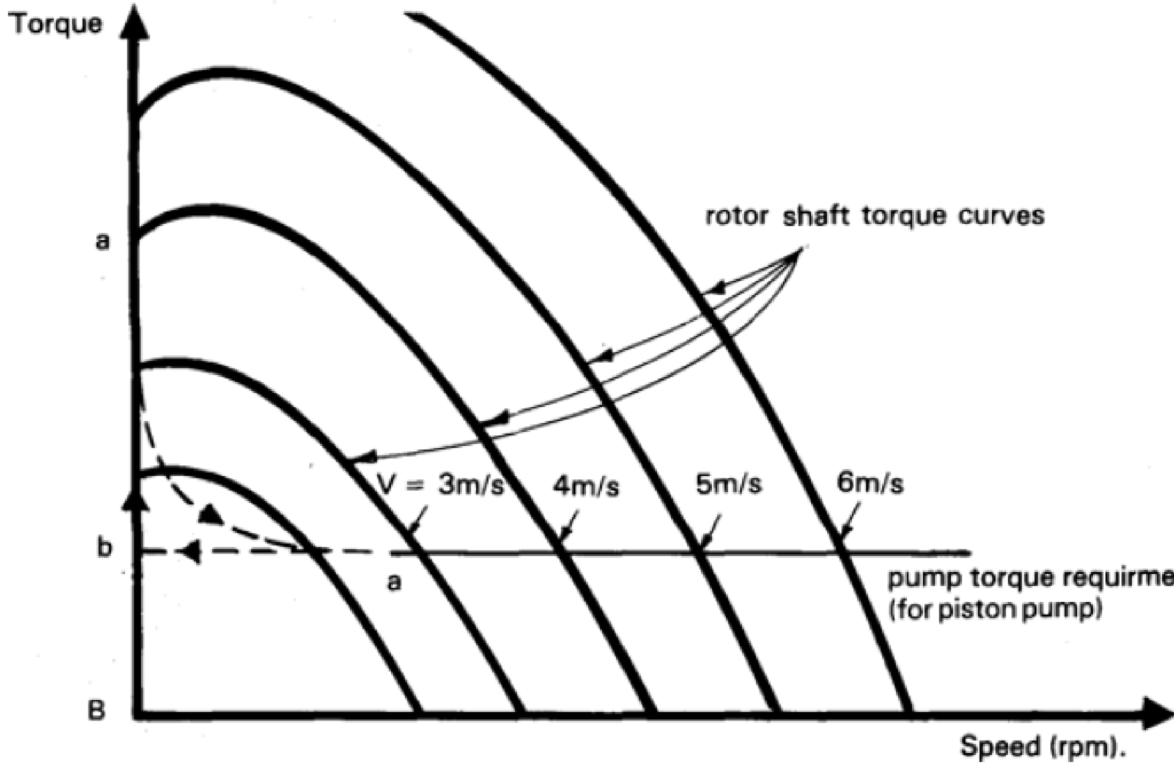


Fig3.3: Torque of a windmill rotor as a function of the rotational speed for the various wind speeds. Source S. B. Kedare (2003)

3.3.3 Power Coefficient (C_P) and Torque Coefficient (C_T) against Tip Speed ratio (λ)

Different configurations of windmills result in different power coefficients as well as torque coefficients. They also have different tip speed ratios. From empirical methods, it is much easier to predict the region where a particular windmill or wind turbine design configuration will lie. The most common designs are; Savonius (Chinese panamone), American Multi-blade, Cretan sail rotor, Four bladed (curved steel plate), three bladed airfoil, Darrieus Airfoil and the two bladed airfoil.

The graphs below illustrate the relationship of the power coefficient and torque characteristics against the tip speed ratio for these windmill and wind turbine designs.

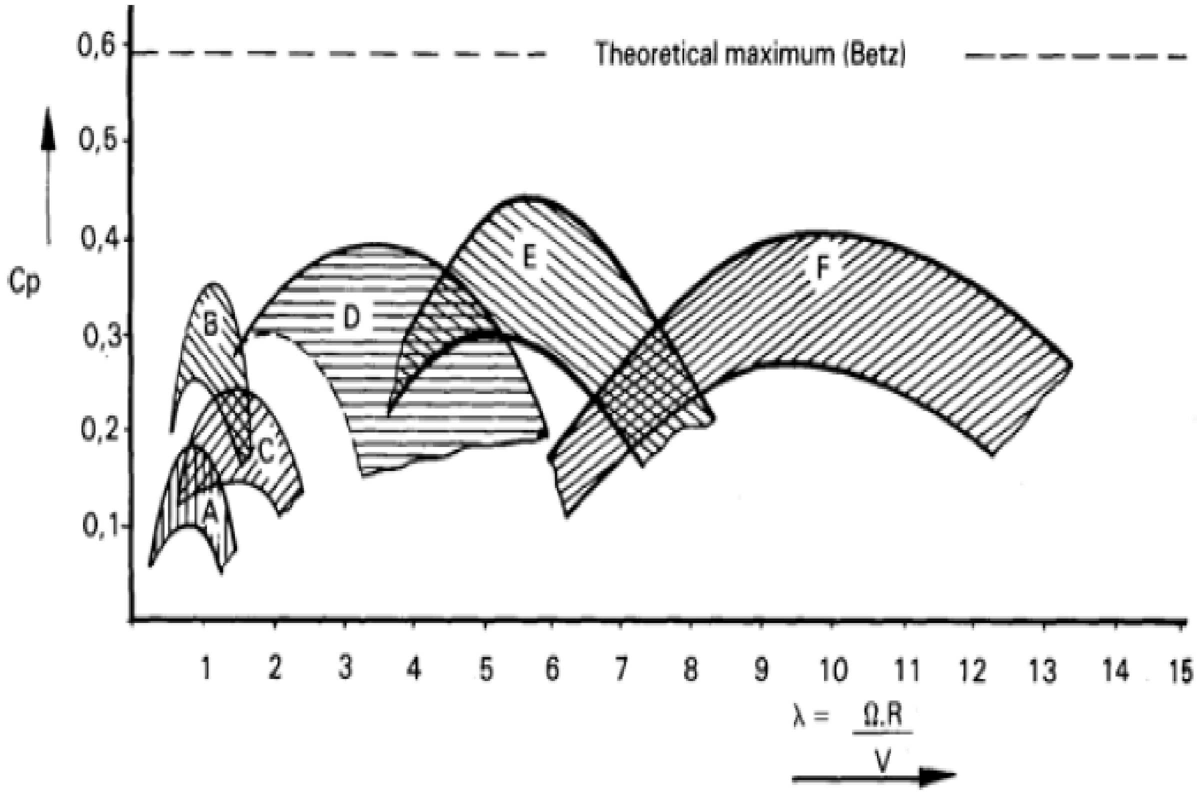


Fig3.4: Graph of the Coefficient of Power (C_p) against the tip-speed ratio (λ) for different windmill and wind turbine design

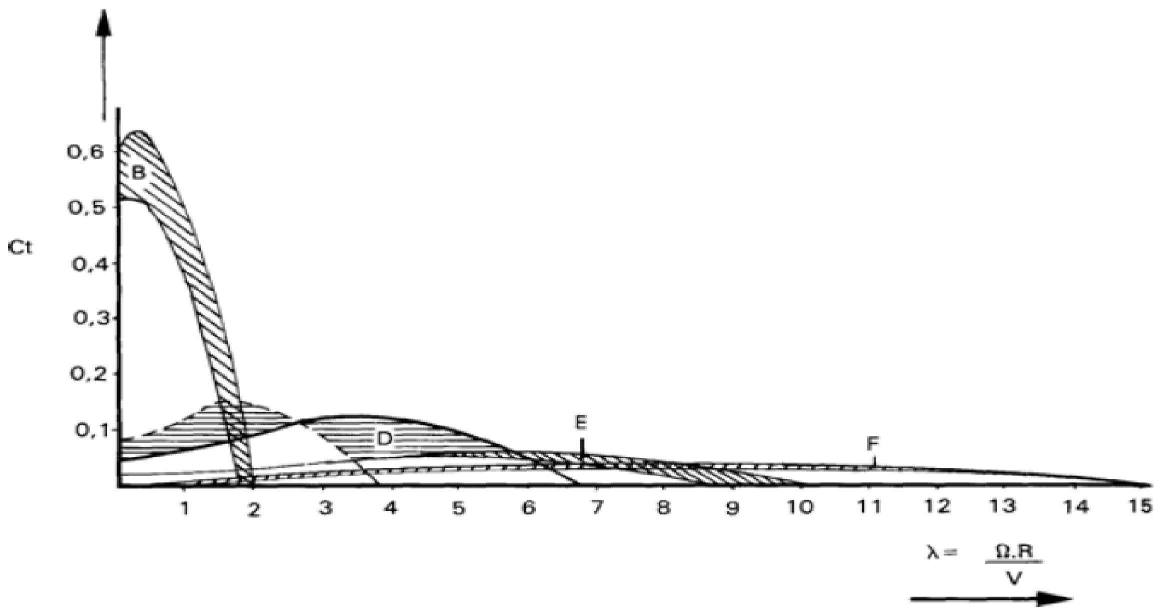


Fig 3.5 : Graph of the Coefficient of Torque (C_t) against the Tip-speed ratio (λ) for different windmill and wind turbine design

KEY:

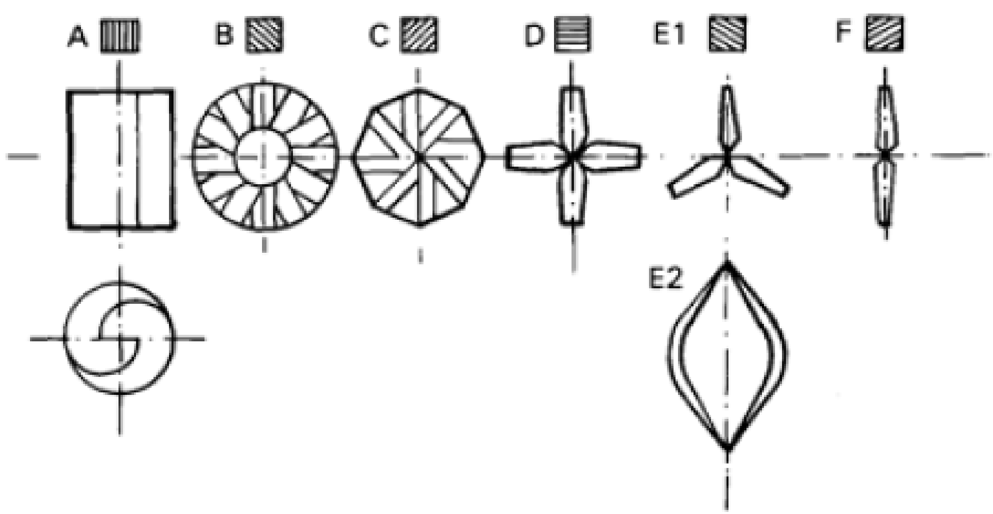


Fig3.5: A key to the different windmill and wind turbine designs that have been used in the graphs above

4. DESIGN

4.1 BLADE

4.1.1 Blade Design Consideration

Betz' Theorem Revisited

Power is extracted from the wind by decelerating it. There's however a limit as to how much power can be extracted from the wind. According to Betz' Theorem, this deceleration is as much as a third of the upstream velocity. This translates to 59.3%. Any further deceleration of the wind will divert the wind away from the rotor.

According to Newton's third law of motion; for every action, there is equal and opposite reaction. Therefore, the decelerating force on the wind is equal to the thrust force which the wind applies to the rotor. In designing the rotor, the main goal is to make sure that the thrust produced is able to produce Betz' optimum deceleration.

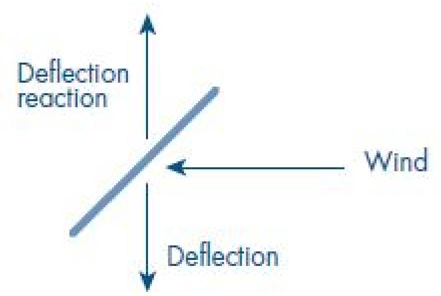


Fig4.1: Illustration of the reaction force that causes thrust

The other source of the thrust that enables the rotors rotate is the Bernoulli Effect. According to the Bernoulli Effect theorem, faster moving air has lower pressure. The blades of the windmill are shaped such that the air molecules moving around the blade travel faster downwind side of the blade than those moving across the upwind side of the blade. This shape is known as an aerofoil. The curve in the downwind side of the blade is much larger whereas the one on the upwind side is relatively flat. Given that the air moves at a faster velocity on the curved downwind side of the blade, the pressure on this side of the blade is less. This difference in pressure on the opposite sides of the blade causes the blade to get a 'lift' towards the curve of the aerofoil.

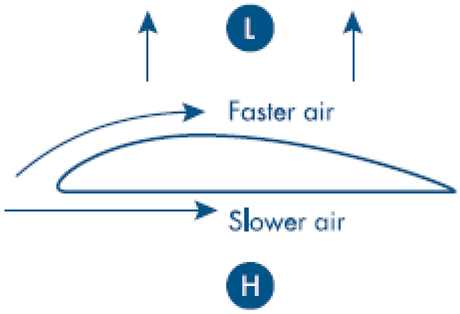


Fig4.2: Illustration of the Bernoulli Effect causing lift (blade cross-section)

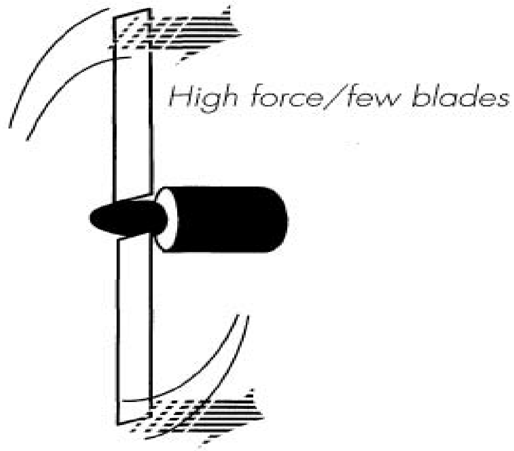
Torque and Speed

The two main components in mechanical power are force and speed. Torque is a twisting or turning force and pumping requires a lot of torque particularly when starting off from an idle state. Generators used for power production on the other hand require a lot of speed. The power might be the same but the pump and generator will utilize this power differently. The power from the rotor is a function of both torque and rotations per minute (angular velocity of the rotors).

The following table shows the typical choices for the tip speed ratio and blade number for the case of pump (windmill) and generators (wind turbine).

Tip Speed Ratio	No. of Blades	Functions
1	6-20	Slow pumps
2	4-12	Faster pumps
3	3-6	Dutch 4-bladed
4	2-4	Slow generators
5-8	2-3	Generators
8-15	1-2	Fastest Possible

Table4.1: Typical choices for the tip speed ratio and blade number for pump and generator (*Wind Power workshop, Hugh Piggot, Centre for Alternative Technology publication, London, UK)*



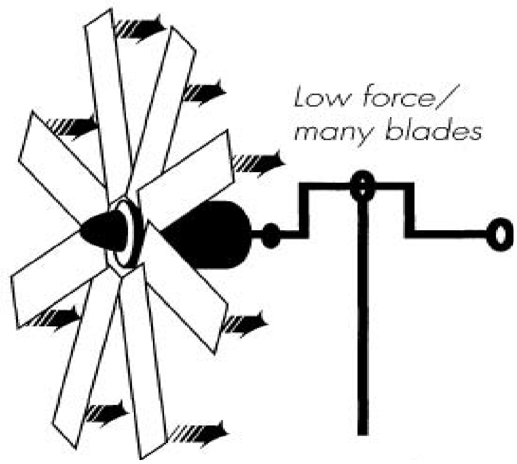


Fig 4.3: Generator – High Speed and low torque *Fig4.4: Pump – Low Speed and high torque*

(Tip Speed Ratio – 4) (*Tip Speed Ratios – 2)*

The Aerofoil

In the design of the cross-section (aerofoil) of the blades, there are some key considerations and specifications that should also be taken into account to ensure maximum thrust and lift. The following figure shows these important specifications.

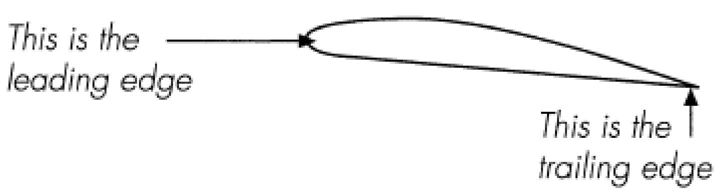


Fig 4.5: Illustration of the leading and trailing edges

The other important specification is the **Angle of attack (α)**. This is the angle between the chord line and the relative air (or wind) movement. It is illustrated in the figure below.

Fig4.6: Illustration of the angle of attack

The angle of attack has a direct impact on the lift experienced and thus the lift coefficient as well. Wind tunnel studies have proved that the drag to lift ratio is not a constant factor. The best ratio is usually at an angle of attack of around 40^0.The **Lift Coefficient (C_L)** is approximately equal to $2\pi\alpha^r$ radians.The typical range of the Lift Coefficient is 0.8 to 1.25.The following table shows data for the different shapes.

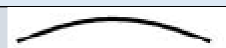
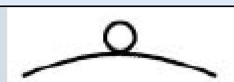
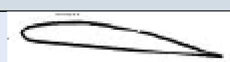
Section	Sketch	Drag/Lift Ratio	Optimum angle of attack	Lift Coefficient (C_L)
Flat plate		0.1	5^0	0.8
Curved Plate (10% curvature)		0.02	40^0	1.25
Curved plate with tube on concave side		0.003	4^0	1.1
Curved plate with tube on convex side		0.2	14^0	1.25
Aerofoil		0.01	4^0	0.8

Table4.2: Data for some common blade cross-sections

Blade Design

In the design of a blade, it's convenient to divide the blade into sections. This is particularly due to the fact most blade designs taper towards the blade tip. This is to reduce the drag force to the end of the blade. For convenience, these sections could be referred to as stations.

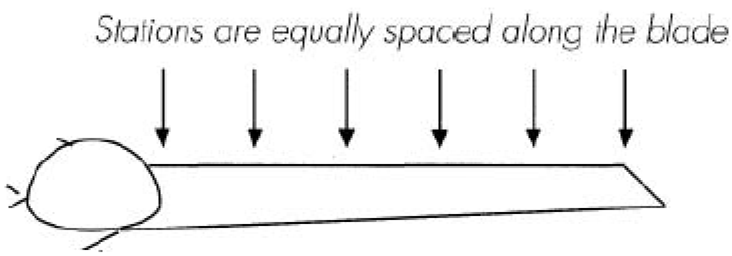


Fig4.7: An Illustration of the stations

At each of these stations, the following parameters have to be noted or taken into account:

- Radius
- Setting Angle (Pitch)
- Chord
- Thickness

Radius

This is the distance from the center of the rotor to the rotor station.

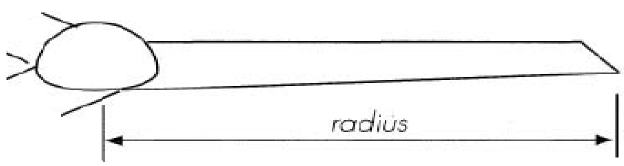


Fig4.8: Illustration of the Radius

Setting Angle (Pitch)

The Setting angle or pitch (β) is the angle between the chord line and the plane of rotation of the windmill rotor.

Both the setting angle and flow angle depend on how far from the root of the blade you go. At the root of the blade, the wind comes in at almost a right angle towards the blade. Therefore, the pitch/setting angle will have to be much larger at the base. Towards the tip, it should be smaller since the headwind is much larger meaning that the relative wind is rotated.

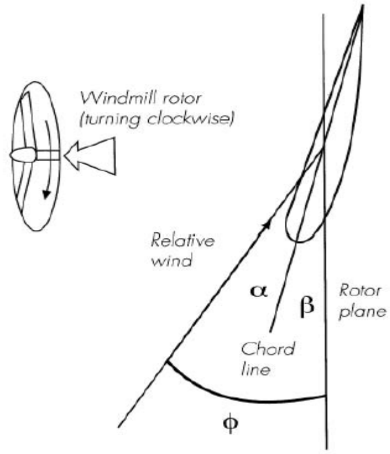


Fig4.9: Illustration of the angle of attack (α), setting angle (β) and flow angle (Φ)

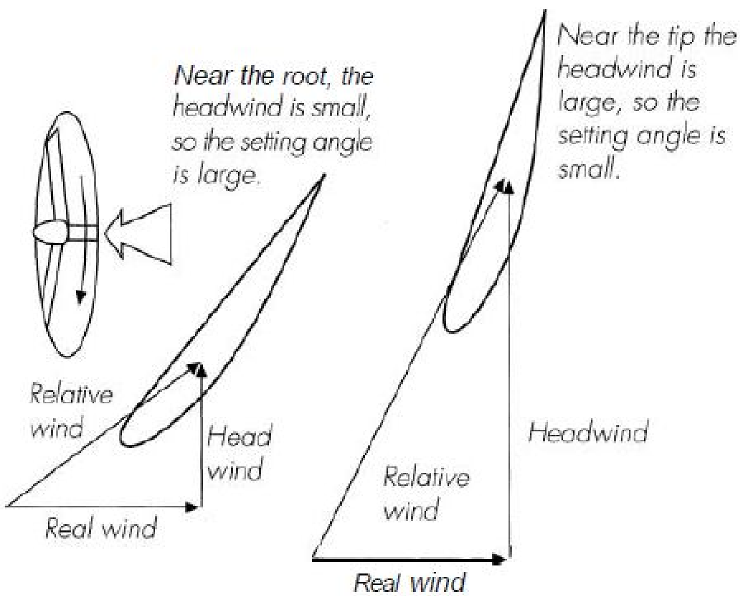


Fig4.10: The variation of the setting angle at the root and at the tip

Chord

This is the width of the blade at that particular station. Theoretically, the chord can be calculated by equating the aerodynamic thrust with the Force required for the Betz' change of momentum. The thrust is got from the lift calculations and the change of momentum is from Newton's third law of motion.

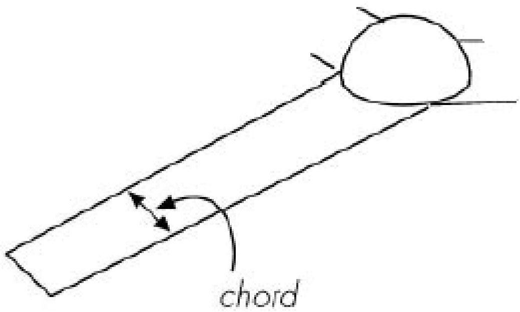


Fig4.11: Illustration of the Chord

Twisted and Tapered Blades

Most of the commercial windmills are made of twisted and tapered blades. The following are the reasons as to why this is mainly so:

- Both the tapered and twisted blades have a slight improvement on the efficiency.
- A tapered blade is much stronger compared to that with an equal chord all-through. Given that the maximum bending stresses are experienced at the root of the blade, a tapered blade means that the blade will be less vulnerable to yielding due to fatigue as compared to a 'straight blade'.
- Starting of tapered blades is better since the wider root gives a slightly better torque.

In the design of windmills, the most important parameter is torque. This means that the taper is not necessary.

4.1.2 Calculation of the Rotor Radius

Power required to pump water is normally determined by the flow rate and the total head generated. This is shown below;

Water Horsepower $P_w = \rho g Q H$ Watts

Where: ρ - Density of water (kg/m^3)

g - Acceleration due to gravity (m/s^2)

Q - Flow rate (m^3/s)

H - Total Pumping head in meters of water (moW)

Proposed design requirements

Proposed head = 30 m

Proposed Volume Flow rate= 15m^3 per day =0.1736×10^{-3}m^3 per second

Taking the daily mean wind speed in Nairobi;

Highest daily mean = 4 m/s

Lowest daily mean = 3 m/s

Given the average wind speeds, the lowest average mean wind speed was selected i.e. 3.5 m/s.

The average atmospheric temperatures for Nairobi are at an average low of 12^{0}C and a high of 28^{0}C.

Power required to pump the water is given by:

Hydraulic Power, $P_h = \rho g Q H$W

Substituting for g=9.81 m^2/s; Q = 0.1736×10^{-3}m^3/s; H = 30 m

$$= 1000 \times 9.81 \times 0.1736 \times 10^{-3} \times 30$$

$= 51.09048\ W$

In order to the power required from the wind, various losses have to be considered.

The table below shows various losses (wind energy handbook)

Factor	Typical efficiency
Rotor to shaft	92-97%
Shaft to gear box	93-96%
Gear box	99 %
Pump	60-75%

Table 4.3: Power losses in a windmill

$$efficiency = \frac{output}{input} \times 100\%$$

Hence before losses occur, the input power to the system is given by:

$$input = \frac{output}{efficiency} \times 100$$

1. **Before pump loss**

   Average Efficiency of pump is 67.7%

   $$P = \frac{51.09048}{67.5} \times 100$$

   $= 75.6897$ W

   This is the power that is transmitted from the shaft to the pump.

2. **Before shaft losses**

   Average efficiency of the shaft from the gearbox to the pump is 94.5 %.

   $$P = \frac{75.6897}{94.5} \times 100$$

   $= 80.0948\, W$

   This is the power that is transmitted from the gearbox to the vertical shaft

3. **Before gearbox losses**

   Average efficiency of the gear box is given as 99% which is the actual efficiency of the gears

   $$P = \frac{80.0948}{99} \times 100$$

   $= 80.9039\, W$

   This is the total power transmitted from the horizontal shaft to the gearbox.

4. **Before shaft losses**

   The efficiency of the horizontal shaft is given by 94.5%

   $$P = \frac{80.9039}{94.5} \times 100$$

   $= 85.6125\, W$

   This is the amount of power required from the wind.

5. However, according to the **Betz limit**, the efficiency in wind power extraction is a function Power Coefficient C_p, where C_p is the ratio of power Extracted by the Windmill to the total contained in the wind resource.

   $C_p = \frac{P_T}{P}$

   Where: Windmill Power Extracted $P_T = P \times C_P$

   $= \frac{1}{2}\rho A v^3 \times C_P$ (Watts)

   The Betz Limit is the maximum possible value for $C_p$which is equal to $\frac{16}{27}$ and therefore making the maximum efficiency being 59%.

   However the maximum practical value of Cpis 0.3 (wind power handbook)

   Hence the power intercepted by the wind is given by:

   $$P_{Wind} = \frac{85.6125}{0.3} \times 100$$

   $$= 285.3751\, W$$

   This is the power in the wind.

6. The power can also be calculated by the formula;

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

   Where: A- Area covered by the blades

   Atmospheric air conditions;

   Mean wind speed = 3.5 m/s

   Mean temperature = $\frac{28 + 12}{2} = 20℃$

   Pressure = 1.02 Bars

   Calculating air properties at 20℃ and 1.02 Bars [ ]

Temperature (K)	ρ (kg/m^3)
275	**1.284**

293	**?**
300	**1.177**

Table 4.4: Density of Air (Source: Thermodynamic and Transport Properties of Fluids, arranged by G.F.C Rogers and Y. R. Mayhew)

Interpolating;

ρ_{293K} = 1.284 + $\left[\left(\frac{293-275}{300-275}\right) \times (1.177 - 1.284)\right]$ kg/m^3

= 1.207 kg/m^3

Recall $P_w = \frac{1}{2}\rho A v^3$ kW

Making A the subject;

$A = \frac{P_w}{0.5\rho v^3}$ m^2

$$\pi R^2 = \frac{P_w}{0.5\rho v^3}$$

$R^2 = \frac{P_w}{0.5\rho v^3}$

$$= \frac{285.375}{0.5 \times 1.207 \times \pi \times 3.5^3}$$

$= 3.4969\, m^2$

$R = 1.87\, m \approx 1.9\, m$

But this is the radius at mean speed of 3.5m/s, hence a safety factor of 1.0526 was used to ensure its functionality even at very low speed of 3m/s.

Hence $R = 1.052631 \times 1.9$

$R = 2$

$Diameter\ of\ rotor\ (D) = 4\, m$

4.1.3Final Blade Design

The blade is designed such that it is narrower at the base and wider at the top. The projected area of the blade is a trapezium as shown below.

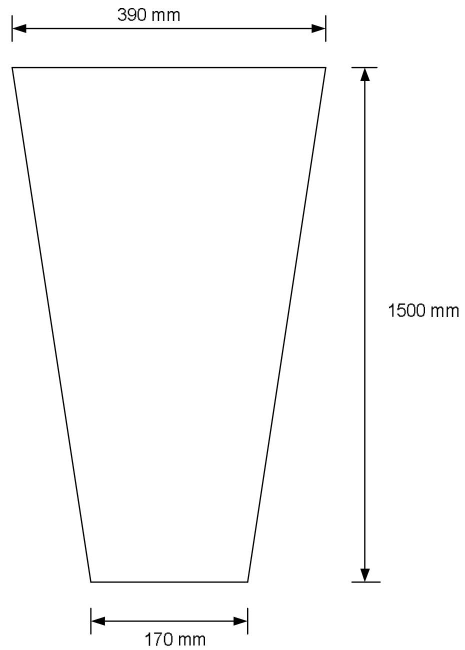


Fig4.12: Projected Area of the blade

To achieve the optimum interaction of the blade with the wind, the blade is usually of a 10% curvature. The top end therefore has a curvature **109.163^0** and a radius of **225.2 mm**. The bottom end has a curvature of **109.163^0** as well with a radius of **98.2 mm**.

The blade is made of Gauge 16 sheet metal which has a thickness of **1.626 mm**. Using Gauge 16 Mild Steel sheet metal ensure that the rotors are rigid and can therefore withstand the various wind conditions that it may be exposed to during operation. The rotor assembly is made up of 24 blades.

In designing the rotor assembly, the solidity should be at least 0.8 for optimum torque characteristics.

Calculating the Solidity of the rotor assembly:

$$Solidity = \frac{NA_B}{A_S} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(4.1)}$$

Where:

N – Number of Blades

A_B – Area of one blade (m^2)

A_s – Swept Area (m^2)

$$A_B = \frac{1}{2} \times 1.5(0.39 + 0.17) = 0.42\,m^2$$

$$A_s = \pi R^2 = \pi \times 2^2 = 12.5664\,m^2$$

$$\therefore Solidity = \frac{24 \times 0.42}{12.5664} = \mathbf{0.8021}$$

The solidity for the rotor design is 0.8021 which meets the threshold to achieve the best torque characteristics.

At the base of the blade, it is bolted to a support that is also bolted onto the inner ring. This base support ensures a rigid framework. The blade is joined to the outer ring by a weld joint. A support similar to that at the base could be added for extra support. This might be vital for windmills that might be exposed to high wind speeds for prolonged periods of time.

4.2 POWER TRANSMISSION MECHANISMS USED

Gears, shafts and bearings are the fundamental transmission mechanisms used in most engineering processes. They are the primary components used in gearboxes, drive trains and transmissions.

4.2.1 Gears

These are wheel-like shaped components that have equally spaced teeth around their outer periphery and it engages another toothed mechanism in order to change the speed or direction of transmitted motion .Gears are mounted on rotatable shafts with the teeth on one gear meshing with the teeth of the other gear and thus transmitting rotary motion in the process. This also causes transfer of torque from one part of the machine to the other.

Gear terminologies:

- **Pinion-** this is the smaller gear of the two gears.
- The larger gear is usually called **gear** regardless of which is doing the driving.
- C**learance-**this is equal to the whole depth minus the working depth.
- **Dedendum-**this is the radial depth of a tooth below the pitch circle.
- **Addendum-**this is the radial height of the tooth above the pitch circle.
- **Circular pitch-** this is the distance along the pitch circle from a point on one gear tooth to a similar point on an adjacent gear tooth.
- **Diametral pitch (**P_d): The number of teeth of a gear unit pitch diameter. A toothed gear must have an integral number of teeth. The circular pitch, therefore, equals the pitch circumference divided by the number of teeth. The diametral pitch is, by definition, the number of teeth divided by the pitch diameter. That is:

  – …………………………………… (4.2)

- **Module** (m): Pitch diameter divided by number of teeth. The pitch diameter is usually specified in inches or millimeters; in the former case the module is the inverse of diametral pitch.

  $$m = D/N$$

- **Pressure angle** (α): The angle between the common normal at the point of tooth contact and the common tangent to the pitch circles. It is also the angle between the line of action and the common tangent.

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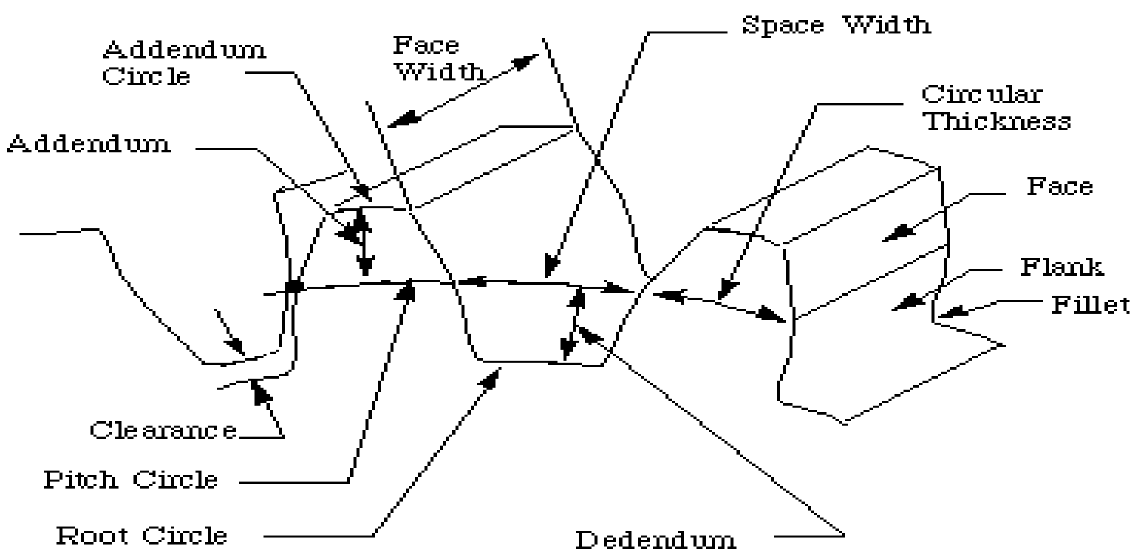

Fig4.13: Nomenclature of gear (source Shigley's Mechanical Engineering design 8th Edition Budyas Nisbett)

There are four main types of gears being used and they include:

4.2.1.1 Spur Gears

These types of gears have teeth that are aligned in a direction parallel to the gear axis and they are designed to mesh with another spur gear that is in a parallel shaft. They are the most commonly used and are the least expensive. However they cannot be used when direction change between two shafts is required.

The figure below shows an illustration of a spur gear.

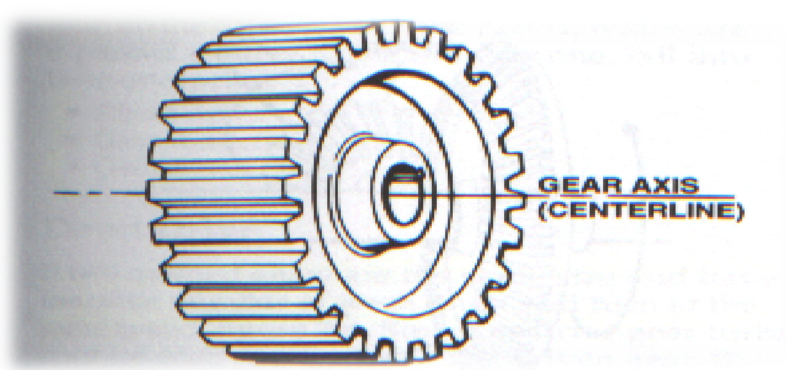

Fig4.14: Spur gear

4.2.1.2 Helical Gears

They are similar to the spur gears but the teeth are at an angle to the shaft. The angle is called the helix angle and normally ranges from 10^0 to 30^0.For helical gears; the contact of the teeth begins at one end of the tooth and then traverses diagonally across the width of the tooth to the other end. This phenomenon makes them smoother and quiet and they can carry higher loads but at a lower efficiency.

They have higher tooth strength than spur gears because the teeth are longer and the gears in mesh can produce thrust forces in the axial directions and are mostly used in transmission components, automobile and speed reducers but they are expensive than spur gears.

The figure below is an illustration of the helical gears.

Fig4.15: Helical gear. (Source: Shigley's Mechanical Engineering Design, 8th Edition, Budyass Nisbett*)*

4.2.1.3 Bevel Gears

These are gears used to transmit speed and torque between two shafts that are not parallel but are at an angle to each other. The teeth of this type of gear are formed on a conical surface. They can be spiral bevel in which the teeth are oblique to provide gradual engagement or they can also be straight in which the teeth are straight.

Bevel gears are useful when the direction of a shaft's rotation needs to be changed. They are usually mounted on shafts that are 90 degrees apart, but can be designed to work at other angles as well. The teeth on bevel gears can be straight, spiral or hypoid. Straight bevel gear teeth actually have the same problem as straight spur gear teeth as each tooth engages; it impacts the corresponding tooth all at once

Fig4.16: Bevel gears (Source: Shigley's Mechanical Engineering Design, 8th Edition, Budyass Nisbett*)*

The table below shows the efficiencies of various types of gears and their standard gear ratios

Type	Standard gear ratio	Pitch line velocity	Efficiency range
Spur	1:1 to 6:1	25	98-99%
Helical	1:1 to 10:1	50	98-99%
Double helical	1:1 to 15: 1	150	98-99%
Bevel	1:1 to 4:1	20	98-99%
Worm	5:1 to 75: 1	30	20-98%
Crossed helical	1:1 to 6:1	30	70-98%

Table 4.5: efficiencies of various types of gears

4.2.2 Materials Used in the Manufacture of Gears

Gears are mostly made of steel, Bronze, plastic and iron. Iron is not mostly used but it is used when good castability properties are desired. Plastic gears are used due to their good moldability properties but are limited in their load carrying capacity. Bronze is used where friction is a major concern but they are very costly. Steel is the most commonly used and they exist in a wide range of alloys .gears steel is available in grades such as grade 1, 2, and 3 as classified by the American gear manufacturers association. Higher grades represent higher quality steels. This is dependent on the amount of carbon i.e. low carbon which cost less but is less durable.

Steel gears are usually heat treated, or some are carburized and hardened to improve their performance.

In our design, the following were the requirements of the gears that were required:

- Transmit motion from one horizontal shaft to another
- Be readily available.
- Ease to machine and therefore economically viable.
- High efficiency in transmission.

From the above analysis, the gear that best suits our design requirements was a spur gear made of steel .Our design used two pairs of gears both of which are enclosed in the gearbox. The illustration is shown below:

The small pair of gears (pinion) is used to drive the bigger gears and this greatly reduces the speed thus increasing the torque output.

Gear ratio is given by: $\frac{number\ of\ teeth\ on\ driven}{number\ of\ teeth\ on\ driver}$

Hence substituting for numbers of teeth,

$gear\ ratio = \frac{60}{19}$ =3.15

The pressure angle is standard (20^0) and the other properties are based on the above calculated specifications of the gear. They are standard properties and are obtained from standard ASTM tables for gear.

The table below shows various properties of the gears.

Gear ratio	3.15
Module	5.5 mm
Centre distance	217.25
Pressure angle	20^0
Number of teeth on driver	19

Number of teeth on driven	60
Efficiency	98%
Bending fatigue	352Mpa
Contact fatigue	114Mpa
Modulus of elasticity	206 GPa
Poisson's ratio	0.3
Material	Mild steel
Density	7850Kg/m^3
Gear thickness	20mm

Table4.6: Properties of Gears

4.2.3 Bearings

These are machine components designed to provide support for rotating machine elements by taking pure radial loads, pure thrust loads or a combination of the two.

4.2.3.1 Materials Used in the Manufacture of Bearings

Babbitt

These are tin and lead alloys. Are used for heavy duty and engine bearing, have ability to embed dirt and have excellent properties under boundary lubrication conditions.

Bronze and copper alloys

These are mainly grouped into copper-lead, lead bronze, tin-bronze and Aluminium.

Aluminium

Aluminium bearing alloys have high wear resistance, load carrying capacity, thermal conductivity and corrosion resistance. They are used in hydraulic gear pumps, main bearings and in reciprocating compressors.

Plastics.

This type of bearings have high strength and require little or no lubrication.

Bearings are classified as:

- **Rolling contact bearings**.

These are of three main types namely ball bearings, roller bearings and needle bearings. The load is transferred through rolling elements such as balls, needles, straight tapered cylinders and rollers.

- **Sleeve (journal) bearings**

The load is transferred through a thin film of lubricant in a sliding contact

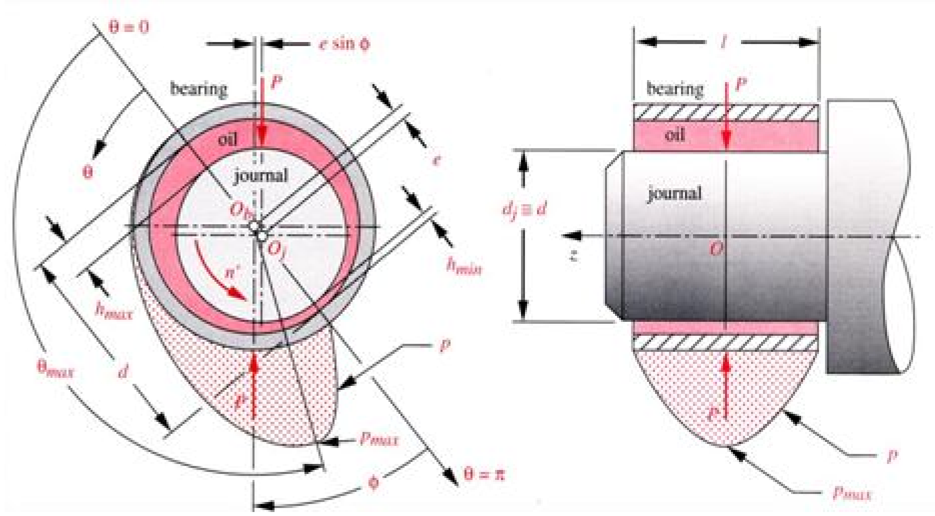

Fig 4.17: A rotating shaft, metal contact and the acting forces. (Source: Shigley's Mechanical Engineering Design, 8th Edition, Budyass Nisbett*)*

Our design was mainly based on rolling elements bearings and they are discussed below. The main advantages of these types as compared to sleeve are:

- They can be used both in low and high temperature applications.
- They are easy to lubricate and consume less lubricant.
- They are internationally standardized, interchangeable and readily obtainable.

The table shown below was used in the selection of the best bearing.

Bearing types / Characteristics	Deep groove ball bearings	Angular contact ball bearings	Double row angular contact ball bearings	Duplex angular contact ball bearings	Self-aligning ball bearings	Cylindrical roller bearings	Single-flange cylindrical roller bearings	Double-flange cylindrical roller bearings	Double row cylindrical roller bearings	Needle roller bearings
Load Carrying Capacity (Radial load / Axial load)										
High speed ❶	☆☆☆☆	☆☆☆☆	☆☆	☆☆☆	☆☆	☆☆☆☆	☆☆☆	☆☆☆	☆☆☆	☆☆☆
High rotating accuracy ❶	☆☆☆	☆☆☆	☆☆	☆☆☆		☆☆☆	☆☆	☆	☆☆☆	
Low noise/vibration ❶	☆☆☆☆	☆☆☆		☆		☆	☆	☆	☆	☆
Low friction torque ❶	☆☆☆☆	☆☆☆		☆☆	☆	☆				
High rigidity ❶			☆☆	☆☆		☆☆	☆☆	☆☆	☆☆☆	☆☆
Vibration/shock resistance ❶			☆		★	☆☆	☆☆	☆☆	☆☆	☆☆
Allowable misalignment for inner/outer rings ❶	☆				☆☆☆	☆				
Stationary in axial direction ❷	◎	○	◎	◎ For DB and DF arrangement	◎		○	◎		
Moveable in axial direction ❸	○		○	○ For DB arrangement	○	◎			◎	◎
Separable inner/outer rings ❹						○	○	○	○	○
Inner ring tapered bore ❺					○	○			○	

Key

❶ ☆ The number of stars indicates the degree to which that bearing type displays that particular characteristic.
★ Not applicable to that bearing type.

❷ ◎ Indicates dual direction. ○ Indicates single direction axial movement only.

❸ ◎ indicates movement in the axial direction is possible for the raceway surface; ○ indicates movement in the axial direction is possible for the fitting surface of the outer ring or inner ring.

❹ ○ Indicates both inner ring and outer ring are detachable.

❺ ○ Indicates inner ring with tapered bore is possible.

Table 4.7: Selection of bearings. (Source: Gears and bearing selection standard tables, SKF Product catalogue, New York)

4.2.3.2 Types of Bearings

NEEDLE BEARINGS

These are roller bearings that have a length that is four times or more the width i.e. their length to diameter ratios are very high. They can carry very high radial loads in small diameters and don't have space between the rollers hence do not require a cage. Their main advantage is the high coefficient of friction which is roughly 0.0025 compared to 0.0015 for roller bearings.

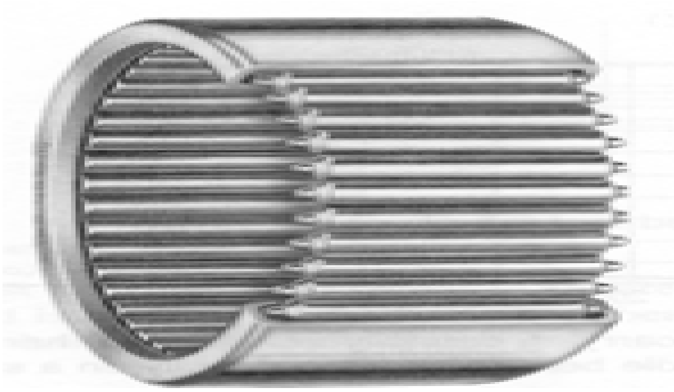

Fig4.18: Needle Roller bearing (Source: Bearing Training Manual, Koyo Corporation USA*)*

BALL BEARINGS

These are made of an inner race and outer race with incorporated hardened steel balls which geometrically have contact with the two races at a point.

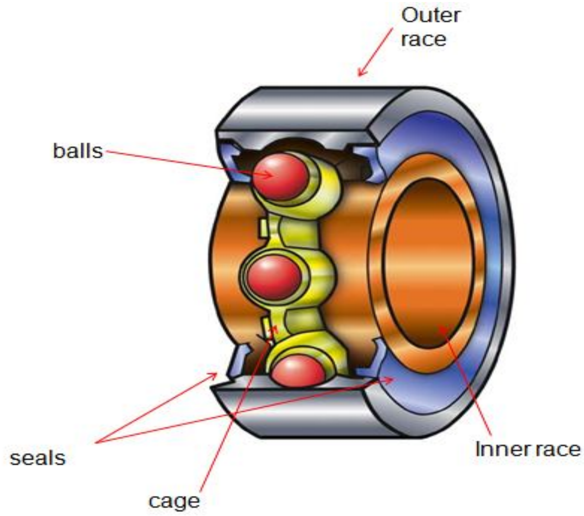

Fig 4.19: Ball Bearing (Source: Bearing Training Manual, Koyo Corporation USA*)*

ROLLER BEARINGS

These have two tracks of races but the balls in ball bearings are replaced by various types of hardened steel cylindrical rollers which may be straight, barrel-shaped or truncated cones. The line of contact deforms into areas larger than ball bearings hence are capable of carrying higher radial loads.

Fig4.20: roller bearing(Source: Bearing Training Manual, Koyo Corporation USA*)*

The bearings were used in the following components

- Main rotor shaft (horizontal) has two roller bearing gears one at the entrance to the gear box and the other at the exit. The roller bearings were used due to their ability to support both radial and axial loads and an increased contact surface area.
- The shaft supporting the driven gears has a pair of roller bearings that are found at the contact between the gear and the shaft.
- The sleeve rod (vertical) has one ball bearing.
- At the base of the turntable i.e. between the turntable and the rig, is one needle thrust bearing that has the ability to support very high loads and also provides a large surface that will prevent slip.
- The actuating rod has one short cylindrical roller bearing. This is to minimize friction on the inner sleeve.

4.3 TAIL DESIGN

The tail of the windmill is on the opposite side if the rotor assembly. The tail has two primary functions:

(i) The first function is to ensure that the rotors are always facing the direction from which the wind is blowing. This ensures optimal interaction with the wind and thus resulting in the rotation of the rotor assembly. The tail is made up of a rigid frame and a flat sheet metal. The frame ensure that the tail can sustain the force exerting by the wind on the assembly. The flat sheet metal is the major component that is responsible for the turning of the tail. It does this by balancing of the lift forces on either of its sides as shown below.

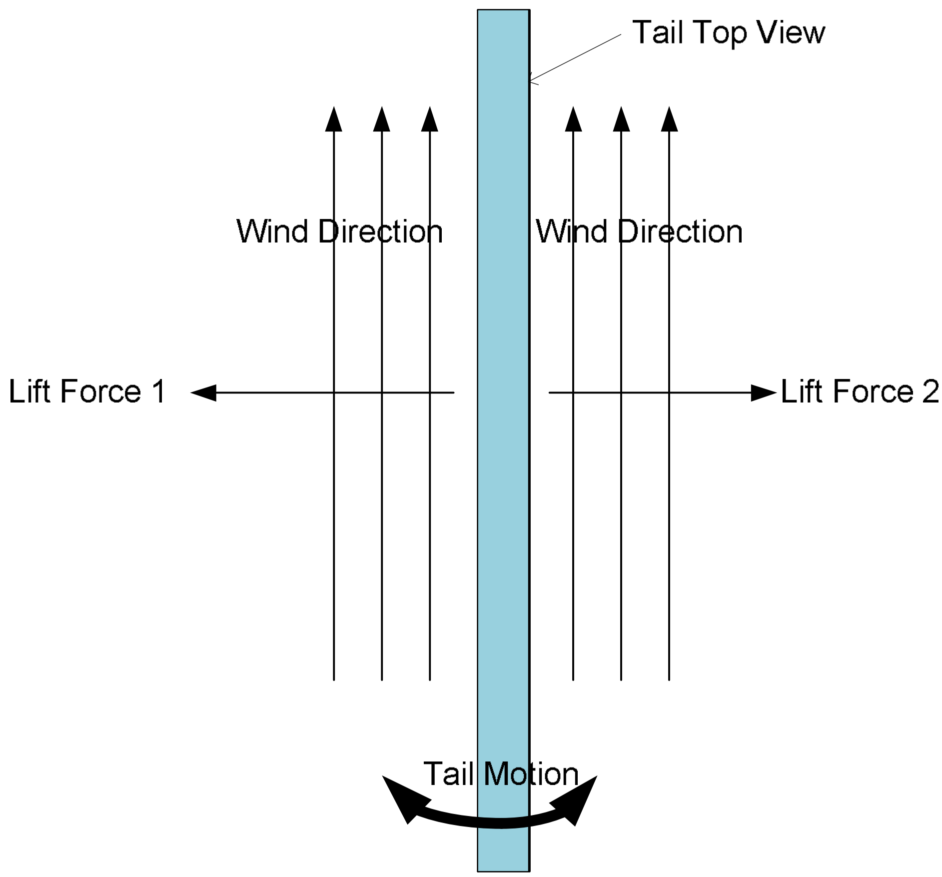

Fig4.21: Illustration of the lift forces on the windmill tail

The winds cause Lift Force 1 and Lift Force 2. If the two lift forces are not equal, the tail turns in the direction of the Tail Motion shown above till it balances. Since it's connected to the gear box, it will consequently turn the rotor assembly till it faces the direction of wind.

(ii) The second function of the tail is to balance the moments that are caused by the rotor assembly. If this moment is not balanced out, it might lead to the eventual failure at the turntable.

To get the ideal dimensions for the tail, calculation of moments in the Windmill structure was done as illustrated below:

4.3.1 Calculations of Moments

The following are the basic equations that govern the calculation of moments.

$$Volume = Cross\ Sectional\ Area \times Breadth$$

$V = A \times b$ m^3 .... (4.3)

$$Mass = Density \times Volume$$

$M = \rho \times V$Kg .... (4.4)

$$Moment = Force \times Distance$$

But: $Force = Mass \times Gravitational\ Acceleration$

$F = mg$ N .... (4.5)

$\therefore Moment = mg \times d$ Nm .... (4.6)

The windmill components that contributed to the bending moments were made of Mild Steel whose density is **7850 kg/m^3**.

4.3.1.1 Anticlockwise Moments

(i) Blades

$$Volume = \frac{214452.152}{1000^2} \times 1.465 \times 10^{-3} \times 24 = 7.5401 \times 10^{-3} m^3$$

$$Mass = 7850 \times 7.5401 \times 10^{-3} = 59.1901\ kg$$

$$Moment = 59.1901 \times 9.81 \times 0.447 = \mathbf{259.5526\ Nm}$$

(ii) Side Blocks (On main shaft)

$$Volume = \frac{14823.416}{1000^2} \times 13 \times 10^{-3} = 1.9270 \times 10^{-4} m^3$$

$$Mass = 7850 \times 1.9270 \times 10^{-4} = 1.5127\ kg$$

$$Moment = (1.5127 \times 9.81 \times 0.491) + (1.5127 \times 9.81 \times 0.39) = \mathbf{13.0737\ Nm}$$

(iii) Inner Ring

$$Total\ Cross\ Sectional\ Area = (31612.223 + 1023.2 + 2234.194 + 4683.324)$$

$$= 39552.941\ mm^2$$

$$Volume = \frac{39552.941}{1000^2} \times 2.5 \times 10^{-3} \times 8 = 7.9106 \times 10^{-4} m^3$$

$$Mass = 7850 \times 7.9106 \times 10^{-4} = 6.2098\ kg$$

$$Moment = 6.2098 \times 9.81 \times 0.447 = \mathbf{27.2305\ Nm}$$

(iv)Outer Ring

$$Volume = \frac{11281.459}{1000^2} \times 0.5 = 5.6407 \times 10^{-4} m^3$$

$$Mass = 7850 \times 5.6407 \times 10^{-4} = 4.4280\ kg$$

$$Moment = 4.4280 \times 9.81 \times 0.447 = \mathbf{19.4170\ Nm}$$

(v) Main Shaft

$$Volume = \frac{507.053}{1000^2} \times 0.504 = 2.5555 \times 10^{-4} m^3$$

$$Mass = 7850 \times 2.5555 \times 10^{-4} = 2.0061\ kg$$

$$Moment = 2.0061 \times 9.81 \times 0.252 = \mathbf{4.9593\ Nm}$$

TOTAL ANTICLOCKWISE MOMENTS:

$$= (259.5526 + 13.0737 + 27.2305 + 19.4170 + 4.9593)Nm$$

$$= \mathbf{324.2331\ Nm}$$

4.3.1.2 Clockwise Moments

(i) Main Shaft

$$Volume = \frac{507.053}{1000^2} \times 0.496 = 2.5150 \times 10^{-4} m^3$$

$$Mass = 7850 \times 2.5150 \times 10^{-4} = 1.9743\ kg$$

$$Moment = 1.9743 \times 9.81 \times 0.248 = \mathbf{4.8032\ Nm}$$

(ii) Sheet Metal

$$Volume = 1 \times 0.5 \times 1.56 \times 10^{-3} = 5.5836 \times 10^{-4} m^3$$

$$Mass = 7850 \times 5.5836 \times 10^{-4} = 4.3831\ kg$$

$$Moment = 4.3831 \times 9.81 \times 1.75 = \mathbf{75.2472\ Nm}$$

(iii) Tail

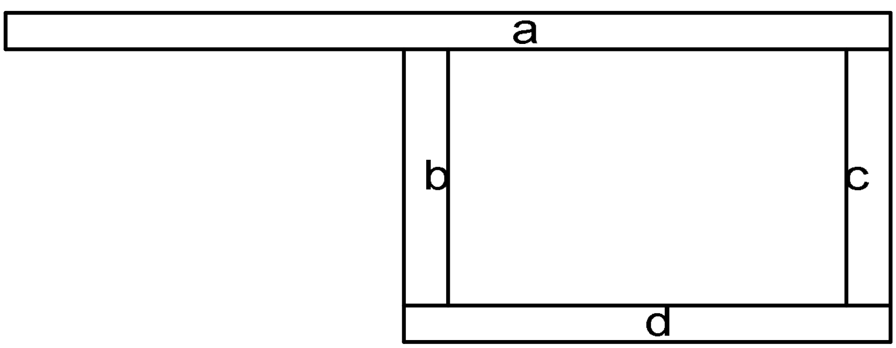

Fig4.22: Sections of the tail that contribute to the bending moments

Member a:

$$Cross\ Sectional\ Area = (0.0508^2) - (0.0445^2) = 6.0039 \times 10^{-4} m^2$$

$$Volume = 6.0039 \times 10^{-4} \times 2 = 1.20078 \times 10^{-4} m^3$$

$$Mass = 7850 \times 1.20078 \times 10^{-4} = 9.4261\ kg$$

$$Moment = 9.4261 \times 9.81 \times 1.25 = \mathbf{115.5876\ Nm}$$

Member b:

$$Volume = 6.0039 \times 10^{-4} \times 0.5 = 3.00195 \times 10^{-4} m^3$$

$$Mass = 7850 \times 1.20078 \times 10^{-4} = 2.3565\ kg$$

$$Moment = 2.3565 \times 9.81 \times 1.25 = \mathbf{28.8970\ Nm}$$

Member c:

$$Volume = 6.0039 \times 10^{-4} \times 0.5 = 3.00195 \times 10^{-4} m^3$$

$$Mass = 7850 \times 1.20078 \times 10^{-4} = 2.3565\ kg$$

$$Moment = 2.3565 \times 9.81 \times 2.25 = \mathbf{52.0145\ Nm}$$

Member d:

$$Volume = 6.0039 \times 10^{-4} \times 1 = 6.0039 \times 10^{-4} m^3$$

$$Mass = 7850 \times 6.0039 \times 10^{-4} = 4.7131\ kg$$

$$Moment = 4.7131 \times 9.81 \times 1.25 = \mathbf{80.9115\ Nm}$$

TOTAL ANTICLOCKWISE MOMENTS:

$$= (4.8032 + 75.2472 + 115.5876 + 28.8970 + 52.0145 + 80.9115) Nm$$

$$= \mathbf{357.461\ Nm}$$

The extra clockwise moment (**33.2279 Nm**) is there to cancel out the moments that are generated by the support fittings and welds that are on the rotor assembly.

4.4 RIG DESIGN

The rig design is based on an existing rig that is of height 5.5 m. However due to safety reasons and rotor assembly dimensions, an extra height of 2 m is required.

In the design of the extra 2 meters, a number of design consideration had to be factored:

(i) The first and most important design consideration is that the rig had to support the weight of the rotor, gear box and tail assembly. To achieve this, the frame of the tower had to be of a metal with the ideal strength properties.

(ii) The rotor, gear box and tail assemblies generated both clockwise and anticlockwise moments. Though the tail primarily balances these moments, the rig design had to be rigid enough to also ensure that it could withstand such moments. This meant that in addition to the material properties, the joining processes that had to be carried out on the rig members had to provide sufficient strength as well.

(iii) The rig had to stand on its own. This implies that the design had to be balanced. Any imbalance would generate a moment that would also weigh down on the base of the rig. Over time, this will result in fatigue failure of members that are experiencing the extra stress.

For stability, three vertical members were used at the edges. They were inclined at an angle of 10^0 from the vertical axis. This results in the cross section reducing up the rig. The wide base increase stability by lowering the center of gravity of the tower. A lower center of gravity decreases the chance of the rig tipping over.

The three 'legs' are made of square section [1] of 1.5 inch and a thickness of ½ inch. The diagonal members are for extra support and they are made of L shape Mild Steel [2] of 1.5 inch sides and thickness 3/16 inches. This minimizes the stresses that might be induced as the rig resists the wind.

The Rig is made of Mild steel with the following properties:

Density (ρ)	7850 kg/m3
Young's Modulus (E)	220 GPa
Poisson's Ratio (ν)	0.275
Yield Strength (σ_y)	207 MPa
Ultimate Tensile Strength (σ_{UTS})	345 MPa

Table 4.8: Mild Steel Properties

[1] ANSI AISC HSS Square 1-1/2 x 1-1/2 x 1/8 – Mild Steel

[2] ANSI L (Equal Angle) 1.5 x 1.5 x 3/16 – Mild Steel

4.5 TURNTABLE

The turntable plays a very crucial role in the entire windmill design. It ensures that it is able to move along the vertical axis. With a proper turntable design, the windmill is able to turn and face the wind thanks to the tail assembly.

In the windmill design, the turntable design is made up of cylindrical sleeves. The outer sleeve is permanent and welded to the top of the rig. There is the inner jacket that is concentric to this sleeve. This inner jacket is welded to the base of the gear box. It has two bearing that ensure its rotation along the vertical axis is possible. There is a needle bearing at the base and a ball bearing at the neck. This ensure minimum friction possible when the rotating to face the wind.

The figure below shows a cross section of the turntable design.

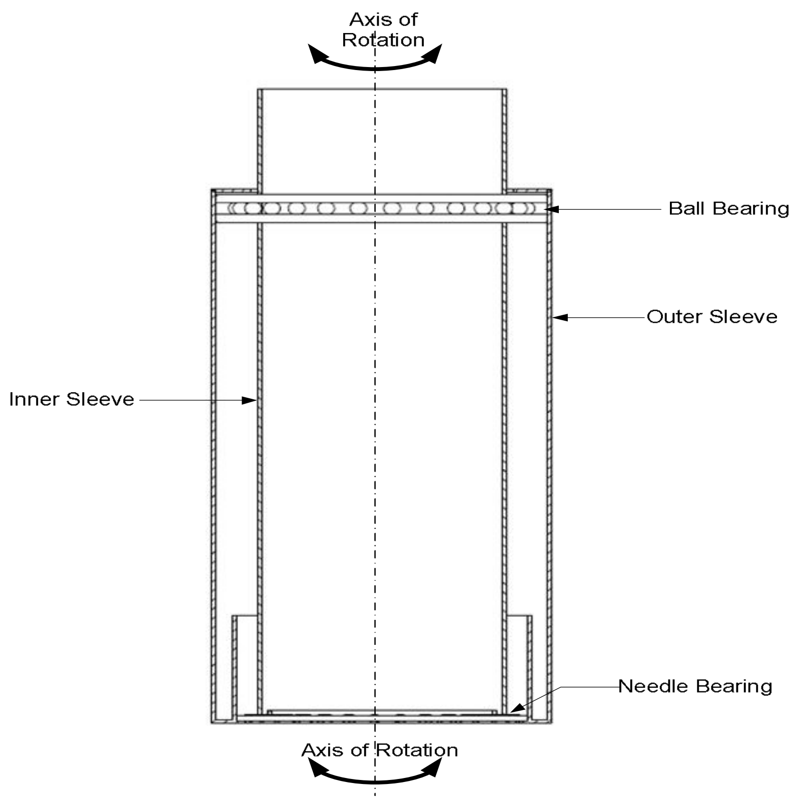

Fig4.23: Cross-Sectional View of the turntable design

The weld to the top of the rig should have some reinforcement in addition to the strong weld. This ensures that the turntable is also able to withstand the stress that might result from the various bending moments that result from the interaction of the windmill and the wind.

4.6 GEAR BOX

The gear box is made of cast iron which is an alloy of iron with carbon content 2% and silicon being more than 0.1 %.

The gear box will house the gear transmission assembly, the actuating mechanism and the pulley transmission mechanism. The movements caused by the gears and the actuation mechanism lead to high vibration intensities hence need for use of a material that will withstand this condition.

The graphite morphology and matrix characteristics of cast iron has a great effect on the physical and mechanical properties. The large graphite flakes in cast iron produce good dampening capacity, dimensional stability, resistance to thermal shock and ease of machining. The small flakes result in high modulus of elasticity and resistance to crazing.

The most critical design specification for selection of cast iron is its ability to be molded into very trivial shapes that are very complex to design using other machining processes.

The figure below shows the gear box with its components and the dimensions

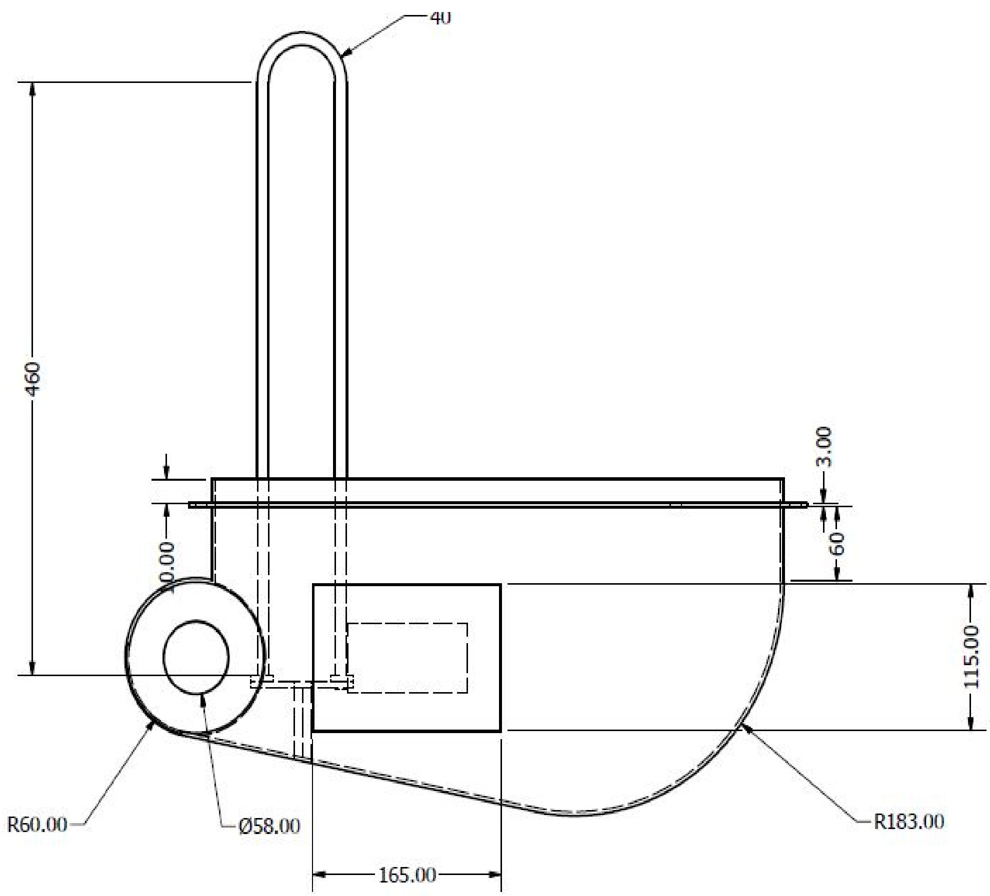

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Fig4.24: Final gear box design

The gears and actuating components are very sensitive to abrasive effects by foreign matter e.g. dust or prolonged exposure to the sun and rain. Therefore a gearbox cover was necessary in our design. It is made of cast iron and fits perfectly on the gear box with the dimensions as shown in the figure below.

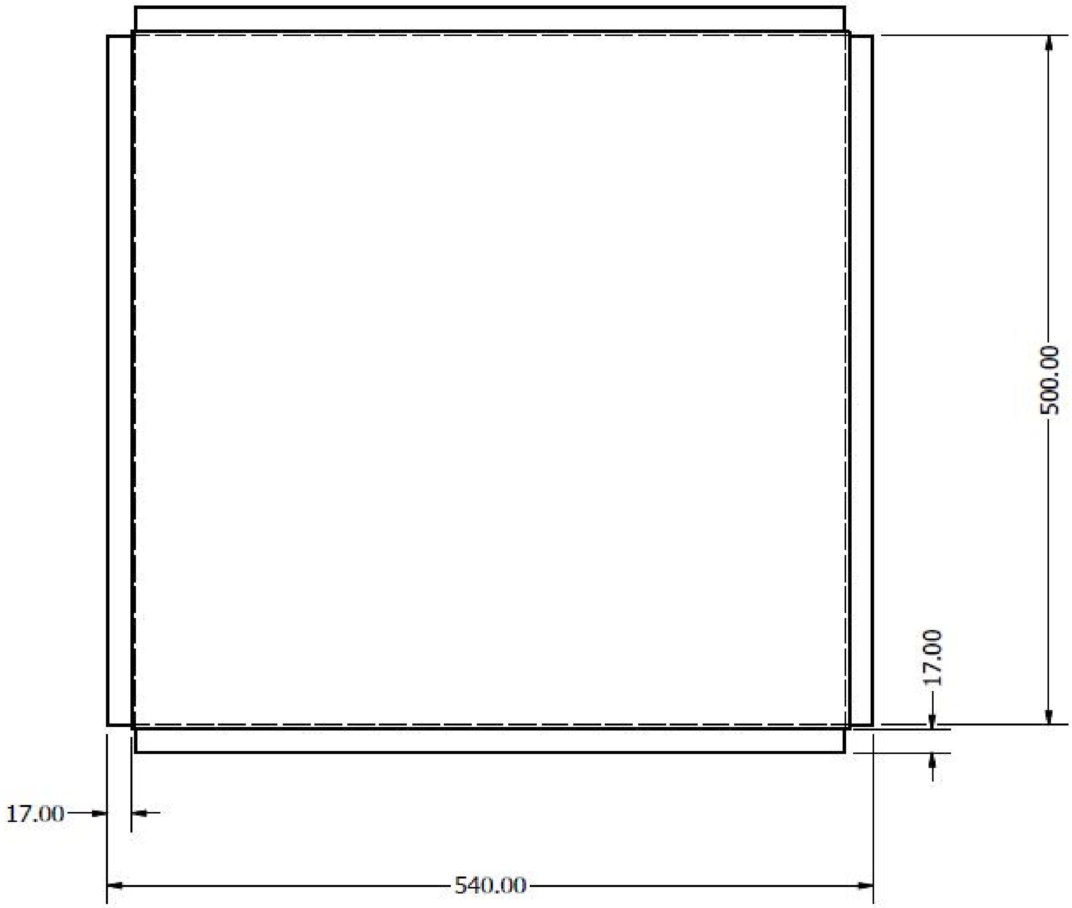

Fig 4.25: Gear box cover design

5.0 DISCUSSION AND CONCLUSION

5.1 TORQUE DEVELOPED

From torque equation:

Torque, T= $\frac{1}{2}\frac{\rho\pi R^3}{\lambda}v^2 C_p$ Nm;

Given:

ρ = 1.207 kg/m^3

R = 2 m

$$\lambda = \frac{4\pi}{N} = \frac{4\pi}{24} = 0.5636$$

V = 3.5 m/s

C_p = 30% = 0.3

Therefore:

$$T = \frac{1}{2} \times \frac{1.207 \times \pi \times 2^3}{0.5236} \times 3.5^2 \times 0.3 = \mathbf{106.4572\ Nm}$$

5.2 SHAFT STRESS ANALYSIS

The main shaft is made of mild steel with the following properties

Density (ρ)	7850 kg/m3
Young's Modulus (E)	220 GPa
Poisson's Ratio (ν)	0.275
Yield Strength (σ_y)	207 MPa
Ultimate Tensile Strength (σ_{UTS})	345 MPa

Table 5.1: Mild steel properties

The shaft is a thin walled tube since the thickness t $< R/_{10}$.

Given:

Ri = 22.4 mm

Ro = 25.4 mm

T = 3 mm

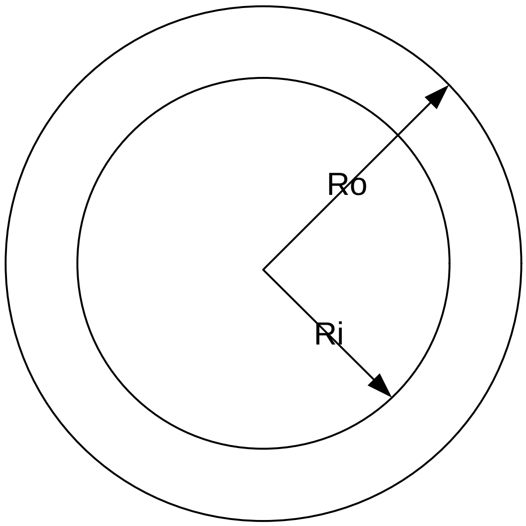

Fig5.1: Main shaft cross section

R_m is the mean radius of the thin walled tube

$$R_m = \frac{25.4 + 22.4}{2} = 23.9\ mm$$

$\tau = \frac{T}{J} R_o$ …………………………………… (5.1)

Where:

τ - Shear Stress (N/m^2)

T – Torque (Nm)

J – Polar Moment of Inertia (m^4)

R_o – Outer Radius (m)

But:

$$J = \frac{\pi}{2}\left({R_o}^2 + {R_i}^2\right)(R_o + R_i)(R_o - R_i) = \frac{\pi}{2}\left(2{R_m}^2\right)(2R_m)t = 2\pi {R_m}^3 t$$

$$J = 2\pi \times (23.9 \times 10^{-3}) \times 3 \times 10^{-3}$$

$$J = 2.5733 \times 10^{-7} m^4$$

Therefore:

$$\tau = \frac{106.4572}{2.5733 \times 10^{-7}} \times 25.4 \times 10^{-3} = \mathbf{10.5092\ \mathit{Mpa}}$$

5.2.1 Moment Generated by the Rotors

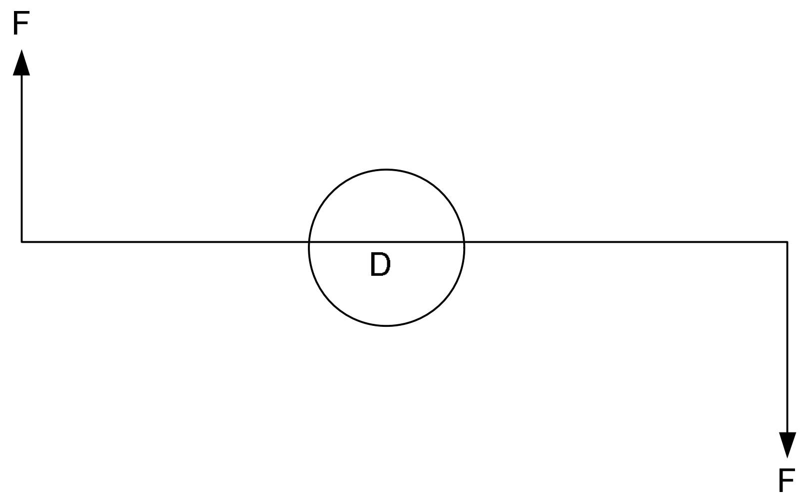

Fig5.2: The Couple resulting from the rotor assembly.

$$F = mg\ N$$

But:

m = 2.4663 kg

g = 9.81 N/kg

Therefore: $F = 2.4663 \times 9.81 = \mathbf{24.1940}\ \boldsymbol{N}$

Moment, M = $FD = 24.1940 \times 4 = \mathbf{96.776}\ \boldsymbol{Nm}$

But remember that there are 12 couples.

Moment of Inertia, I = $\frac{1}{2}J = \frac{1}{2} \times 2.5733 \times 10^{-7} = 1.28665 \times 10^{-7} m^4$

$$\sigma_x = \frac{M}{I}y = \frac{96.776 \times 12}{1.28665 \times 10^{-7}} \times 23.9 \times 10^{-3} = \mathbf{215.718}\ \boldsymbol{Mpa}$$

To get the principal stresses:

$\sigma_{1,2} = \frac{\sigma_x+\sigma_y}{2} \pm \sqrt{\left(\left(\frac{\sigma_x-\sigma_y}{2}\right)^2 + {\tau_{xy}}^2\right)}$ …………………… (5.2)

$$\sigma_{1,2} = \frac{215.718 \times 10^6 + 0}{2} \pm \sqrt{\left(\left(\frac{215.718 \times 10^6 - 0}{2}\right)^2 + (10.5092 \times 10^6)^2\right)}$$

$$\sigma_1 = \mathbf{107.8699}\ \boldsymbol{Mpa}$$

$$\sigma_2 = \mathbf{107.8481}\ \boldsymbol{Mpa}$$

Therefore, the maximum stress is 107.8699 Mpa.

The factor of safety S.F. = $\frac{\sigma_y}{\sigma_{max}} = \frac{207\times10^6}{107.8699\times10^6} = \mathbf{1.9190}$

5.3 DESIGN SPECIFICATIONS OF THE WINDMILL

Rated wind speed	**3.5m/s**
Starting wind speed	**2.5 m/s**
Power Output	**51.09048 W**
Maximum Torque Output	**106.4572 Nm**
Rotor diameter	**4 m (13.12 ft.)**
Number of blades	**24**
Standard tower height	**7.5 m (24.61 ft.)**
Maximum head of water	**30 m (98.43 ft.)**
Maximum water flow rate	**15 m^3 per day (0.1736*10^{-3} m^3/s)**
Solidity	**0.8**
Area of blades	**0.5585 m^2**

Table5.2: Final windmill design specifications

5.4BILL OF QUANTITIES

ITEM	DESCRIPTION	AVAILABILITY	PRICE IN Ksh PER UNIT +VAT	QUANTITIES	TOTAL COST KSH
16 gauge sheet metal	1.2 m width	Apex steel	537.79 per meter	10	5,377.90
0.5" thickness steel plate	8inches by 4 inches	Apex steel	4022.54 per plate	2	8,045.08
0.5" diameter steel rods		Apex steel	1015.56 per meter	18	18,280.80
2" diameter hollow steel pipe	With 3mm thickness	Apex steel	477.17 per meter	1	477.17
Angle block	2.5inches by 2.5inches	Apex steel	430 per meter	20	8,600.00
Lubricant (Gear Oil)		Total Kenya	450 per litre	2	900.00
Spur gear	Driven with 60 teeth 340.50 mm 0utside diameter	Numerical machining complex	2310	2	4,620.00
Spur gear	Driver with 19 teeth and 105mm outside diameter	Numerical machining complex	1094	2	2,188.00
Roller bearing (SKF)	68mm outside diameter and 50 mm inside diameter	Ramco hardware	2120	2	4,240.00
Roller bearings(SKF)	56mm outside diameter and 40m inside diameter	Ramco hardware	3580	2	7,160.00
Ball bearing(SKF)	115mm outside diameter and 90 mm inside diameter	Ramco hardware	3538	1	3,580.00
Thrust needle bearing (SKF)	72 mm outside diameter and 50	SKF Kenya	4230	1	4,230.00

	mm inside diameter.				
Cylindrical roller bearing (SKF)	105 mm outside diameter and 80mm inside diameter	SKF Kenya	4004	1	4,004.00
Fasteners		Metrix hardware	10	200	2,000.00
Fabrication costs		Light Industries (Kariobangi)	15000		15,000.00
				TOTAL	**86,902.95**

Table 5.3: Costing of the various windmill components

5.5 CONCLUSION

From the design, it can be concluded that

- The total torque output of the windmill is **106.4572 Nm** and this is sufficient to sustain the desired flow rate of (**0.1736 X 10^{-3}) m^3 per second** with a maximum head of **30m**, and also to overcome other barriers to motion e.g. friction.
- The number of blades used is **24** with a total surface area of **0.5585 m^2** and this gives a solidity of **0.8**, the minimum (optimum) value of solidity for a windmill and therefore ensures conformity with the standard specifications.
- Gears and bearings are subject to very high heat losses due to friction and this will be greatly minimized by application of oil and grease and therefore greatly improving the efficiency.
- All materials used are locally available and at a low cost making the model economically viable.

5.6 RECOMMENDATIONS

The next phase of the project should be the fabrication and testing of the windmill. The focus should then shift to the effectiveness of the windmill by improving the efficiency and reducing losses that might be in the present design.

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APPENDIX 1: Sheet Metal Thickness and British Standard Wire Gauge, SWG

SWG No	Inches	MM
1	0.300	7.620
5	0.212	5.385
10	0.128	3.251
12	0.104	2.641
14	0.080	2.032
16	0.064	1.626
18	0.048	1.219
20	0.036	0.914
22	0.028	0.711
24	0.022	0.559
26	0.018	0.457
28	0.015	0.376
30	0.0124	0.315
32	0.0108	0.274
34	0.0092	0.234
36	0.0076	0.1930
38	0.0060	0.1524
40	0.0048	0.1219
42	0.0040	0.1016

APPENDIX 2: Parts and Assembly Drawings

Drawing Number	Title/Description
001	Actuating Pulley
002	Actuating Rod 2
003	Actuating Rod 1
004	Actuator Pulley
005	Assembled Shaft
006	Assembly 1
007	Bearing 52610 GOST 8328-75-1
008	Blade Assembly
009	Blade Base Connecting Plate
010	Windmill Assembly 3
011	Blade on Base Plate
012	Blade
013	BS 290 SKF – SKF NA 4910 RS
014	BS 290 SKF – SKF NA 4910 RS SECTION
015	BS 5773_Part 4 80x105x4
016	Complete Turntable
017	Finished windmill
018	Gear box cover
019	Windmill Assembly (Front View of Assembled Windmill)
020	Windmill Assembly 2

021	Gear Box
022	Gear shaft
023	Gear support arm
024	Gears Assembly
025	Main shaft
026	Outer Jacket 1
027	Outer Jacket Assembly 1
028	Pulley Connecting Rod
029	Outer Jacket Section
030	Outer Sleeve Section
031	Windmill Assembly
032	Rig Assembly
033	Rotor Assembly
034	Rotor Base plate
035	Side Blocks
036	SKF 2RZ61818-2RZ
037	SKF 2RZ61818-2RZ_Section
038	Spur Gears
039	Tail End Plate
040	Tail Frame
041	Turntable (Outer)
042	Windmill Assembly (Detail View of Windmill Assembly)
043	Windmill Assembly 1