DESIGN, CONSTRUCTION AND TESTING OF A 130W WIND-POWERED AIR COMPRESSOR SYSTEM FOR OPERATION IN ZARIA, NIGERIA

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Declaration

I declare that the work in this Thesis entitled Design, Construction and Testing of a 130W Wind-Powered Air Compressor System for operation in Zaria, Nigeria has been carried out by me in the Department of Mechanical Engineering. The information derived from the literature has been duly acknowledged in the text and a list of references provided. No part of this thesis was previously presented for another degree or diploma at this or any other institution.

_____ _____

Ahmed Shehu ABDULWAHAB Date

Certification

This thesis entitled DESIGN, CONSTRUCTION AND TESTING OF A 130W WIND-POWERED AIR COMPRESSOR SYSTEM FOR OPERATION IN ZARIA, NIGERIA by Ahmed Shehu ABDULWAHAB meets the regulations governing the award of the degree of M.Eng. (Mechanical Engineering) of the Ahmadu Bello University, and is approved for its contribution to knowledge and literary presentation.

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Abstract

A wind rotor system to power a rotary air compressor of maximum discharge pressure of 3.53 barg, a free air delivery (FAD) of 0.001179 m^3/s at maximum pressure, and a nominal power requirement of 130W was designed, constructed and tested.

The wind air compressor system included a wind rotor, a transmission mechanism, an air compressor and a storage reservoir.

The wind rotor was coupled to a selected air compressor and tested at Ahmadu Bello University, Zaria, Kaduna State, Nigeria.

The compressor discharge pressure and flow rate increased with an increase in the wind velocity. A discharge pressure of 1.0barg was obtained during the testing period at the rated wind velocity of 5.10 m/s. A maximum compressor capacity of 0.000833 m^3/s at a discharge pressure of 2.0 barg and a wind velocity of 7.90 m/s was obtained.

At the rated wind velocity of 5.10 m/s, the power output was calculated as 98.56 W.

The overall efficiency increased as the wind velocity increased until it reached a maximum of 27.6 % and then started to decrease gradually thereafter to a minimum of 7.6 %.The actual efficiency of the system was found to be 24.2% at the rated wind velocity of 5.10m/s compared to the design efficiency of the system of 35 %.

The installed capital costs for the 130W wind air compressor was ₦61,150.00.

Nomenclature

P_a = available power in the wind (W)

ρ_a = air density (kg/m^3)

A = swept rotor area (m^2)

V = wind speed (m/s)

P_m = power (in watts) available from the rotor of the machine

C_p = the (power coefficient) performance of the wind machine.

P_{pn} = pneumatic power (in watts)

$\eta_{overall}$ = overall efficiency of the wind compressor.

$\dot{m}$ = mass flow rate(kg/s)

T_1 = air compressor inlet temperature

T_2 = air compressor outlet temperature

p_1 = air compressor inlet pressure

p_2 = air compressor discharge pressure

$r = \frac{p_2}{p_1}$ = pressure ratio

$\gamma = \frac{C_p}{C_v}$ = ratio of specific heats for air

R = specific gas constant for air(J/kg.K)

η_{comp} = overall compressor efficiency

η_{mech} = mechanical efficiency of the compressor

η_{ad} = adiabatic efficiency of the compressor

B = number of blades

λ = tip speed ratio given by $\frac{r_r \omega}{V}$

r_r = rotor radius

ω = rotor tip rotational speed

d_r = rotor diameter

d_s = shaft diameter

φ = angle of apparent wind

α = angle of attack

T = Twisting moment (torque) acting upon the shaft

M = bending moment

N_i = Speed of the driver (pulley)

d_i = Pitch circle diameter of the driver (pulley)

i = 1 for driver and 2 for driven pulley.

D_i = Diameter of the pulley

CD = Centre distance between pulleys

L = Rating Life of a bearing

C=Basic dynamic load rating of a bearing

W= Equivalent dynamic load of a bearing

k_e = energy pattern factor of a location

$\overline{V^3}$ = mean cube of wind speeds within a selected period

$\overline{V}$ = mean of wind speeds within the same period for a given location

CHAPTER ONE

INTRODUCTION

1.1 Background

Nigeria, a developing nation of 140 million (2006 census) with a growth rate of 3.20 % (Energy Commission of Nigeria, 2013), has an installed production capacity of about 6,000 MW of power as at 2009 (Sambo, 2009). In 2012, the installed production capacity stood at 9,955.4 MW with an average availability of 5, 516.38MW (Energy Commission of Nigeria, 2013). The estimated daily power generation as at December, 2009 was about 3,700MW while the peak load forecast for the same period was 5,103 MW, based on existing connections to the grid (Nigeria Vision 20-2020, 2010). The Business day Newspaper of 21st July, 2009 reported that manufacturers alone require about 2,000MW to power their factories, based on installed capacities as at 2009. The power requirements of the nation have been projected at 28,360MW by 2015 at a modest economic growth of 7% (Sambo, 2009). There is, therefore, a need to bridge this energy gap.

"Fossil" fuel driven machines are used all over the country to augment supply to meet these power requirements. The Energy Commission of Nigeria estimated that fuel driven machines provide about 42% of the power needs of the country between the year 2000 and 2004 (Sambo, 2008). The Nigerian Tribune Newspaper of 21st August, 2009 estimated a daily diesel consumption of 13 million litres. The Vanguard of October, 2010 reported that the Central Bank of Nigeria (CBN) estimated that about $13bn (₦1.989trillion) per annum is expended on diesel for power generation. The use of fossil fuels adds to the carbon emissions in the world with their devastating effect such as global warming and acid rain to mention a few. It is an established fact that fossil fuels are an irreversible source of energy and their supply is depleting. In fact, the continued unrest in the Middle East creates an oil shortfall around the world. From February to April, 2011, crude oil prices have surpassed $120/bbl

with oil prices reaching the $5 mark in the United States. Oil demand increase is expected to be about 40% over the next 20 years. Oil supply increase in the world is not likely to be 50% over the next 30 years (Cambridge Energy Research Associate(CERA), 2011). There is, therefore, a need for reliable, available systems to power light commercial applications such as production of compressed air for pressurizing tyres, powering pneumatic tools for paint spraying, drilling, etc.

The wind pump, used for centuries to lift water, but largely abandoned after the introduction of engine-driven pumps (generally fuelled by diesel or kerosene) and electric pumps, is now being reconsidered as one of several alternative technologies that can be used for these light applications. The classic multi bladed windmill that was a familiar sight in the Great Plains of the US until the 1940s is still being manufactured today. However, engineers have recently begun to make improvements to the design of these pumps, and adapt them for use in developing countries specifically for water lifting.

Automotive industries in Nigeria on the average pressurise one and a half million tyres each year based on registered cars in Nigeria as provided by the Federal Road safety Corps in its 2013 annual report. This is currently achieved using air compressors mostly powered by fossil fuel fired generators. Typical cost for the purchase of the fuel is inhibitive especially with the partial removal of subsidy by the government in December, 2013.. The fuel unavailability in Nigeria is also an inhibiting factor.

In view of the above, the use of renewable energy sources such as wind, solar, biomass, etc to produce “clean” and sustainable energy for domestic uses is considered timely and necessary.

1.2 Statement of the Problem

Hundreds of vulcanisers litter the Nigerian streets today with car air condition compressors converted to air compressors for car tyre pressurising. These compressors are mostly powered by 3 hp engines powered by fossil fuels. A walk around revealed about 20 of these 3 hp

engines in use in surrounding environs (about a geographical area of $4km^2$) of Ahmadu Bello University, Zaria main campus of Samaru. It is clear that these vulcanisers do not have any other source of energy apart from these engines. Each of these engines requires about 6 litres of petrol. The green house emission is estimated at about 300 kg of CO_2 gas per day (1litre of petrol produces about 2.3 kg of CO_2 when burnt) (Lawrence and Thomas, 2011). This is besides the noise emission as well as the economic drain on the country for this domestic use of a "highly" subsidised resource. It is also worthy of note that majority of these vulcanisers cannot even afford these fossil fuel fired compressors.

Equally, other heavy users of compressed air such as car service stations, car assembly factories such as Peugeot Automobile Nigeria (PAN), furniture factories, etc. produce their own enormous quarter of emissions from their fossil fuel powered compressors.

In summary, the present scenario leads to a huge greenhouse gas emission with its devastating effect, noise pollution, and "wastage" of a "highly" subsidized resource.

1.3 The Present Work

Past Works on wind energy utilization reviewed, that involved the use of Compressed air, concentrated on its use as an energy storage medium in an electric power supply grid.

Most of the reviewed works focus on the production of electricity from the wind, which is then converted to mechanical power for air compression using an electric air compressor.

Also none of the works reviewed have used wind energy for air compression for day to day uses to the best of my knowledge.

This work utilised the wind as an energy source to produce "clean" compressed air. The produced compressed air was then stored in reservoirs for sale in "units" of compressed air to various users. The design was for operation in Zaria, Kaduna state.

The focus was on direct conversion of wind energy to mechanical power in the shaft of an air compressor. The losses associated with the conversion first to electric power and then to

mechanical power were avoided. The costs of electrical components necessary for conversion to electric power were also eliminated.

1.4 Aim and Objectives

The aim of this work is to design, construct and test a wind air compression system of 130 W for the purpose of producing compressed air of 3.45 barg in Zaria, Kaduna State.

The specific objectives are:

i. to carry out a design analysis for the wind rotor and a suitable transmission system for the wind compressor system.
ii. to select appropriate materials for the wind compressor system.
iii. to construct and/or procure the components of the system.
iv. to evaluate the performance of the system.
v. to estimate the cost of a prototype of the system.

1.5 Significance of Work

The use of wind power as a replacement for fossil fuels would make Nigeria's energy use greener which would reduce our carbon emissions. This would substantially reduce Nigeria's contribution to global warming.

The burning of fossils fuels also results in the formation of sulphur and nitrogen oxides which are released. These compounds combine with atmospheric moisture to form acids, leading to 'acid rain'. This can lead to destruction of forests and the progressive erosion of rock and masonry structures, both natural and man-made. The use of wind energy as an alternative helps reduce this impact.

This work would reduce noise pollution in the Nigerian society.

It would help to increase Nigeria's industrial output.

It would serve as a source of income for the "compressed air from wind" investors.

It would free the scarce resource of local vulcanisers for more profitable use helping with the governments' poverty reduction program.

The work will reduce domestic use of fossil fuels (a highly subsidised resource in Nigeria) for compressed air production, allowing its use for more pertinent purposes like power production in remote locations for medical purposes example.

CHAPTER TWO

LITERATURE REVIEW

2.1 Wind and Wind Energy

Wind is air in motion caused by the rotation of the earth and the uneven heating of the atmosphere by the sun. Wind flow is caused by the differing temperature profile of air around the earth, together with the effect of the earth's rotation. Wind patterns vary according to seasonal, diurnal, and landscape variability and due to changing wind patterns, physical obstacles, in the area, slope, etc. Figure 2.1 gives a global air temperature profile. Wind is also caused by a phenomenon known as land and sea breeze. The land and the sea have different heating coefficients and therefore retain heat differently. During the day, the sea is cooler and hence a temperature differential is therefore created, which results in a breeze (wind) moving from the sea to the land. In the night time the reverse is the case and the breeze moves from the land to the sea. Figure 2.2 illustrates this phenomenon.

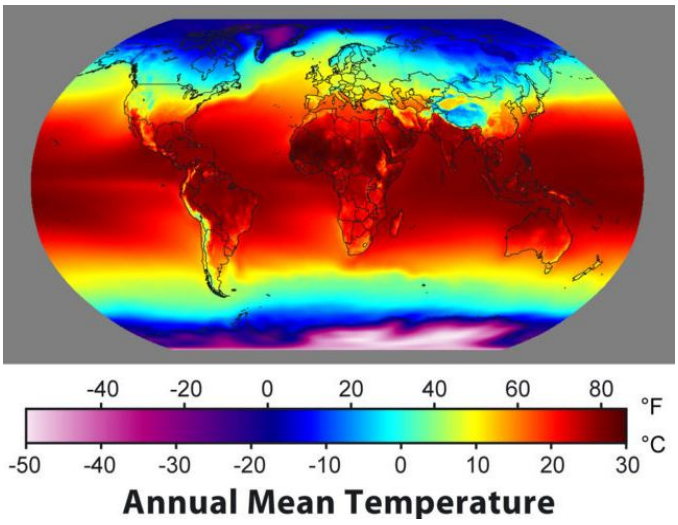


Figure 2.1: Global Air Temperature Profile

Source: http://www.forumbiodiversity.com

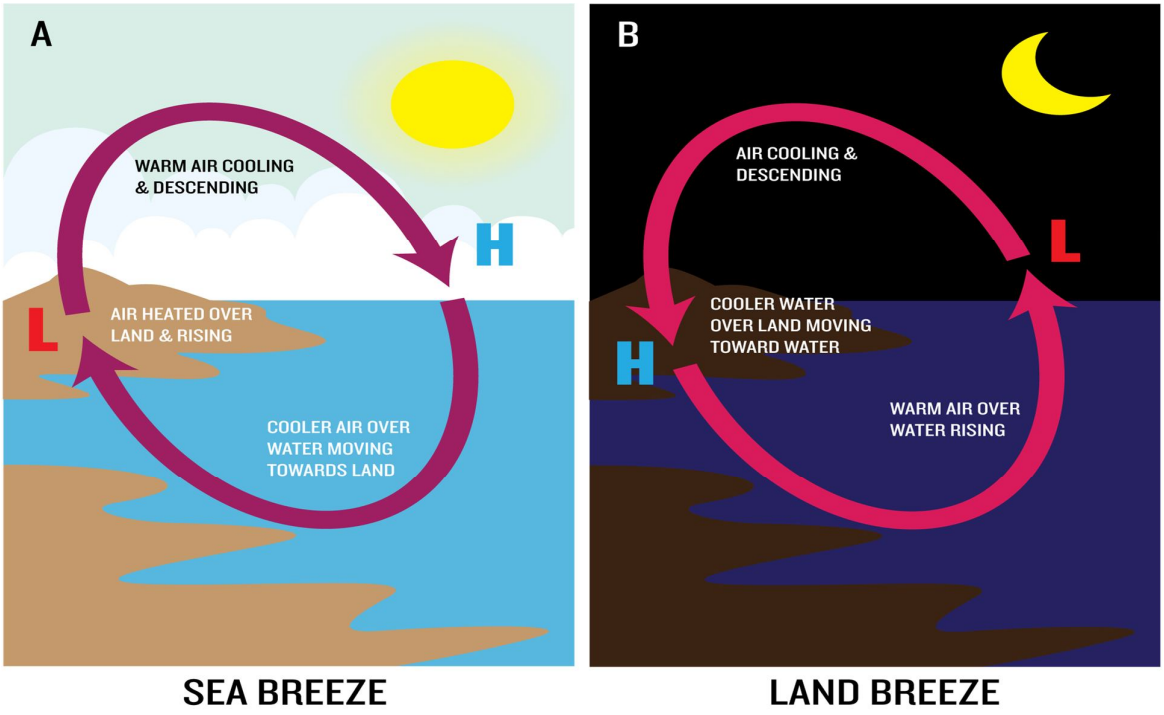


Figure 2.2: Land and Sea Breeze

Source: https://sites.google.com/site/earthscienceinmaine/air-movement

2.1.1 The History of Wind Energy

The wind has played a long and important role in the history of human civilisation. The first known use of wind dates back 5,000 years to Egypt, where boats used sails to travel from shore to shore. The first true windmill, a machine with vanes attached to an axis to produce circular motion, may have been built as early as 2000 B.C. in ancient Babylon. By the 10th century A.D., windmills with wind-catching surfaces as long as 16 feet and as high as 30 feet were grinding grain in the area now known as eastern Iran and Afghanistan. Windmills were modified and used in the 12th century to pump water and reclaim much of Holland from the sea.

Figure 2.3 shows a picture of an ancient windmill off the Mediterranean coast.



Figure 2.3: An early sail-wing horizontal-axis mill on the Mediterranean coast

Source: (Dodge, 2006)

The familiar multi-vane "farm windmill" of the American mid west and west was invented in the United States during the latter half of the l9th century. In 1889, there were 77 windmill factories in the United States, and by the turn of the century, windmills had become a major American export. Until the diesel engine came along, many transcontinental rail routes in the United States depended on large multi-vane windmills to pump water for steam locomotives.

Farm windmills are still being produced and used, though in reduced numbers, and show no sign of becoming obsolete. They are best suited for pumping ground water in small quantities to livestock water tanks. Without the water supplied by the multi-vane windmill, beef production over large areas of the West of the United States would not be possible.

In the 1930s and 1940s, hundreds of thousands of electricity producing wind turbines were built in the United State. They had two or three thin blades which rotated at high speeds to drive electrical generators. These wind turbines provided electricity to farms beyond the reach of power lines and were typically used to charge storage batteries, operate radio receivers and power a light bulb or two. By the early 1950s, however, the extension of the central power grid to nearly every American household, via the Rural Electrification Administration, eliminated the market for these machines. Wind turbine development lay nearly dormant for the next 20 years (Dodge, 2006)

Following the Organisation of Petroleum Exporting Countries (OPEC) Oil Embargo of 1973, interest in wind energy resurfaced in response to climbing energy prices and questionable availability of conventional fuels (Dodge, 2006). Federal and state tax incentives and aggressive government research programmes triggered the development and use of many new wind turbine designs. Some experimental models were very large. With a blade diameter of 92 metres, a single machine was able to supply enough electricity for 700 homes. A wide variety of small-scale models also became available for home, farm and remote uses.

In the 1970s there were nearly 50 domestic wind turbine manufacturers in the United States(Dodge, 2006). Since then, the wind industry has undergone massive consolidation, resulting in less than a dozen domestic manufacturers in 1997. Roughly half of these deal exclusively with small-scale models. This consolidation followed the expiration of the tax incentives in the mid-1980s and the easing of the energy crisis, both of which reduced market demand. A competitive marketplace to weed out inferior products further contributed to consolidation.

Meanwhile, a new market for wind systems, "wind farms," began in the early 1980s in the US. This market evolved thanks in part to a new Federal law, the Public Utility Regulatory Policies Act of 1978. This legislation requires utilities to buy electricity from private, non-

utility individuals and developers. California has been home to most wind farm development due to very attractive electricity buy-back rates and the availability of windy, sparsely populated mountain passes. According to the California Energy commission, 1.5 % of California's total electricity is generated by the wind as at 2004. As the cost of the technology has continued to decline, other areas of the United States, namely the Great Plains, Pacific Northwest and Northeast, are now beginning to see greater wind farm development.

2.2 Review of Related Past Work

Abed (1997) studied a system consisting of a wind turbine, compressor, storage tank and an air lift pump. The characteristics of the air lift pump were investigated by using a numerical model. The author found that wind turbines with compressed air storage and capacity factors greater than 40 % are feasible with air lift pumps. He also found that the efficiency of the system reaches 22 % when the ratio of water to air flow rates equals 2.15 and decreases thereafter.

Benet al (2003) designed a hybrid wind powered system that includes a compressed air storage system. The system is designed for the storage of wind energy and utilises above-ground compressed-air storage with 8,278 kPa (1,200-psia) tanks. The design is most economic for between 0.5 and 100 Megawatt electric power systems. The authors were able to show that utilising the wind energy storage in the hybrid power system reduced the cost of delivered electricity from USD 0.076/kWh to USD 0.057/kWh.

Bragg and Schmidt (1979) outlined a procedure, which helps with matching a of water pumps with windmills to give optimum wind pumping system performance. Dimensional analysis techniques were used to predict the most suitable type of pump for the water lifting applications, the best type of wind machine for the design wind speed and the final

configuration to maximise total system performance. The design and off-design performance of the complete system was also predicted.

Georges and Henrik (2008) analysed the energy-balance effects of adding Compressed-Air Energy-Storage (CAES) to the Western Danish energy-system. Results show that even with an unlimited CAES plant capacity, excess power production is not eliminated because of the high percentage of Combined Heat and Power plants production. The optimal wind-power penetration for maximum CAES operation is found to be around 55 %. The minimum storage size for CAES to fully eliminate condensing power plants operation in the optimised system is over 500 GWh, which corresponds to a cavern volume of around 234 million m^3 at an average pressure of 60 bar. Such a storage size would be technically and economically unfeasible. The analysis, however, did not include the potential role of a CAES plant in regulating the power services.

Isyaku (2011) designed, constructed and tested a 700W horizontal axis wind turbine for household use in Sabon Gida village Kano, Nigeria. The machine was designed based on 5 year wind data and was to operate at 10 metres height at a wind speed of 6.225 m/s in Kano, Nigeria.

The machine developed was tested on the roof of Mechanical Engineering Department, Ahmadu Bello University Zaria, Kaduna, Nigeria at 22 meters height and average wind speed of 6.15 m/s. The machine achieved an efficiency of 21.7 %.

Jia et al (2004) put forward a new approach to application of wind energy by using compressed air power to some extent instead of electricity. The authors presented a theoretical and computational simulation of the wind energy- compressed air power system. The authors believe the compressor capacity versus wind power relationship put forward

would be useful in the design of the wind power- compressed air system. The results of the investigations on the application of high-pressure compressed air for pressure reduction led to the conclusion that pressure reduction with expander is better than the throttle regulator in energy saving.

Mathew and Pandey (2000) proposed a method for predicting the output of simple multi bladed wind mill mechanically coupled to a reciprocating pump. The method adopts an integrated approach by considering the interaction between the rotor, pump and the wind regime to model the wind pump discharge.

The expected outputs at sites from the southern part of India were predicted using the proposed method with wind data from the sites.

Mathew and Pandey (2002) proposed a mathematical model for estimating the performance of wind driven roto-dynamic pumps at various operating conditions. The model was validated using the field performance data from a 5m, five bladed experimental rotor coupled with a low speed centrifugal pump.

Performance of the system at fluctuating conditions of wind regimes was estimated and compared with that of the conventional system with reciprocating pumps.

The authors recommended a low speed roto-dynamic pump with reasonable specific diameter for coupling with wind rotors for water pumping applications.

2.3 Theoretical background

2.3.1 Available wind power

The power in the wind is proportional to the area of windmill being swept by the wind, cube of the wind speed and the air density, which varies with altitude. The power in the wind is given by (Manwell et al, 2002):

$$P_a = \frac{1}{2}\rho_a A V^3 \tag{2.1}$$

where,

P_a = available power in the wind (W)

ρ_a = air density (kg/m^3)

A = swept rotor area (m^2)

V = wind speed (m/s)

2.3.2 Extractable wind power

Although equation (2.1) gives us the power in the wind, the actual power that can be extracted from the wind is significantly less due to losses in the system.

There are physical limits to the amount of power which can be extracted realistically from the wind. It can be shown theoretically that any windmill can only possibly extract a theoretical maximum of 59.3 % of the power from the wind, this is known as the Betz limit (Baumeister, 1966). In reality, for a wind pump, this figure is usuallyaround 45 % maximum (Magdi*et al,* 2011).

The extractable wind power, which is also the rotor power, is given by (Argaw, 2001):

$$P_m = \frac{1}{2}\rho_a C_p A V^3 \tag{2.2}$$

where,

P_m = power (in watts) available from the rotor of the machine

C_p = power coefficient of the wind machine

Since the wind air compressor does not utilise all the energy from the wind due to mechanical losses, the actual useful power at the compressor shaft would be less.

For a wind pump, the useful power, called the hydraulic power, P_h is given by (Argaw, 2001):

$$P_h = \frac{1}{2}\eta_{overall}\rho_a A V^3 \tag{2.3}$$

For an air compressor, the equivalent useful power at the shaft, called the pneumatic power would be:

$$P_{pn} = \frac{1}{2}\eta_{overall}\rho_a A V^3 \tag{2.4}$$

Where

P_{pn} = pneumatic power (in watts)

$\eta_{overall}$ = overall efficiency of the wind compressor

2.3.3 Compressor power

The adiabatic head for low pressure air compressors is given by (Gresh, 2001);

$$H_{ad} = RT_1\left(\frac{\gamma}{\gamma-1}\right)\left(r^{\left(\frac{\gamma-1}{\gamma}\right)}-1\right) \tag{2.5}$$

The power needed to compress a given quantity of air is given by (Gresh, 2001);

$$P_{ad} = \frac{H_{ad}\dot{m}}{\eta_{ad}} \tag{2.6}$$

Substituting equation (2.5) into (2.6) gives:

$$P_{ad} = \frac{\dot{m}RT_1}{\eta_{ad}}\left(\frac{\gamma}{\gamma-1}\right)\left(r^{\left(\frac{\gamma-1}{\gamma}\right)}-1\right) \tag{2.7}$$

This is equivalent to the pneumatic power. Hence equation (2.7) can be written as:

$$P_{pn} = \frac{\dot{m}RT_1}{\eta_{comp}}\left(\frac{\gamma}{\gamma-1}\right)\left(r^{\left(\frac{\gamma-1}{\gamma}\right)}-1\right) \tag{2.8}$$

Where,

$$\eta_{comp} = \eta_{ad} \times \eta_{mech} \tag{2.9}$$

The adiabatic efficiency, η_{ad} is defined as (Gresh, 2001):

$$\eta_{ad} = \frac{T_1\left(r^{\left(\frac{\gamma-1}{\gamma}\right)} - 1\right)}{T_2 - T_1} \tag{2.10}$$

where,

$\dot{m}$ = mass flow rate (kg/s)

T_1 = air compressor inlet temperature

T_2 = air compressor outlet temperature

p_1 = air compressor inlet pressure

p_2 = air compressor discharge pressure

$r = \frac{p_2}{p_1}$ = pressure ratio

$\gamma = \frac{C_p}{C_v}$ = ratio of specific heats for air

R = specific gas constant for air(J/kg.K)

η_{comp} = overall compressor efficiency

η_{mech} = mechanical efficiency of the compressor

η_{ad} = adiabatic efficiency of the compressor

2.4 Wind Air Compressor System Description

A wind air compressor system consists of the following components:

i. Wind rotor

ii. Transmission system

iii. Air compressor

iv. Storage reservoir

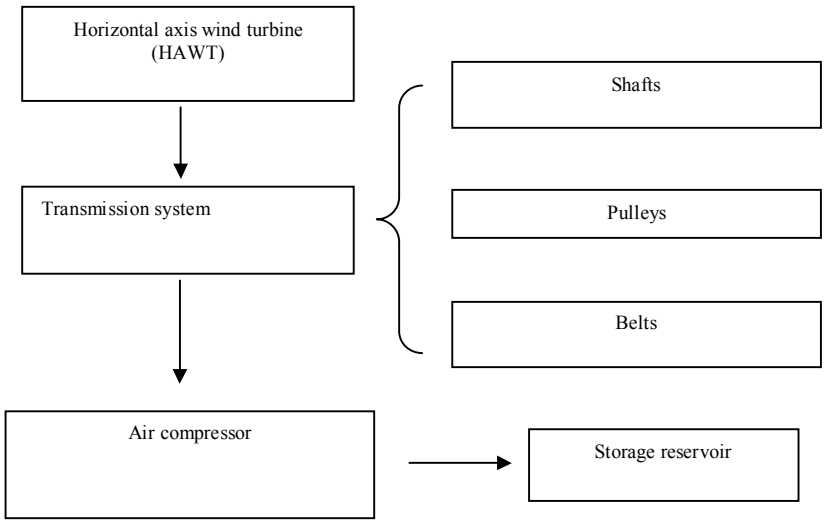


Figure 2.4: Overview of a wind air compressor system

2.4.1 The wind rotor

The rotor is the part of the wind turbine that extracts energy from the wind. It usually consists of two or more blades made from wood, fibreglass or metal, which rotate about an axis (horizontal or vertical). The rotor rotates at a rate determined by the wind speed and the shape of the blades. The blades are attached to the hub, which in turn is attached to the main shaft.

2.4.1.1 Rotor design

Theory

Blade designs operate on either the principle of drag or lift. For the drag design, the wind literally pushes the blades out of the way. Drag powered wind turbines are characterised by slower rotational speeds and high torque capabilities. They are useful for pumping, sawing or grinding work. The lift blade design employs the same principle that enables airplanes, kites and birds to fly. The blade is essentially an aerofoil, or wing. When air flows past the blade, a wind speed and pressure differential is created between the upper and lower blade surfaces. The pressure at the lower surface is greater and thus acts to "lift" the blade. When blades are attached to a central axis, like a wind turbine rotor, the lift is translated into rotational motion. Lift-powered wind turbines have much higher rotational speeds than drag types and therefore well suited for electricity generation.

Windmill rotors in most cases rotate about a horizontal axis (that is the rotor shaft is horizontal). Some designs, however, have rotors that rotate about a vertical axis. The advantage of the vertical axis rotors is that the need does not exist to orient them to face the wind since they present the same cross section to the wind from any direction. However, a serious disadvantage is that the vertical axis rotor cannot be turned away from the wind under stormy conditions to reduce the wind loadings on it. The rotor must therefore have to be designed to withstand the most severe wind speeds in the chosen location.

Design

For the design of horizontal-axis wind rotors, in which lift forces are the driving forces, the following two steps are important:

i. the choice of basic parameters such as number of blades, the radius of the rotor, the type of airfoil, the tip speed ratio and,

ii. the determination of the blade setting angle β and the chord c at each position along the blade.

Number of blades

The number of blades is best calculated from (Hugh, 2008)

$$B = \frac{80}{\lambda^2} \tag{2.11}$$

Where

B = number of blades

λ = tip speed ratio given by $\frac{r_r \omega}{V}$

r_r = rotor radius

ω = rotor tip rotational speed

V = wind speed

Width of the blades

The blade width is obtained from (Hugh, 2008)

$$C = \frac{4d_r}{\lambda^2 B} \tag{2.12}$$

Where

d_r = rotor diameter

Blade setting angle

The blade setting angle is given by (Hugh, 2008):

$$\beta = \varphi - \alpha \tag{2.13}$$

Where

φ = angle of apparent wind

α = angle of attack

$$\tan\varphi = \frac{\left(2V/3\right)}{\left(\frac{\tau}{r_r}\right)\gamma V} = \frac{2r_r}{3r\lambda} \tag{2.14}$$

Therefore,

Blade angle β is (Hugh, 2008)

$$\beta = \tan^{-1}\left(2r_r/3r\lambda\right) - \alpha \tag{2.15}$$

Where

r_r = rotor radius

V= wind speed

r = radius at particular section of the blade. At the tip, $r = r_r$

Airfoil chord

The chord is calculated from the relation (Lysen, 1983):

$$c = \frac{8\pi r_r}{BC_{ld}}(1 - \cos\varphi) \tag{2.16}$$

Width of the blades or Chord (C_w)

The blade design process specifies the chord width at various stations along the span. The chord length is given by (Piggot, 2003):

$$C_w = \frac{16\pi r_r (\frac{r}{r_r})}{9\lambda^2 B} \tag{2.17}$$

Tip Speed Ratio

The tip-speed is the ratio of the rotational speed of the blade to the wind speed. The larger this ratio, the faster the rotation of the wind turbine rotor at a given wind speed. Electricity generation requires high rotational speeds. Lift-type wind turbines have maximum tip-speed ratios from 4 to 10, while drag-type ratios are between 1 and 2(Lysen, 1983).

The number of blades that make up a rotor and the total area they cover affect wind turbine performance. For a lift-type rotor to function effectively, the wind must flow smoothly over the blades. To avoid turbulence, spacing between blades should be great enough so that one blade will not encounter the disturbed, weaker air flow caused by the blade which passed before it. It is because of this requirement that most wind turbines have only two or three blades on their rotors.

The tip speed ratio is given by (Hugh, 2008)

$$\lambda = \frac{r_r \omega}{V} \tag{2.18}$$

Angle of attack (α), Lift coefficient (C_l)

The values of α and C_l at minimum drag-lift ratio, $C_d/C_l$are important parameters in the design process.

Typical values of C_d/C_l and the angle of attack, α and lift ratio C_l for a number of airfoils is given in table 2.1.

Table 2.1: Typical values of the drag-lift ratio C_d/C_l and the angle of attack, α and lift ratio C_l for a number of airfoils

AIRFOIL	SHAPE	C_d/C_l	α	C_l
Flat plate		0.1	5^o	0.00
Curved Plate (10% curvature)		0.02	3^o	1.25
Curved Plate with tube on concave side		0.03	4^o	1.10
Curved Plate with tube on convex side		0.2	14^o	1.25
Airfoil NACA 4412		0.01	4^o	0.80

Source: (Lysen, 1983)

The maximum power co-efficient

It has been shown by Betz, that the maximum power coefficient is equal to 16/27 or 59.3%. This, however, is the power coefficient of an ideal wind rotor with an infinite number of zero drag blades.

In practice there are three effects which cause a further reduction in the maximum attainable power coefficient:

i. The rotation of the wake behind the rotor

ii. The finite number of blades

iii. C_d/C_l ratio is not zero.

Rotor Design Steps

To design a rotor, the value of the chord, c and the setting angle β of the blades (Fig 2.5) at a number of positions along the blade must be obtained.

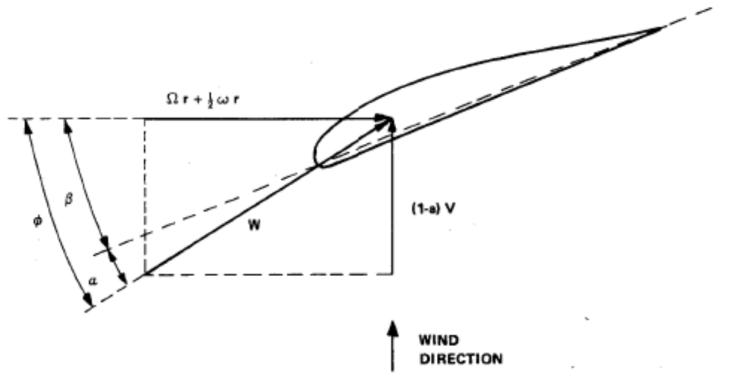


Fig. 2.5 The angle of attack and the setting angle β of the blade of a wind rotor

Source: (Lysen, 1983)

The values of the parameters in table 2.2 were chosen beforehand:

Table 2.2: Parameters of Rotor and Airfoil

Rotor	r_r	Radius
	λ_d	Design tip speed ratio
	B	Number of blades

Airfoil	C_{ld}	Design lift coefficient
	α_d	Corresponding angle of attack

The radius of the rotor was calculated with the required energy output in a period (or in critical month), given the average wind speed and its distribution.

The choices of λ_d and B are more or less related as the guidelines in Table 2.3 suggests:

Table 2.3 Guidelines for the choice of the design tip speed ratio and the number of blades

λ_d	B
1	6 - 20
2	4 - 12
3	3 - 6
4	2 - 4
5 - 8	2 - 3
8 – 15	1 - 2

Source: (Lysen, 1983)

The type of load will determine λ_d: water pumping windmills driving piston pumps have $1 < \lambda_d < 2$ and electricity generating wind turbines usually have $4 < \lambda_d < 10$(Lysen, 1983).

The airfoil data are selected from table 2.1.

The chord is calculated from equation (2.17)

The rotor parameters could be selected from known designs such as in figure 2.6.

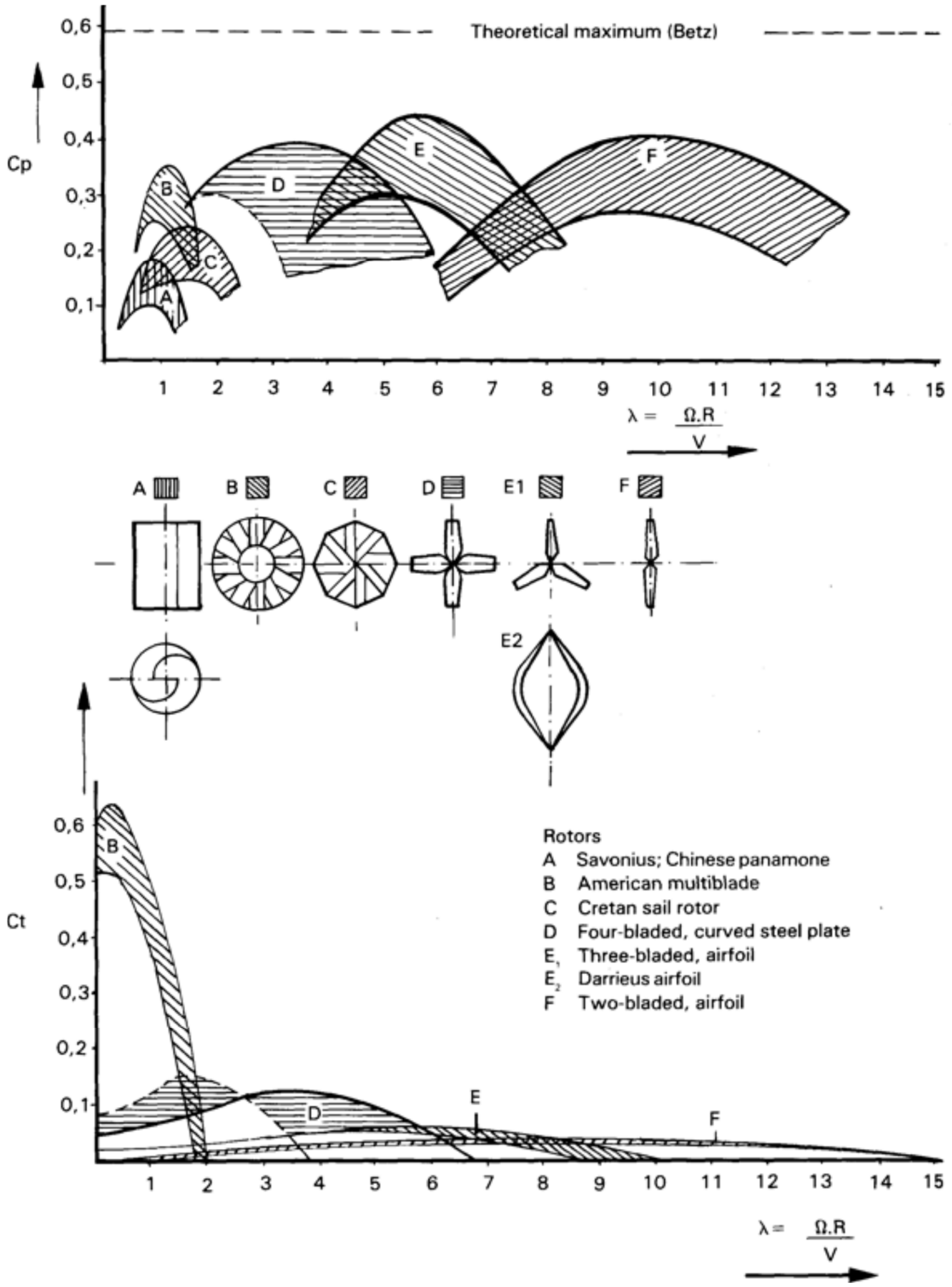


Fig 2.6 Power/Torque coefficient versus tip speed ratio for different types of rotors

Source: (Frankael, 1986)

The rotor blade could be designed in two formats:

A. Constant lift coefficient - the chord and the setting angle continuously varies along the blade. This is relatively difficult to manufacture.

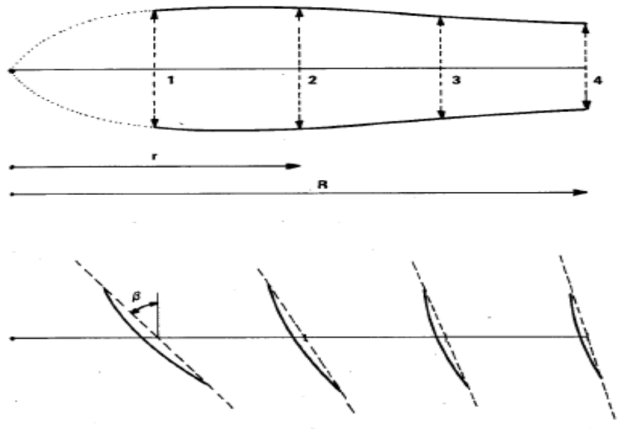


Fig. 2.7 Constant lift coefficient blade

Source: (Lysen, 1983)

B. Constant chord blade- a deviation of the constant lift coefficient blade to ensure ease of manufacture.

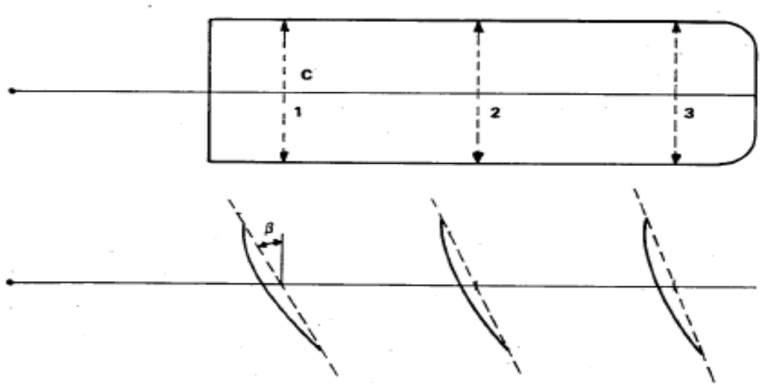


Fig 2.8 Constant chord blade

Source: (Lysen, 1983)

2.4.1.2 Material Selection

Plastic Materials

Plastic materials (Polyethylene, Low density (LDPE) and High density (HDPE)) have the following properties:

i. Density of 0.925-0.959 g/cm^3
ii. Modulus of elasticity of 0.172- 0.282 GPa.
iii. A Poisson ratio of 0.33-0.40
iv. A yield strength and Tensile strength of 9.0-33.1 MPa and 8.3-31.4 MPa respectively.

In addition they have the following characteristics:

i. Low maintenance
ii. Wear resistance when running dry
iii. Low noise
iv. Vibration dampening
v. Corrosion-proof
vi. Low inertia due to low rotating mass, light weight
vii. Low manufacturing cost
viii. Susceptible to wear and tear

Wood

Wood is a hygroscopic, anisotropic material of biological origin. The properties of wood are as follows:

i. Density of 0.46-0.67 g/cm^3
ii. Modulus of elasticity of 0.54-13.6 GPa.
iii. A yield strength and Tensile strength of 2.4-112 MPa and 0.35-16.3 MPa respectively.

In addition wood has these additional properties:

i. Wood is a very strong material in compression parallel to the grain.

ii. Low manufacturing cost

iii. Low inertia due to low rotating mass, light weight

iv. Corrosion-proof

v. Susceptible to wear and tear

Polyvinyl chloride (PVC)

Polyvinyl chloride (PVC) is a polymer with the following properties:

iv. Density of 1.30-1.58 g/cm^3

v. Modulus of elasticity of 2.41-4.14 GPa.

vi. A Poisson ratio of 0.38

vii. A yield strength and Tensile strength of 40.7-44.8MPa and 40.7-51.7MPa respectively.

Materials used for shafts

The material used for shafts have the following properties:

i. High strength.

ii. Good machinability.

iii. Good heat treatment properties.

iv. High wear resistant properties.

v. Low notch sensitivity factor.

The materials used for ordinary shafts are carbon steel of grades AISI 1045, 1050, 1060 and 1080.The mechanical properties of these grades of carbon steel are given in table 2.4

Table 2.4 Mechanical properties of steels used for shafts

SN	AISI standard designation	Ultimate tensile strength, MPa	Yield strength, MPa
1	1045	570-630	310 - 530
2	1050	620-690	340 - 580
3	1060	680	370
4	1080	770	420

Source: (Budynas and Nisbeth, 2011)

When a shaft of high strength is required, an alloy of steel such as nickel, nickel-chromium or chrome-vanadium can be used.

Materials used for belts

The materials used for belts include leather, canvass, rubber, balata, etc. A belt could be single or double woven. The choice of a belt material is dependent on its strength, mass density and the level of friction between the belt material and the pulley material. Tables 2.5and 2.6 provide the density of belt materials and the coefficient of friction between belt and pulley respectively for selected materials.

Table 2.5 Density of Belt materials

SN	Belt Material	Mass density in kg/m^3
1	Leather	1000
2	Canvass	1220
3	Rubber	1140
4	Balata	1110
5	Single woven belt	1170
6	Double woven belt	1250

Source: (Khurmi and Gupta, 2008)

Table 2.6 Coefficient of friction between belt and pulley

SN	Belt material	Pulley material						
		Cast iron steel			Wood	Compressed paper	Leather face	Rubber face
		Dry	Wet	Greasy				
1	Leather oak tanned	0.25	0.20	0.15	0.30	0.33	0.38	0.40
2	Leather chrome tanned	0.35	0.32	0.22	0.40	0.45	0.48	0.50
3	Canvass stitched	0.20	0.15	0.12	0.23	0.25	0.27	0.30
4	Cotton woven	0.22	0.15	0.12	0.23	0.25	0.27	0.30
5	Rubber	0.30	0.18	-	0.32	0.35	0.40	0.42
6	Balata	0.32	0.20	-	0.35	0.38	0.40	0.42

Source: (Khurmi and Gupta, 2008)

2.4.2 Shaft Design

The design of a wind turbine shaft is based on the two moments to which it is subjected to- the twisting and bending moments. Various models have been suggested to account for the elastic failure of materials when they are subjected to various types of combined stress. (Khurmi and Gupta, 2008).

The following theories are important to the shaft design for use in a wind turbine:

i. Maximum shear stress theory (Guest's theory), which is used for ductile materials such as mild steel.
ii. Maximum normal stress theory (Rankine's theory), which is used for brittle materials such as cast iron.

Let τ be shear stress induced due to twisting moment, and σ be bending stress (tensile or comprehensive) induced due to bending moment

According to maximum shear stress theory, the maximum shear stress in the shaft is given by (Khurmi and Gupta, 2008):

$$\tau_{\max} = \frac{1}{2}\sqrt{\left(\sigma^2 + 4\tau^2\right)} \tag{2.19}$$

$$T_e = \frac{\pi}{16} \times \tau_{\max} \times d^3 = \sqrt{M^2 + T^2} \tag{2.20}$$

Where

T = Twisting moment (torque) acting upon the shaft

M = bending moment

The expression $\sqrt{M^2 + T^2}$ is known as the equivalent twisting moment and is denoted by T_e. From this expression, diameter of shaft (d) may be evaluated (Khurmi and Gupta, 2008).

From,

$$M_e = \frac{\pi}{32} \times \sigma_{b\max} \times d^3 = \frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right] \tag{2.21}$$

The expression $\frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right]$ is known as the equivalent bending moment and is denoted by M_e. From this expression, diameter of shaft (d_s) may be evaluated (Khurmi and Gupta, 2008).

2.4.2.2 Belt and Pulley design

Belts are used to transmit power between two rotating shafts in parallel. The belt is attached on pulleys mounted on the shafts. Pulleys are used which rotate either on the same or at different speeds. The pulley which supplies the power is known as the driving pulley while that which accepts the power is known as the driven pulley.

The amount of power transmitted is dependent on the following factors:

i. The belt velocity

ii. The tension under which the belt is placed on the pulleys

iii. The angle of contact between the belt and the smaller of the two pulleys

iv. The conditions under which the belt is used.

Belt drive selection

Factors to be considered include:

i. Speed of the driving and driven shafts

ii. Speed reduction ratio

iii. Power to be transmitted

iv. Centre distance between the shafts

v. Shafts layout

vi. Space available

vii. Service conditions

Types of belt drives

Belt drives are usually classified into the following three groups based on amount of power transmitted (Khurmi and Gupta, 2008).

i. Light drives- For light power transmissions at belt speeds up to about 10m/s.

ii. Medium drives- For medium power transmissions at belt speeds over10m/s.

iii. Heavy drives- For heavy power transmissions at belt speeds above 22m/s.

Types of belts

There are many types of belts, the most common belts are:

i. Flat belt

ii. Vee - belt

iii. Circular belt or rope

Velocity Ratio of Belt Drives

Speed ratio is given by (Khurmi and Gupta, 2008):

$$N_1 / N_2 = d_1 / d_2 \tag{2.22}$$

Where,

N_i = Speed of the driver

d_i = Pitch circle diameter of the driver

i = 1 for driver and 2 for driven pulley.

Length of Belt Drives

The length of open belt drives is given by (Khurmi, 2008):

$$l = ((0.5\pi \times (D_1 + D_2) + (2 \times CD) + \frac{(D_1 - D_2)^2}{4 \times CD} \tag{2.23}$$

For a cross belt drive, the length is given by (Khurmi and Gupta, 2008):

$$l = ((0.5\pi \times (D_1 + D_2) + (2 \times CD) + \frac{(D_1 + D_2)^2}{4 \times CD} \tag{2.24}$$

Where,

D_i = Diameter of the pulley

CD = Centre distance between pulleys

i = 1 for driver and 2 for driven pulley.

2.4.2.3 Bearings

Bearings are needed whenever one part of a machine slides against another. Bearings can be classified as providing sliding or rolling contact.

A sliding bearing typically uses a lubricant to reduce friction between the sliding surfaces. A shaft and bushing bearing is known as a journal bearing.

Rolling contact bearings are often referred to as antifriction bearings. Antifriction bearings, as their name implies, minimize friction by removing any possible sliding between bearing surfaces and replacing all contacts with rolling interfaces.

Fig 2.9 gives a pictorial view of bearings used.

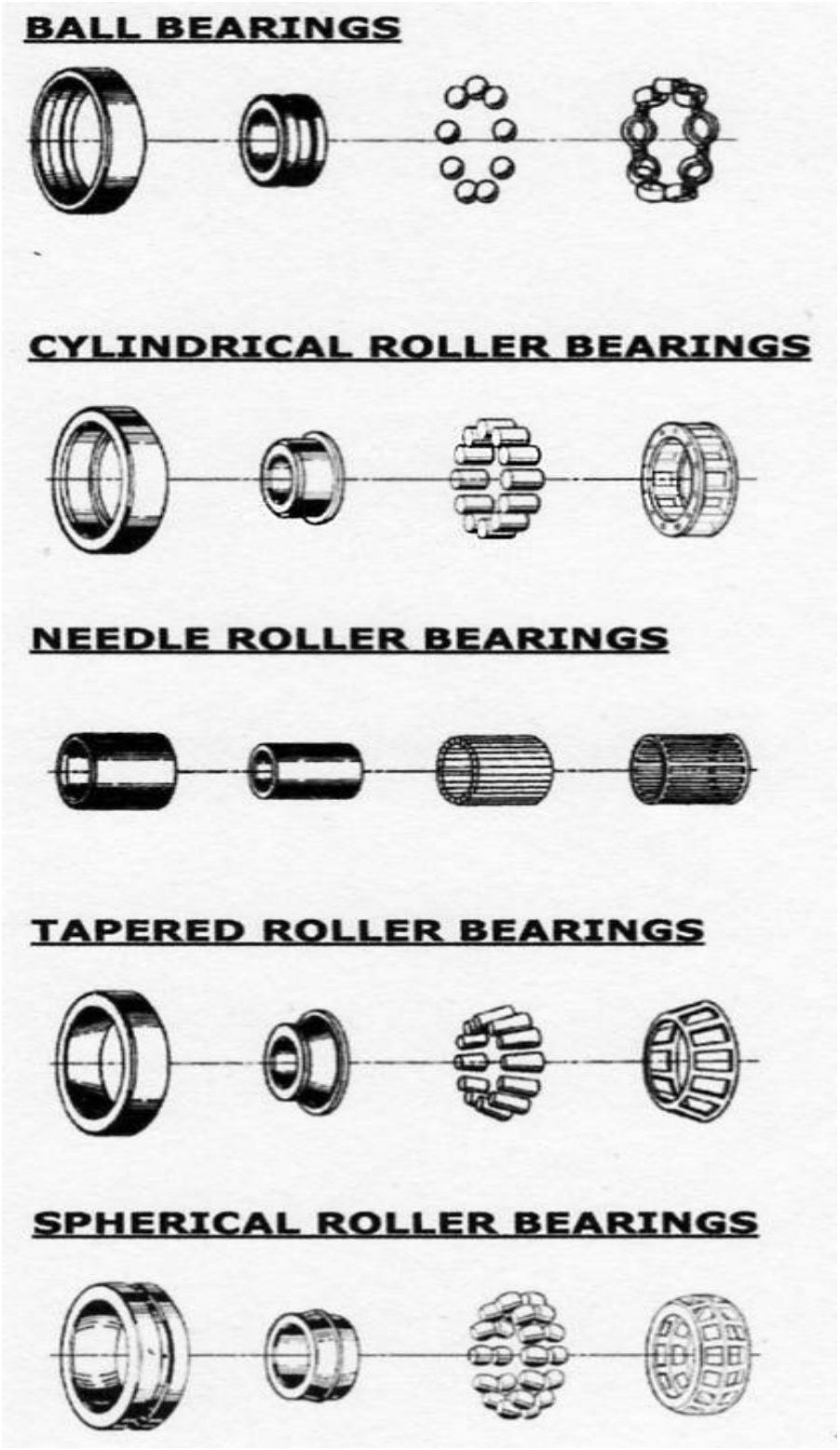


Fig 2.9 Bearing types

Source: (http://www.thebearingshop.com)

Factors to consider when selecting a bearing

i. Configuration and purpose of bearing application
ii. Application conditions (temperature, humidity, vibration, dust, etc.)
iii. Loads
iv. Shaft and housing dimensions and materials
v. Rotation: RPM, Rotation precision, Rotation Ring
vi. Bearing Torque
vii. Noise
viii. Life
ix. Regulated substance

Radial Ball Bearing selection steps

i. Calculate the dynamic radial load based on bearing application
ii. Obtain the design dynamic radial load by multiplying the dynamic radial load by a service factor.
iii. Select appropriate bearings based on information from manufacturers catalog with the design dynamic radial load as criteria.

Table 2.7 shows typical values of service factor K_s.

Table 2.7: Typical values of service factor for radial ball bearings (K_s)

S/N	Type of service	K_s
1	Uniform and steady load	1.0
2	Light shock load	1.5
3	Moderate shock load	2.0
4	Heavy shock load	2.5
5	Extreme shock load	3.0

Source: (Khurmi and Gupta, 2008).

The dynamic rating load of a bearing is given by (Khurmi and Gupta, 2008):

$$C = W\left(\frac{L}{10^6}\right)^{1/k} \tag{2.25}$$

Where

L = Rating Life

C=Basic dynamic load rating

W= Equivalent dynamic load, and

K= 3 for ball bearings and 10/3 for roller bearings

The relationship between life in revolutions (L) and the life in working hours (LH) is given by (Khurmi and Gupta, 2008):

$$L = 60NL_H \tag{2.26}$$

Where

N= revolution in rpm

2.4.3 Compressors

2.4.3.1 Air compressor

Basic Operation

The air compressor takes in air at a certain pressure and discharges it at a higher pressure, determined by the compressor's rated pressure ratio.

An air compressor operates by converting mechanical energy into pneumatic energy via the compression process. The input energy could come from a number of sources such as motor drive, petrol/diesel engine, or renewable energy sources such as the wind or solar power.

The ordinary hand bellows used by early smelters and blacksmiths was a simple type of air compressor. It admitted air through large holes as it expanded. As the bellows were compressed, it expelled air through a small nozzle, thus increasing the pressure inside the bellows and the velocity of the expelled air. Modern compressors use pistons, vanes, and other pumping mechanisms to draw air from the atmosphere, compress it, and discharge it into a receiver or pressure system.

Classification

Compressors generally fall under two categories:

- Displacement compressors which take in a certain amount of air at a time and compress it by reducing its volume and then discharges the smaller volume at a higher

pressure. Examples include reciprocating, diaphragm, screw, rotating vane, multi lobe compressors, etc. These compressors are associated with high pressure ratios and low compression volumes.

- Dynamic compressors or non- positive displacement compressors such as centrifugal and axial flow compressors. Dynamic compressors are associated with low pressure ratios and high compression volumes.

The flow and pressure requirements of a given application determine the suitability of a particular type of compressor. Figure 2.10 gives a classification of compressors.

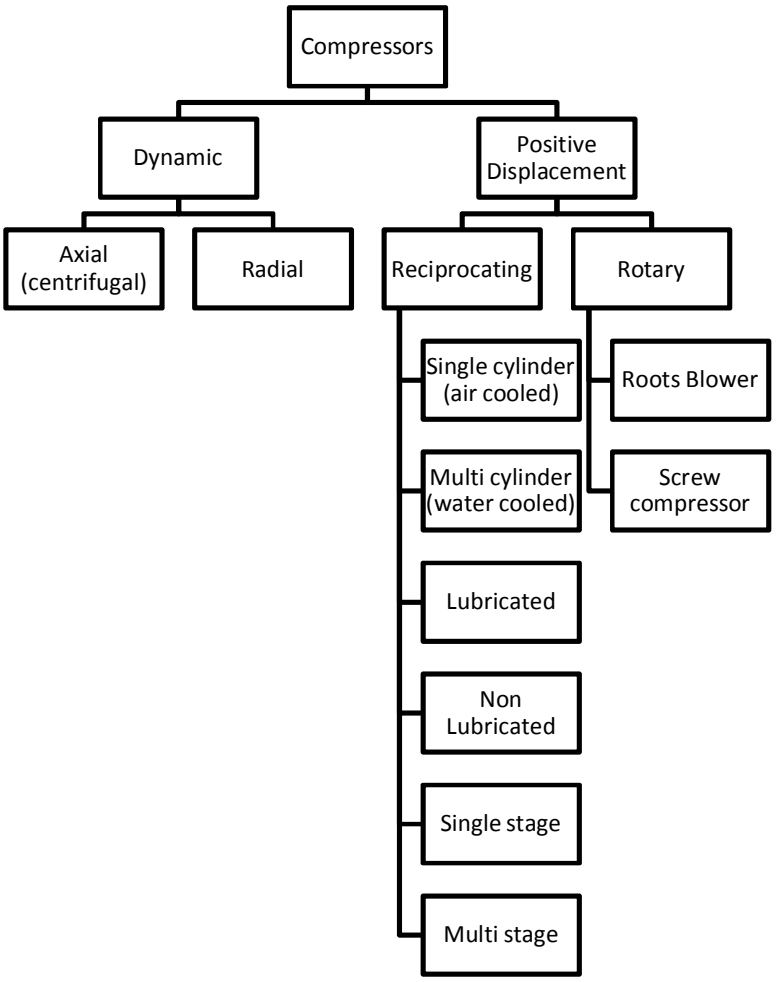


Figure 2.10 Compressor chart

Source: (Compressor selection, 2012)

2.4.3.2 Positive Displacement Compressors

Positive displacement compressors generally provide the most economical solution for systems requiring relatively high pressures. Their main disadvantage is that the displacing mechanism provides lower mass flow rates than non-positive displacement compressors.

A compressor with a positive displacement pumping mechanism has these important pressure characteristics:

- The pressure against which the compressor works rises to higher and higher values as pumping continues. It must be limited by some external pressure control device.
- The rate of free air delivery is highest at 0 barg and very gradually drops to lower values as pressure increases.
- The amount of heat generated progressively rises as pressure increases, causing substantial increases in temperature of both the air handled and the compressor structure.

Types of Positive Displacement Compressors

Positive displacement compressors are divided into those which compress air with a reciprocating motion and those which compress air with a rotary motion.

Reciprocating compressors

Reciprocating compressors are the most widely used type for air compression. They are characterized by a flow output that remains nearly constant over a range of discharge pressures. Also, the compressor capacity is directly proportional to the speed. The output, however, is a pulsating one.

Reciprocating compressors are available in many configurations, the four most widely used of which are horizontal, vertical, horizontal balance- opposed and tandem. Vertical type

reciprocating compressors are used in the capacity range of 0.0236 -0.0708 m^3/s. Horizontal balance opposed compressors are used in the capacity range of 0.0945-2.3597 m^3/s in multi-stage design and upto4.7195 m^3/s in single stage designs.

Reciprocating compressors are also available in variety of types:

i. Lubricated and non-lubricated
ii. Single or multiple cylinder
iii. Water or air-cooled.
iv. Single or multi stage

In the case of lubricated machines, oil has to be separated from the discharge air. Non-lubricated compressors are especially useful for providing air for instrumentation and for processes which require oil free discharge. However non-lubricated machines have higher specific power consumption ($kW/m^3/s$) as compared to lubricated types. Single cylinder machines are generally air-cooled, while multi-cylinder machines are generally water cooled, although multi-stage air-cooled types are available for machines up to 100 kW. Water-cooled systems are more energy efficient than air-cooled systems. Two stage machines are used for high pressures and are characterized by lower discharge temperature (140 to 160 °C) compared to single-stage machines (205 to 240 °C). In some cases, multi-stage machines may have a lower specific power consumption compared to single stage machines operating over the same total pressure differential. Multi-stage machines generally have higher investment costs, particularly for applications with high discharge pressure (above 7 barg) and low capacities (less than 0.0118 m^3/s). Multi staging has other benefits, such as reduced pressure differential across cylinders, which reduces the load and stress on compressor components such as valves and piston rings.

Fig. 2.11 is an illustration of a reciprocating compressor action during intake and discharge.

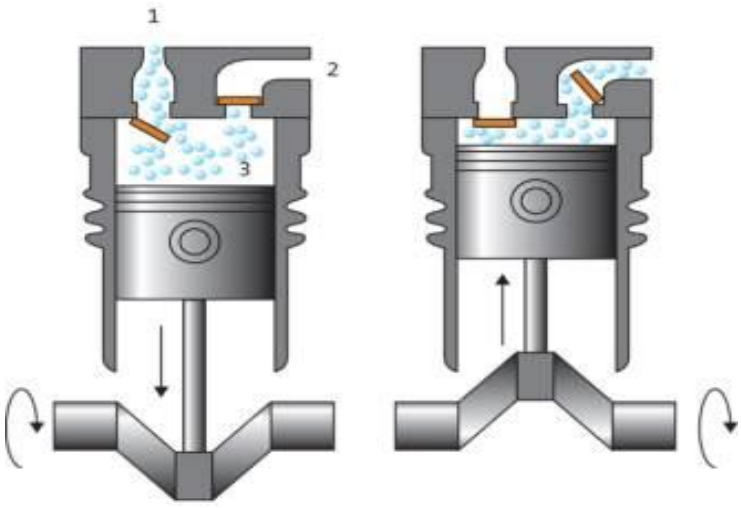


Fig 2.11 Reciprocating compressor intake and discharge stages

Source: (Compressor selection, 2012)

Rotary Compressors

Rotary compressors have rotors in place of pistons and give a continuous, pulsation free discharge air.

They are directly coupled to the prime mover and require lower starting torque as compared to reciprocating machine. They operate at high speed and generally provide higher throughput than reciprocating compressors. Also they require smaller foundations, vibrate less, and have a lower number of parts - which means less failure rate.

Among rotary compressor, the Roots blower (also called as lobe compressor) and screw compressors are among the most widely used. The roots blower is essentially a low-pressure blower and is limited to a discharge pressure of 1 bar in single-stage design and up to 2.2 bar in two stage design. The most common rotary air compressor is the single stage helical or spiral lube oil flooded screw air compressor. These compressors consist of two rotors, within a casing where the rotors compress the air internally. There are no valves. These units are basically oil cooled (with air cooled or water cooled oil coolers) where the oil seals the internal clearances. Since the cooling takes place right inside the compressor, the working

parts never experience extreme operating temperatures. The oil has to be separated from discharge air. Because of the simple design and few wearing parts, rotary screw air compressors are easy to maintain, to operate and install. The oil free rotary screw air compressor uses specially designed air ends to compress air without oil in the compression chamber producing true oil free air. These compressors are available as air-cooled or water cooled types and provide the same flexibility as oil flooded rotary compressors. There is a wide range of availability in configuration and in pressure and capacity. Dry types deliver oil-free air and are available in sizes up to 9.4389 m^3/s and pressure up to 15 bar. Lubricated types are available in sizes ranging from 0.04719 - 0.4719 m^3/s, with discharge pressure up to 10 bar.

Fig. 2.12 is an illustration of a reciprocating compressor.

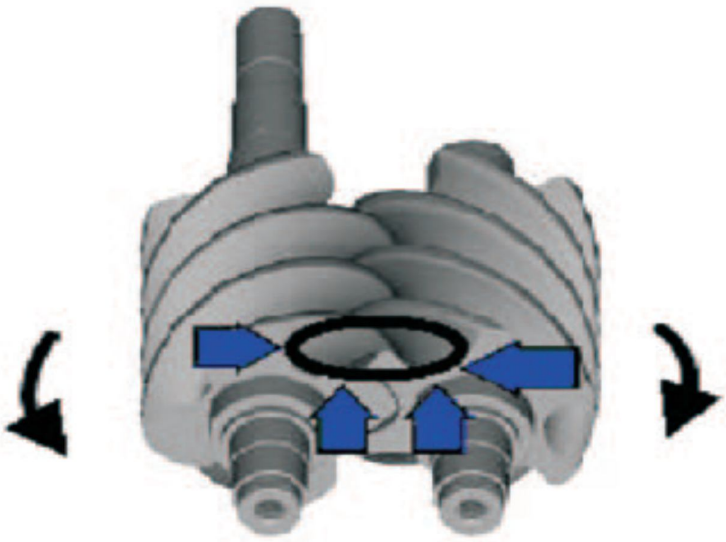


Fig 2.12 screw compressor internals

Source: (Compressor selection, 2012)

2.4.3.3 Non-positive Displacement Compressors

These compressors are also called "dynamic," "continuous-flow", and "velocity-type" compressors, this category comprises machines that use changes in kinetic energy to create pressure gradients.

Unlike the positive displacement compressor, in which distinct volumes of air are isolated and compressed, a non-positive displacement compressor does not provide a constant-volume flow rate over a range of discharge pressures. This is because the compartments are not isolated from each other and leakage between them increases as pressure rises. Initial acceleration of the air produces a negative (suction) pressure at the inlet port, drawing air in. Partial deceleration of air at the discharge port converts some of the kinetic energy to pressure. Speed of the rotating impeller determines the pressure change. Higher pressure differences require either faster impeller speeds or additional stages. The most important advantage of nonpositive displacement machines is their ability to provide very high mass flow rates. On the other hand, multiple stages are required to provide pressures above 0.28-0.34 bar and such machines are cost effective only for flow rates above 0.03776 - 0.04719 m^3/s.

Non-positive displacement devices are sometimes called fans or blowers rather than compressors. By some definitions, a fan provides less than 0.034bar pressure and a blower between 0.034-0.7 bar. The distinction is frequently blurred in common use, however.

The three common types of nonpositive displacement compressors are centrifugal, axial, and peripheral (or regenerative). These names derive from the direction of air flow through their compression chambers.

Centrifugal compressor

Centrifugal compressors are best suited to the continuous movement of large air volumes through small pressure ranges. Fig. 2.13 shows the basic operation of a centrifugal

compressor. Air leaving a rotating impeller passes radially outward to the casing. Centrifugal action builds up velocity and pressure levels.

In its simplest form, a centrifugal compressor consists of a high-speed rotating impeller that receives air through an inlet nozzle at the center. The impeller vanes are fixed (unlike those in the rotary vane design). They throw the air centrifugally outward toward the casing, increasing its velocity and energy. Here, an outlet discharges the air into a stationary passageway known as a "diffuser." The diffuser reduces the air velocity, thus raising the pressure. Beyond the diffuser, the velocity may be further reduced and pressure increased by a "collector." Staging can yield higher pressures. Staging is accomplished by directing the output from the diffuser of one stage into the nozzle of the next. Because the flow from the impeller is continuous, a smooth, surge-free output is obtained. Furthermore, discharge pressure depends only on impeller speed. It is nearly constant, despite variations in flow, over the stable operating range. But this can be a drawback if the demand falls far enough below the rated flow, allowing system pressure to build up. The compressor continues to deliver air at about the same pressure until the back-pressure exceeds that developed by the compressor. The result is "surge"-a reversal of flow. This reversal immediately allows the back-pressure to go down, and regular compression is resumed.

Surge can be prevented if flow remains above a limit established for each design. Various models have minimum operating flows between 45 and 90 percent of rated capacity. Centrifugal compressors are available in both small and very large sizes. Units with up to six stages and supplying 14.1584 m^3/s of air are commercially available. Operating speeds are very high compared with other types-up to 20,000 rpm in standard applications.

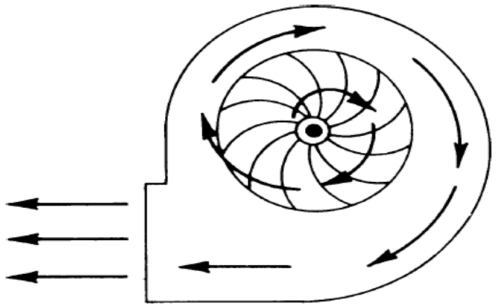


Fig 2.13 Centrifugal blower

Source: (Compressor selection, 2012)

Axial Flow

This category is generally used for ultrahigh flow applications (14.1584 to 471.9475 m^3/s). Air flow is through a duct, primarily in a direction parallel to the axis of rotation. In multistage versions, this flow channeling is provided by the fixed guide vanes or stator blades positioned between each stage (Fig. 2.14). An axial flow compressor requires about a third the floor space of a centrifugal design, and it weighs about a third as much. Below capacities of 47.19475 m^3/s, though, the axial design is seldom competitive in price.

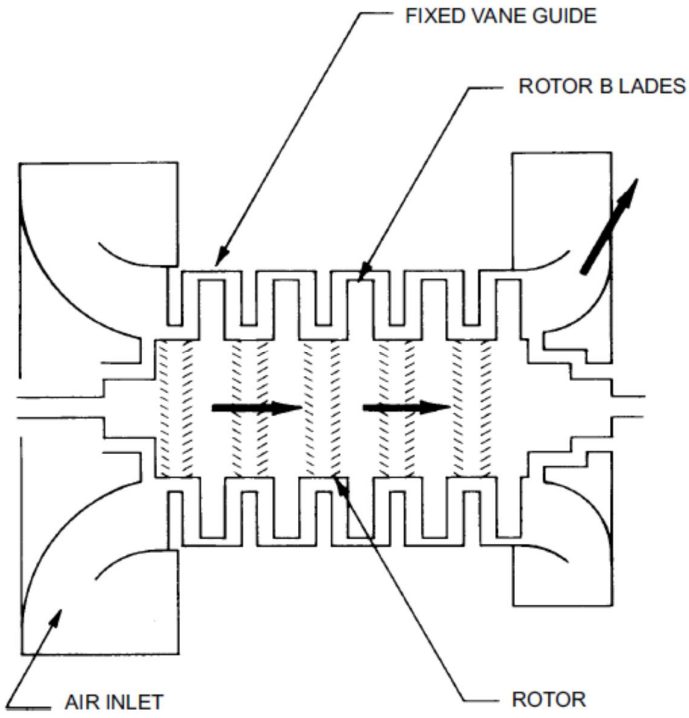


Fig. 2.14 Axial compressor

Source: (Compressor selection, 2012)

Peripheral (Regenerative)

These units (Fig. 2.15) provide somewhat higher pressures than do the other dynamic designs. With some units, a single-stage regenerative action provides pressures similar to those obtained with multistage centrifugal compressors. The compression space consists of a hollow, circular ring between the tips of the impeller blades and the walls of the peripheral passage. In operation, the rotating impeller draws air from the inlet port into the compression space. The air moves radially outward to the curved housing by centrifugal force. The action is called "regenerative" because a certain amount of air slips past each impeller blade during rotation and returns to the base of a succeeding blade for reacceleration. The effect is like the pressure buildup in a multistage blower, and higher pressures can be generated.

Single-stage peripheral blowers are available in capacities up to several hundred m^3/s and can generate pressures close to 0.34 barg. Multistage versions are also available. A significant

advantage for peripheral blowers is that they are highly immune to operating conditions that might otherwise cause blockage of inlet and discharge flows. They also provide oil-less operation and continuous pulse-free flow.

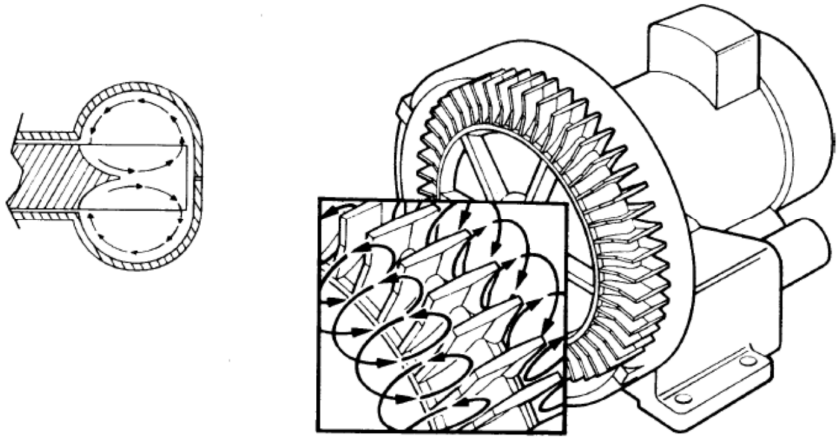


Fig. 2.15 Regenerative blower

Source: (Compressor selection, 2012)

2.4.3.4 Selection of Compressor Type

Factors to consider for a selection of an air compressor include:

i. Pressure required to meet specific application needs.

ii. Air flow required to meet specific application needs.

iii. The drive power requirements for the compressor and how they will be met.

iv. Cost

v. Space

vi. Weight limitations

vii. Possible needs for oil- free or pulseless air.

viii. Availability.

Maximum Pressure Rating

The primary criterion for evaluating performance of a compressor is its maximum pressure rating. This is defined as the maximum pressure at which the compressor can deliver air to the system in commercial operation. For any compressor, the physical design sets limits (because of such factors as air leakage and drive power limitations) on the pressure that can be generated. But in many cases, it is heat build-up that determines the actual pressure rating. The easier the compressor is to cool, the higher the pressure rating. This is also why many compressors have continuous-duty ratings that are considerably lower than their intermittent-duty ratings. Basically, the selection of the appropriate compressor type is determined by comparing the maximum pressure requirements of the application with the maximum pressure rating of available compressor types.

As Table 2.8 shows, an application where the system pressure requirement is relatively low will give the designer a greater variety of compressor types from which to choose. As the system pressure requirement increases, the number of available alternatives diminishes sometimes to a point where only a single compressor is applicable. Sometimes it may be a rather costly unit and have relatively high power requirements per cubic metre of air delivered.

Table 2.8 Pressure ratings and Applicable compressors

Required System Pressure (barg)		Compressor types available	
Continuous	Intermittent	Preferred type	Optional type
6.90 – 12.07	12.07 – 13.79	Piston (Two stage) Piston (one stage)	-
3.45 – 6.90	-	Rocking Piston (One stage)	-
2.07 – 4.14	-	Diaphragm	Rocking Piston (One stage)
1.72 – 2.07	1.72 – 2.07	Rotary vane(oil lubricated)	Any of the above
0.69 – 1.03	1.03 – 1.38	Rotary vane(oil less)	Any of the above
0.69	0.69	Rotary vane(oil lubricated and oil less)	Any of the above
0.24		Regenerative(peripheral) blower	-

Pressure ratings based on 10 minutes on 10 minutes off duty cycle

Source: (Compressor selection, 2012)

The working pressure requirements of individual devices can be determined from handbook formulae, performance curves, catalog data, or actual tests made with prototype systems. Normally, the working pressure needed will depend on the size of the actuator (work device) used. Sometimes system pressure requirements can be reduced when it is practical to use a larger actuator. That is, the required force can be generated by a lower pressure if it acts over a larger area. Overall system costs usually determine such decisions. In some cases, the required working pressure may depend on the "pressure heads" needed for specific air flow velocities or rates.

Air Flow

While flow capacity is the primary factor in selecting a given size compressor, it also enters when selecting a compressor type. This is because different types of compressors tend to offer trade-offs between air flow and pressure capability. Not only do dynamic compressors provide higher flows and lower pressures than do positive displacement units, but the same is often true within each group. Rotary vane compressors, for example, tend to provide more air but less pressure than do comparably sized piston compressors. Thus, after determining the required air pressure, the usual procedure is to select the type of compressor that will provide maximum air flow at that pressure, subject to other constraints. At a minimum, it is necessary to be sure those units offering the required flow are available in the type of compressor selected. Table 2.9 lists typical capacities for various types of compressors.

Table 2.9 Capacity ranges, Pressure ratings and Applicable compressors

Available compressor types	Maximum Pressure (barg)		Range of capabilities(m^3/s at 0 barg)	
	Continuous	intermittent	Smallest	Largest
Piston(1 stage)	3.45 – 6.90	3.45 – 6.90	0.000614	0.005191
Rocking Piston(1 stage)	0.69 – 6.90	0.69 – 6.90	0.000283	0.001534
Diaphragm (1 stage)	3.45 – 6.90	3.45 – 6.90	0.000241	0.001793
Rotary vane (oil lubricated)	0.69 – 1.72	0.83 – 2.07	0.000614	0.025957
Rotary vane (oil less)	0.69 – 1.03	0.69 – 1.38	0.000165	0.025957

Compressors depicted are those manufactured by GAST.

Source: (Compressor selection, 2012)

Power requirements

Power requirements listed in the compressor manufacturer's catalog should assure adequate power for rated speed operation at any pressure up to and including the compressor's maximum pressure rating.

The drive speed determines the amount of air delivered. If the compressor is operated at less than the rated speed given in the manufacturer's catalog, both air flow and the horsepower required will be reduced in the same proportion. The torque requirement will remain unchanged, but only with positive displacement compressors.

Other Factors

Contamination- Oil vapors may cause contamination or deterioration of products and materials in some applications. Oil-less compressors are generally required in such cases. They also reduce maintenance costs by eliminating periodic filling of lubricators. Available oil-less compressors include most of the piston and diaphragm types, regenerative blowers, and some of the rotary vane designs.

Pulse-free Delivery- Applications requiring continuous, pulse-free air delivery-without the extra cost and space requirements of a receiver tank-usually dictate that the compressor use a rotary pumping mechanism, such as the rotary vane design. An additional advantage of this compressor type is that noise and vibrations are considerably below those of reciprocating designs. The regenerative blower type also has pulse-free delivery and low vibration, but the high impeller speeds can generate high-pitched noise.

Mounting Space- Very often, compressor selection is further limited by available space at the installation site. For lower pressure systems, the compact rotary vane compressors usually require less space for a given free air capacity than do piston or diaphragm designs.

Table 2.10 and 2.11 below gives general compressor selection criteria based on a number of characteristics:

Table 2.10 Compressor comparison based on characteristics

Item	Reciprocating	Rotary Vane	Rotary Screw	Centrifugal
Efficiency at full load	High	Medium - high	High	High
Efficiency at part load	High due to to staging	Poor: below 60% of full load	Poor: below 60% of full load	Poor: below 60% of full load
Efficiency at no load (power as % of full load)	High (10% - 25%)	Medium (30% - 40%)	High-Poor (25% - 60%)	High-Medium (20% - 30%)
Noise level	Noisy	Quiet	Quiet-if enclosed	Quiet
Size	Large	Compact	Compact	Compact
Oil carry over	Moderate	Low-medium	Low	Low
Vibration	High	Almo st none	Almost none	Almost none
Maintenance	Many wearing parts	Few wearing parts	Very few wearing parts	Sensitive to dust in air
Capacity	Low - high	Low - medium	Low - high	Medium - high
Pressure	Medium - very high	Low - medium	Medium - high	Medium - high

Source: (Sustainable Energy Development Office, 2002)

Table 2.11 General Selection criteria for compressors

TABLE 3.1 GENERAL SELECTION CRITERIA FOR COMPRESSORS				
Type of Compressor	Capacity (m^3/h)		Pressure (bar)	
	From	To	From	To
Roots blower compressor single stage	100	30000	0.1	1
Reciprocating				
– Single / Two stage	100	12000	0.8	12
– Multi stage	100	12000	12.0	700
Screw				
– Single stage	100	2400	0.8	13
– Two stage	100	2200	0.8	24
Centrifugal	600	300000	0.1	450

Source: (Bureau of energy efficiency, 2002)

2.5 Pressure Vessels

These are vessels (Fig.2.16) capable of holding air at above atmospheric pressure over a given time period. The storage medium is usually spherical or cylindrical in shape. For practical purposes, most storage mediums are elliptical. Each pressurised vessel has a rated pressure, which is the maximum pressure it can hold. The pressurised vessel usually has a calibrated gauge to indicate the pressure of the fluid it is holding.



Fig 2.16 Pressure Vessel

Source: (http://www.charamtech.com/pressure-vessel.htm)

CHAPTER THREE

MATERIALS AND METHODS

3.1 The Wind Compressor System

The wind air compressor system consists of the following components:

- Wind rotor
- Transmission system
- Air compressor
- Storage reservoir

Figure 3.1 shows a schematic diagram of the wind air compressor system

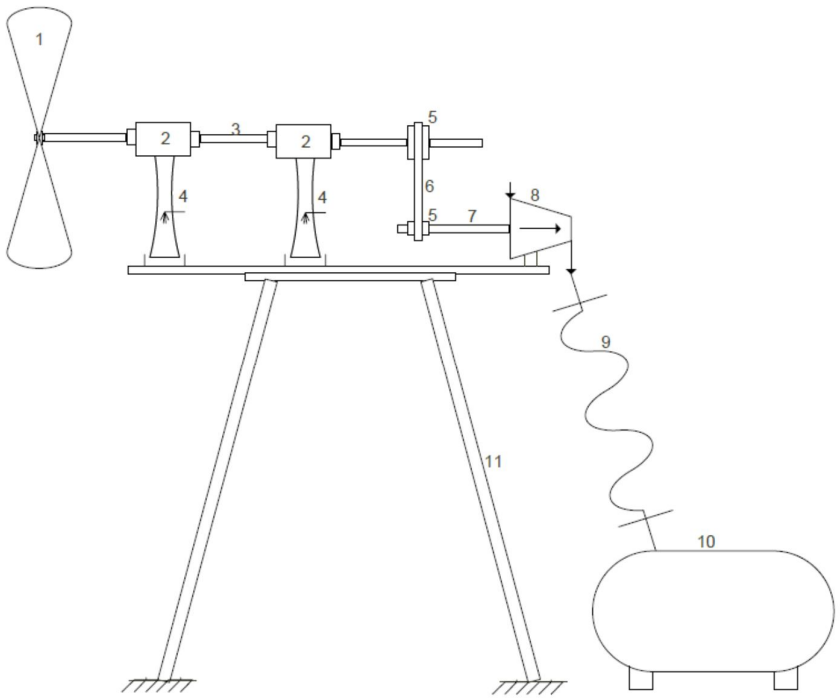


Figure 3.1: Schematic diagram of the wind air compressor system

KEY for Figure 3.1		
1. Wind Rotor	5. Pulley	9. Flexible Hose
2. Bearings	6. Belt	10. Air Vessel
3. Rotor shaft	7. Compressor Shaft	11. Tower
4. Bearing Support	8. Air Compressor	

From figure 3.1, it is seen that the wind rotor consists of blades, which are connected to a hub that is mounted on a driving shaft. The driving shaft on which the rotor is mounted is supported by two bearings to provide adequate support. A driving pulley is placed on the driving shaft. A pulley is located at the edge of the compressor shaft. The two pulleys are connected by means of a vee belt to have adequate grip in operation. The driving shaft, the belt and pulley system and the air compressor shaft make up the transmission mechanism.

The driving shaft, the belt and pulley system and the air compressor are all supported on a horizontal platform which sits atop a tower.

The wind compressor was designed with a tail vane to ensure it always faces the wind (Piggot, 2003).

A flexible hose was used to connect the compressor discharge to a storage vessel at "ground" level.

3.2 Materials

3.2.1 Material Selection

The materials used for each component were selected based on availability and guided by information from the Wind turbine- Materials and Manufacturing Fact Sheet prepared by the Princeton Energy Resources International (PERI) based on current trends in wind turbine technology (Ancona and McVeigh, 2002).

Table 3.1 shows the materials chosen for each component of the system.

Table 3.1: Materials used for the wind compressor

S/N	Component	Material	Reason for selection
1	Blades	Polyvinyl chloride(PVC)	• Light weight • Corrosion free • Relatively cheap
2	Shaft	Steel Alloy A36 (AISI 1045)	• High yield strength • Good machinability
3	Hub	Steel Alloy (AISI 1060)	• High yield strength • High hardness
4	Pulleys	Grey Cast iron	• High hardness • High compressive strength
5	Belt	Rubber	• Low poisson ratio with cast iron
6	Tail	Aluminium alloy 1100	• Relatively low weight
7	Tower	Grey Cast iron	• High yield strength
8	Foundation	Concrete, Steel	• High compressive

			strength

3.2.2 Wind Speed Data

Long term average monthly and annual meteorological wind speed (over 5 years) and distribution data for Zaria was obtained from Pam (2008). Pam (2008) obtained raw wind speed from the Nigerian Meteorological Agency, Oshodi, Lagos. The local wind regime of Zaria was of particular interest since all design and specifications of the system will be peculiar to it. Appendix A gives the local wind regime of Zaria (Pam, 2008). From the data it can be seen that the average annual mean speed is 3.40 m/s at a height of 30m which is used for the design.

3.3 Design Theories

3.3.1.0 Rotor Design

3.3.1.1 Rotor swept Area

The rotor swept area can be determined from equation (2.4):

$$A = \frac{2P_{pn}}{\rho_a \eta_{overall} V^3} \tag{3.1}$$

But,

$$A = \frac{\pi d_r^{\,2}}{4}, \tag{3.2}$$

Where,

d_r = rotor diameter (m)

Hence,

$$d_r = \sqrt{\frac{8P_{pn}}{\pi \rho_a \eta_{overall} V^3}} \tag{3.3}$$

3.3.1.2 Compressor Power

Substituting, $\gamma = 1.40$, $R = 287 J/kg.K$ and $\eta_{comp} = 0.36$ (Gresh,2001) into equation (2.8) ,the pneumatic power becomes:

$$P_{pn} = 2790 \dot{m} T_1 \left(r^{0.29} - 1\right) \tag{3.4}$$

Equating equation (2.3) and equation (2.8) the overall efficiency becomes:

$$\eta_{overall} = \frac{2\dfrac{\dot{m} R T_1}{\eta_{comp}} \left(\dfrac{\gamma}{\gamma - 1}\right)\left(r^{\left(\frac{\gamma-1}{\gamma}\right)} - 1\right)}{\rho_a A V^3} \tag{3.5}$$

Substituting $A = \pi d^2 / 4$, $R = 287 J/kg.K$, $\gamma = 1.40$, $\eta_{comp} = 0.36$ (Gresh,2001) and $\dot{v} = \dfrac{\dot{m}}{\rho_a}$

where d_r is rotor diameter (m), the overall efficiency becomes:

$$\eta_{overall} = 7105 \frac{\dot{v} T_1 \left(r^{0.29} - 1\right)}{d_r{}^2 V^3} \tag{3.6}$$

The average of equation (2.3) gives the mean pneumatic power from a wind compressor over a time period, at a particular location. Thus

$$\overline{P_{pn}} = \frac{1}{2} \overline{\eta_{overall}} \rho_a A \overline{V^3} \tag{3.7}$$

Substituting $\overline{V^3} = k_e \overline{V}^3$ and $A = \frac{\pi d_r^{\ 2}}{4}$, the pneumatic power becomes

$$\overline{P_{pn}} = \frac{\pi}{8} \overline{\eta_{overall}} k_e \rho_a d_r^{\ 2} \overline{V}^3 = 0.4504 \overline{\eta_{overall}} k_e d_r^{\ 2} \overline{V}^3 \tag{3.8}$$

Hence,

$$\overline{P_{pn}} = 0.4504 \, \overline{\eta_{overall}} \, k_e d_r^{\ 2} \overline{V}^3 \tag{3.9}$$

Where

k_e = energy pattern factor of a location

$\overline{V^3}$ = mean cube of wind speeds within the period

$\overline{V}$ = mean of wind speeds within the same period, and

$$\overline{V} = \frac{1}{n} \sum_{i=1}^{n} V_i \tag{3.10}$$

Equating equations (3.4) and (3.9), the mean mass flow rate discharged from a wind compressor defined by its rotor diameter for a location characterised by the mean wind speed and energy pattern factor can be shown to be:

$$\dot{m} = 1.6143 \times 10^{-4} \frac{\overline{\eta_{overall}} k_e d_r^{\ 2} \overline{V}^3}{T_1 \left(r^{0.29} - 1 \right)} \tag{3.11}$$

3.4 Design Analysis

3.4.1 Compressor Selection Considerations

i. Compressor is to be operated intermittently.

ii. Compressor should operate optimally between 2 to 6 bar(g) (typical operating pressure)

iii. Compressor maximum pressure must be greater than 2.5 barg.

iv. Compressor capacity should be a minimum of 0.000000167 m^3/s at maximum operating pressure.

Selected Compressor features

Compressor: Volvo air compressor, DRS-15BH Part No 509630-8562

Description: A rotary air compressor of maximum discharge pressure of 3.53 barg, a free air delivery (FAD) of 0.001179 m^3/s at maximum pressure , and a nominal power requirement of 130 W was procured. The selection was based on the output pressure required at the storage reservoir and the nominal power required by the compressor.

3.4.2 Shaft design

The design of the wind turbine shaft was based on the two moments to which it is subjected to- the twisting and bending moments- using the Maximum shear stress theory (Guest's theory) (Khurmi and Gupta, 2008).

Equations (2.19), (2.20) and (2.21) were used for the design.

From equation (2.20), diameter of shaft (d_s) may be evaluated as:

$$d_s = \sqrt[3]{\frac{16T_e}{\pi\tau_{\max}}} \tag{3.12}$$

From equation (2.21), diameter of shaft (d_s) may be evaluated as:

$$d_s = \sqrt[3]{\frac{32M_e}{\pi\sigma_{b\max}}} \tag{3.13}$$

3.4.3 Bearing selection

Rolling contact bearings were selected based on ease of use, suitability, carrying capacity, relative high service life and size of shaft.

The dynamic load rating for the bearings was calculated from the relation (Khurmi, 2008):

$$C = W\left(\frac{L_{90}}{10^6}\right)^{1/k} \tag{3.14}$$

3.4.4 Tail vane design

To ensure that furling takes place at the right wind speed, the area of the tail vane and the length of the tail boom are both dependent on the diameter of the windmill. Generally, the tail boom should be about one blade length (radius of the wind mill)(Piggot, 2003).

As for the area of the tail vane, it can be determined from the equation (Piggot, 2003):

$$A_t = \frac{d_r{}^2}{40} \tag{3.15}$$

The shape of the tail vane really has very little effect on the mill as long as the area is correct.

The tail vane is made of stainless steel and it is riveted onto the steel plate connecting it to the wind rotor.

3.4.5 Design Considerations

The following parameters were considered for the design:

i. The mean annual wind speed at a height of 30 m was 3.40 m/s

ii. The compressor rating of 130 W was used to reduce the size of the constructed system.

iii. Design was based on power required at compressor shaft and efficiencies of belt drive system and bearing systems.

iv. The ratio of rated speed to mean annual wind speed($\frac{v_r}{v_m}$) of 1.5 was used(Harder,1982)

v. An overall efficiency of 0.35 was used based on studies conducted (Frankael, 1986).

vi. A tip speed ratio of 3 was used based on studies conducted (Frankael, 1986). This falls within a suitable tip speed ratio range for multi bladed wind mills.

vii. Six tapered blades were used. This is from figure 2.5 and is suitable for high torque required for compressor start.

viii. The blade profile is a curved plate with 10^o curvature (Lysen, 1983)

ix. The angle of attack of 4^o for minimum lift coefficient of 1.25(Lysen, 1983)

x. Poly Vinyl Chloride (PVC) is used for blade manufacture due to its Low inertia due to low rotating mass, light weight, Corrosion-proof and Low maintenance required. It also has a relatively low manufacturing cost.

xi. A flat belt made of rubber on a cast iron pulley was selected with a coefficient of friction of 0.30(Khurmi and Gupta, 2008)

xii. A factor of safety of 6 was used for the design. (Khurmi and Gupta, 2008)

xiii. Minimum allowable yield stress of AISI 1045 steel is $320N/mm^2$(Budynas and Nisbett, 2011)

xiv. Minimum allowable yield stress of AISI 1060 steel is $370N/mm^2$(Budynas and Nisbett, 2011)

xv. Modulus of elasticity of steel and cast iron are 200GPa and 100GPa respectively. (Khurmi and Gupta, 2008)

xvi. Friction loss of 5% assumed for driving shaft due to self lubricating nature of bearings used.

xvii. An overload of 20% assumed based on studies conducted by (Frankael, 1986).

3.5 Design Calculations

Initial Data	**Calculations and Sketches**	**Results**
Air compressor $P_{pn} = 130W$ $P_1 = 1\times10^5 N/m^2$ $P_2 = 3.53\times10^5 N/m^2$	**<u>Rotary air compressor parameters calculation</u>** The pressure ratio, r is given by: $r = \frac{P_2}{P_1} = \frac{3.53\times10^5}{1.0\times10^5} = 3.53$	$r = 3.53$
$V_m = 3.40 m/s$	**<u>Rated wind speed of the machine(V_r)</u>** The rated wind speed is given as: $V_r = 1.5V_m = 1.5\times3.40 = 5.10 m/s$	$V_r = 5.10 m/s$

Wind rotor $P_{pn} = 130W$ $\rho_a = 1.147kg/m^3$ $\eta_{overall} = 0.35$ $v_r = 5.10\,m/s$	**Wind rotor design** **i. Rotor diameter(d_r)** The rotor diameter is given by equation (3.3) $d_r = \sqrt{\frac{8P_{pn}}{\pi\rho_a\eta_{overall}V^3}}$ The rotor diameter is calculated thus: $d_r = \sqrt{\frac{8\times130(W)}{\pi\times1.147(kg/m^3)\times0.35\times(5.10)^3(m/s)^3}}$ $d_r = 2.4933\,m \approx 2.50\,m$ $r_r = \frac{d_r}{2} = 1.2467m \approx 1.25m$ $B = 6$ (selected)	$d_r = 2.50m$ $r_r = 1.25m$ $B = 6$
$\alpha = 3^o$ $\lambda = 3$ $B = 6$ $d_r = 2.50m$ $r_r = 1.25m$	**ii. Blade Profile** The blade profile is a curved plate with 10^o curvature i. The blade cord at various sections along the span of the rotor blades is given by equation (2.17): $C = \frac{16\pi r_r(\frac{r}{r_r})}{9\lambda^2 B}$	

At various stations(1-5) as in fig 2.7:

At station 5, r =r_r =1.25m

$$C_{r=1.25} = \frac{16\pi \times 1.25 \times (\frac{1.25}{1.25})}{9 \times (3)^2 \times 6} = 0.2901m \approx 0.04m$$

$C_{r=1.25} = 0.04m$

At station 4, r=r_r= 1.00m

$$C_{r=1.0} = \frac{16\pi \times 1.25 \times (\frac{1.25}{1.00})}{9 \times (3)^2 \times 6} = 0.0514m \approx 0.05m$$

$C_{r=1.00} = 0.05m$

At station 3, r=r_r = 0.75m

$$C_{r=0.75} = \frac{16\pi \times 1.25 \times (\frac{1.25}{0.75})}{9 \times (3)^2 \times 6} = 0.0686m \approx 0.07m$$

$C_{r=0.75} = 0.07m$

At station 2, r =r_r= 0.50m

$$C_{r=0.50} = \frac{16\pi \times 1.25 \times (\frac{1.25}{0.50})}{9 \times (3)^2 \times 6} = 0.1029m \approx 0.1m$$

$C_{r=0.50} = 0.1m$

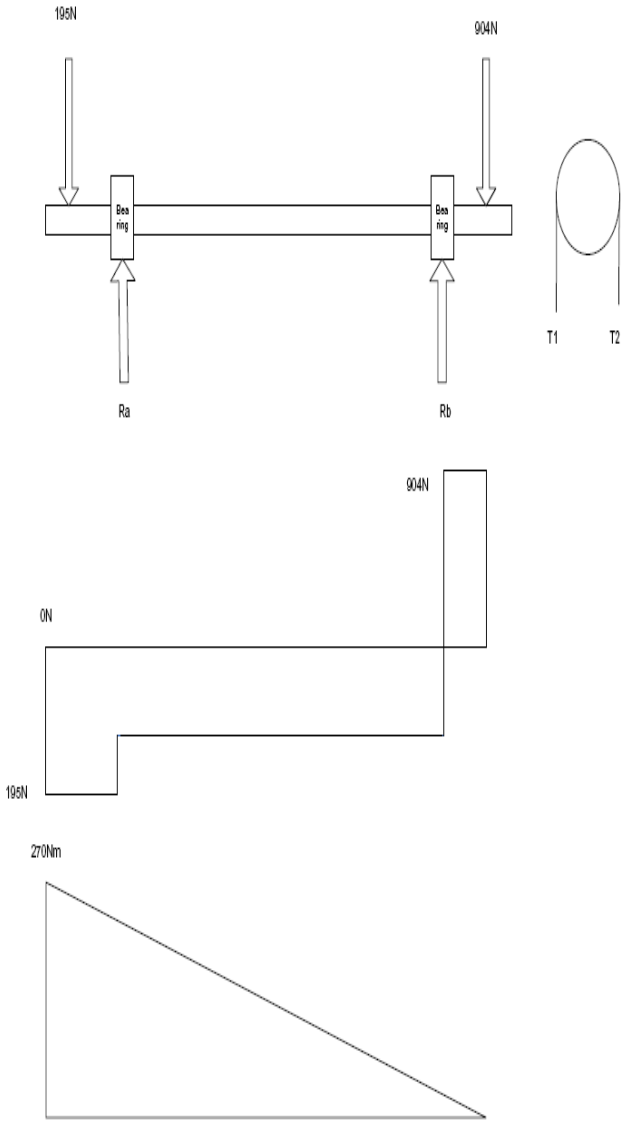
At station 1, r =r_r= 0.25m

$$C_{r=0.25} = \frac{16\pi \times 1.25 \times (\frac{1.25}{0.25})}{9 \times (3)^2 \times 6} = 0.2m$$

$C_{r=0.25} = 0.2m$

ii. The blade setting angle is given by that at station 3 using equation (2.15)(Piggot, 2003)

$$\beta = \tan^{-1}(2r_r / 3r\lambda) - \alpha$$

	Therefore, $\beta = \tan^{-1}(2\times1.25/3\times1.00\times3)-3=14^o$	$\beta = 14^o$
Rotor shaft Maximum tensile strength = 560 MPa Factor of safety = 6. $P_{pn}=130W$ $V_r = 5.10\,m/s$ $J = 60\times10^{6}\,m^4$ $G = 80\times10^{9}\,Pa$	**<u>Rotor shaft design</u>**  From the free body diagram,	

	R_a= 105.78 N R_b= 842.44 N Ultimate tensile strength(UTS) = 36% of Maximum tensile strength= 201.6 MPa Ultimate shear strength(USS) = 18% of Maximum tensile strength=100.8 MPa Allowable tensile stress(σ)= UTS/factor of safety = 201.6 MPa/6 =33.6 MPa Allowable shear stress(τ)= USS/factor of safety = 100.8 MPa/6 =16.8 MPa Maximum shear stress is obtained from equation (2.19) $\tau_{\max} = \frac{1}{2}\sqrt{(\sigma^2 + 4\tau^2)}$ $\tau_{\max} = \frac{1}{2}\sqrt{((33.6\times10^6)^2 + 4\times(16.8\times10^6)^2)}$ $\tau_{\max} = 23.76MPa$ Rotational speed of shaft is obtained from equation(2.18)	$UTS = 201.6MPa$ $USS = 100.8MPa$ $\sigma = 33.6MPa$ $\tau = 16.8MPa$ $\tau_{\max} = 23.76MPa$

$$\omega = \frac{2\lambda V_r}{d_r}$$

$$\omega = \frac{2 \times 3 \times 5.10}{2.50} = 12.24 rad/s$$

$\omega = 12.24 rad/s$

$$N = \frac{60\lambda V_r}{\pi d_r}$$

$$N = \frac{60 \times 3 \times 5.10}{\pi \times 2.50} = 116 rpm$$

Torque to be transmitted by shaft

$N = 116 rpm$

$$T = \frac{60 P_s}{2\pi N_s}$$

$$T = \frac{60 \times 130}{2 \times \pi \times 116} = 10.70 Nm$$

The bending moment across a section x of the shaft is given by:

$T = 10.70 Nm$

$$M_x = 195x - R_a(x - 0.06) - R_b(x - 0.275)$$

The maximum bending moment due to rotor and pulley weight =238Nm (occurs at x = 0 m)

Resultant bending moment, M = 238Nm

$M = 238Nm$

The equivalent twisting and bending moments are given by equations(2.20) and (2.21)

$$T_e = \frac{\pi}{16} \times \tau_{\max} \times d^3 = \sqrt{M^2 + T^2}$$

$$M_e = \frac{\pi}{32} \times \sigma_{b\max} \times d^3 = \frac{1}{2}\left[M + \sqrt{M^2 + T^2}\right]$$

Equivalent twisting moment

$$T_e = \sqrt{238^2 + 32^2} = 240.16Nm$$

Equivalent bending moment

$T_e = 240.16Nm$

$$M_e = 0.5(238 + 32) = 135Nm$$

The diameter of the driving shaft from equivalent twisting moment is given by:

$M_e = 135Nm$

$$d_s = \sqrt[3]{\frac{16T_e}{\pi \times \tau_{\max}}}$$

$$d_s = \sqrt[3]{\frac{16 \times 240.16}{\pi \times 23.76 \times 10^6}} = 0.03720\,m$$

The diameter of the driving shaft from equivalent bending moment is given by:

	$$d_s = \sqrt[3]{\frac{32M_e}{\pi \times \tau_{\max}}}$$ $$d_s = \sqrt[3]{\frac{32 \times 135}{\pi \times 23.76 \times 10^6}} = 0.03868m$$ The maximum diameter is selected as the design diameter of driving shaft $= 0.03868m = 38.68mm \approx 40mm$	$d_s = 0.04m$
Bearings $N_{driving} = 116\ rpm$ $L_H = 24000 hours$ $W = 1000N$ $k = 3$ $K_s = 2.0$	**<u>Dynamic load rating</u>** $$L_{90} = 60\,NL_H$$ Hence, $$L_{90} = 60 \times 116 \times 24000 = 167.04 \times 10^6\ rev$$ The dynamic load rating is calculated from (3.14) $$C = W\left(\frac{L_{90}}{10^6}\right)^{1/k}$$ Which gives	$L_{90} = 167.04 \times 10^6\ rev$

	$C = 1000\left(\frac{167.04\times10^6}{10^6}\right)^{1/3} = 5507.32N = 5.507kN$ The design dynamic load rating is given by: $C_d = K \times C = 2.0 \times 5.507 = 11.014kN$	$C = 5.507kN$ $C_d = 11.014kN$
Belt $d_{driving} = 0.04m$ $d_{driven} = 0.12m$ $\mu = 0.3$ $Slip = 1.2$ $Frictionloss = 5\%$ $Overload = 20\%$ $P_{pn} = 130\,W$ $CD = 0.3m$ $N_{driving} = 116\ rpm$	**<u>Belt design</u>** From equation (2.22): $N_1/N_2 = d_2/d_1$ $N_{driven} = \frac{d_{driving}}{d_{driven}} \times N_{driving}$ $N_{driven} = \frac{0.04}{0.12} \times 116 = 38.67rpm$ Torque acting on driven shaft $T_{driven} = \frac{60P_s}{2\pi N_{driven}} = \frac{60\times130}{2\times\pi\times38.67} = 32.10Nm$ Actual torque due to friction $T_{actual} = (1 + frictionloss) \times T_{driven}$ $T_{actual} = 1.05 \times 32.10\,Nm = 33.71\,Nm$	$N_{driven} = 38.67\,rpm$ $T_{driven} = 32.10Nm$

For overload,

$T_{actual} = 33.71 Nm$

$$T_{design} = (1 + overload) \times T_{actual}$$

$$T_{design} = 1.2 \times 33.71 = 40.45 Nm$$

$T_{design} = 40.45 Nm$

$$T_1 - T_2 = \frac{T_{design}}{D_{driven}} = \frac{40.45}{0.12} = 337.08 Nm$$

$$\sin \alpha = \frac{R_{driven} - R_{driving}}{CD} = \frac{0.12 - 0.04}{0.3} = 0.2667$$

$$\alpha = \sin^{-1} 0.2667 = 0.2699 rad = 15.5^o$$

$$\theta = 180 - 2\alpha = 180 - (2 \times 15.5) = 150^o = 2.62 rad$$

$$\frac{T_1}{T_2} = e^{0.3 \times 2.60} = 2.62$$

$$T_1 = 545.15 Nm$$

$$T_2 = 208.07 Nm$$

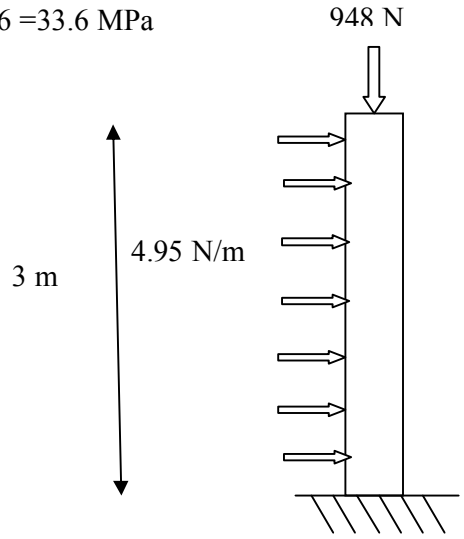
$T_1 = 545.15 Nm$

Length of belt

$T_2 = 208.07 Nm$

$$l = ((0.5\pi \times (D_{driven} + D_{driving}) + (2 \times CD) + \frac{(D_{driven} - D_{driving})^2}{4 \times CD} = 0.90 m$$

$l = 0.90 m$

Tail vane $d_r = 2.50\,m$	**<u>Surface area of the tail vane</u>** The design surface area of the tail vane is obtained from equation (3.15): $A_t = \frac{d_r^{\,2}}{40}$ $A_t = \frac{(2.50)^2}{40} = 0.1554m^2 \approx 0.16m^2$	$A_t = 0.16m^2$
Tower Maximum tensile strength = 560 MPa Factor of safety = 6 $\rho_a = 1.147kg/m^3$ Maximum wind velocity = 10 m/s	Ultimate tensile strength(UTS) = 36% of Maximum tensile strength= 201.6 MPa Allowable tensile stress(σ)= UTS/factor of safety = 201.6 MPa/6 =33.6 MPa  The bending moment across a section x of the shaft is given by: $M_x = 4.95\frac{x^2}{2}$	

	The maximum bending moment =7.425Nm (occurs at x = 1.5 m) The minimum tower diameter is obtained from the equation below (Budynas and Nismett, 2011): $$d = \sqrt[3]{\frac{32M}{\pi\sigma_{\max}}}$$ $$d = \sqrt[3]{\frac{32\times 7.425}{\pi\times 33.6\times 10^{6}}} = 0.0131m$$ Minimum allowable diameter is therefore 1.3 cm	
Storage medium	A carbon steel cylinder of volume 30 m^3 and pressure rating 6 barg was selected as a storage medium.	

3.6 The Wind compressor system design

3.6.1 Design calculations

Design calculations for the wind compressor are as shown in section 3.5. All the major aspects of the wind compressor system were captured in the design.

3.6.2 Design drawings

Detailed design drawings for the wind compressor system components as well as the assembly are as shown in Appendix B. These drawings were done using AutoCAD.

3.7 System Component Construction

3.7.1 Rotor Blades

Six rotor blades of 1.25 m length were produced. The steps as recommended by (Piggot, 2003) were followed carefully for carving each blade to ensure same shape for all blades to produce the curved blade profiles. The tools used were a saw, plane, calipers, compasses, tape measure, ruler, pencil. The flat surface with a setting angle of four degrees will face the wind with the curved surface behind. Holes were drilled on the root of the blades for bolting to the hub. Plate C1 in appendix C shows a picture of the rotor blade.

3.7.2 Rotor Hub

A 256 mm flywheel is used as the rotor hub. Six metal plates of dimension 50 mm X 60 mm X 1.5 mm were welded on the flywheel at 60 degrees spacing around the hub. Holes were drilled on all metal sheets on equivalent points as in the blade roots. The middle hole is used for mounting the hub onto the shaft. Plate C2 shows a picture of the hub.

3.7.3 Rotor Shaft

The rotor shaft was made by cutting, turning, milling and polishing solid mild steel to a diameter of 0.04 m. Key holes were made to lock the bearings onto the shaft. The rotor shaft has a thread at the head for bolting the rotor hub firmly on the shaft; this would enable the assembly to rotate securely as a unit.

3.7.4 Bearings

A single row deep groove ball bearing with static and dynamic radial load capacity of 15.6 kN and 24 kN respectively (bearing number 404) was selected as the design dynamic radial load was calculated as 11.014 kN.

3.7.5 Belt and Pulley

A driving pulley of diameter 0.12 m and driven pulley of diameter 0.04 m were purchased to allow for speed reduction. A Vee belt of 1m length was purchased off shelf to fit the pulleys. Plate C3 on appendix C shows a picture of the belt and pulley system.

3.7.6 Mounting Plate

A 40 mm by 60 mm by 30 mm plate on which the wind compressor system would sit was constructed from a steel plate. The ball bearings as well as the compressor sit on the mounting plate.

3.7.7 Tail vane

The tail vane was manufactured from a steel plate into a hexagonal shape with a surface area of 0.16 m^2. This hexagonal sheet was welded onto a boom made of a rectangular steel plate of dimensions 120 X 50 X 30 mm. The other end of the boom is attached to the mounting plate by a welded joint. Plate C4 on appendix C shows a picture of the tail vane.

3.7.8 Tower

An existing hollow steel pipe of outer diameter 150 mm and inner diameter 145 mm and length 3 m was used as tower. The tower is located on top of the mechanical engineering Department at a height of about 15m. The base of the tower was welded to a thick metal sheet that sits on a concrete foundation. The metal sheet was bolted to the foundation. The tower was assessed as having the ability to withstand the high wind and rotor thrust loads during the system operation (a minimum diameter tower of 23 mm is required to withstand these stresses). It also provides easy access to the rotor for maintenance and repair.

3.8 Cost Evaluation

The cost of the wind compressor was determined based on material, labour and overhead costs. The material cost involves a summation of all materials used for the construction.

Labour cost is the cost of services rendered for hours spent on the production process by all involved parties. Overhead costs include transportation costs and all other miscellaneous costs incurred by the construction work. The detailed costs are as in Tables 3.2 and 3.3.

Table 3.2 Material costs

S/N	Component	Material	Dimension (mm)	Quantity	Unit cost (₦)	Total Cost(₦)
1	Rotor Blades	PVC pipe	Φ30X3000 mm	3	2000	6000
2	Hub	Steel Alloy 1060	Φ256 mm	1	1500	1500
3	Rotor shaft	Steel Alloy 1045	Φ30X2000 mm	1	2500	2500
4	Metal sheets	Mild steel	1500X800X2.5 600X300X2.0	1 1	3500 1500	3500 1500
5	Metal plate	Mild steel	600X300X2.0	1	1500	1500
6	Frame 2X2 angle iron	Mild steel	300	1	1500	1500
7	Deep groove single row ball	UTP	Φ22	2	1500	3000
8	Bolts, nuts and washers	Grade 4.8 steel			2300	2300
9	Tail vane Metal sheet	Mild steel	400X400	1	2000	2000
10	Boom	Frame 2X2 angle iron	300	1	1500	1500
11	Air compressor	Volvo BW6 car air compressor		1	6500	6500

Total Material cost =₦33,300

Table 3.3 Labour and Miscellaneous costs

S/N	Labour type	Description	Cost(□)
1	Rotor blades preparation	Skilled Labour Cutting, polishing, and finishing of PVC pipes to required blade profile as in detailed drawing provided.	6000
2	Hub production	Skilled Labour	2500
3	Linkage mechanism production	Skilled Labour	2500

4	Tail vane production	Skilled Labour	3500
5	Transportation Costs	Unskilled labour	3000
6	Installation costs	Skilled Labour	6000
7	Other costs		3000
8	Miscellaneous costs		1350

TotalLabour and miscellaneous costs = ₦27,850.00

Grand Total = ₦61,150.00

The installed capital costs for the 130W wind air compressor was **₦61,150.00**

3.9 Installing and testing the system

The various components were put together to form the 130W wind compressor system as shown in plates C5-C7 on appendix C. The installation was done on top of the roof of Mechanical Engineering Department's building at a height of about 15m.

3.9.1 Testing Set up

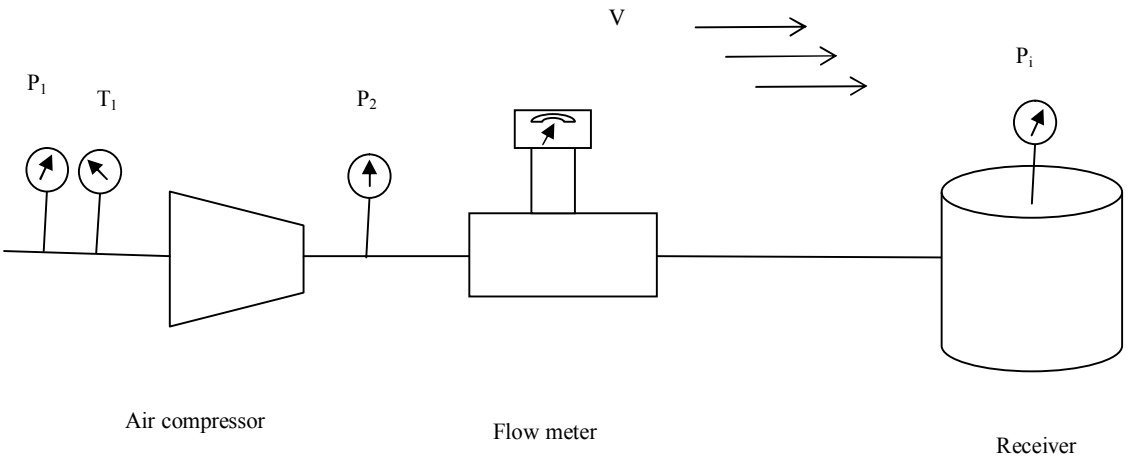


Fig 3.2: Testing set up

3.9.2 Testing Procedure

The installed system operates as detailed in section 3.1 on page 57.

Fig 3.2 shows the set up used for testing the wind air compressor system. The wind speed was measured using a hand held anemometer. The Compressor inlet pressure (P_1) and temperature (T_1) were measured using a pressure gauge and a temperature gauge respectively installed on the inlet line into the compressor. The Compressor discharge pressure (P_2) was measured using a pressure gauge installed on the discharge line from the compressor. The compressor flow rate was determined using an inline variable area flow meter. The air pressure in the receiver (P_i) was obtained using a pressure gauge on it.

Readings of the instantaneous wind velocities at different times were taken at five minutes intervals over an hour period and the corresponding readings for the discharge pressure and the flow rate were taken as in test set up. The readings taken over the specified period are presented in the results in section 4.1 on page 80.

Plates C8-C10 on Appendix C shows some pictures of the testing process.

3.10 Calculations

3.10.1 Energy Pattern factor determination for Zaria

The energy pattern factor for Zaria at a height of 30m was determined based on data from Pam (2008) as in appendix A.

Table 3.4: Energy Pattern factor determination at 30m height for Zaria

Month	**Average wind velocity at 30m height**	**Cube of Wind speed $(m/s)^3$**
January	4.39944	85.1516
February	4.15635	71.8018
March	3.52002	43.6149
April	3.47085	41.8127

May	3.72463	51.6715
June	3.61087	47.08
July	3.43033	40.3652
August	2.72058	20.1365
September	2.3579	13.1092
October	2.45229	14.7474
November	3.02471	27.6726
December	3.9242	60.4303
Annual	3.399348	43.13281

The energy pattern factor, k_e for Zaria at 30 m height is obtained as:

$$k_e = \frac{\overline{V^3}}{\overline{V}^3} = \frac{43.13281}{(3.399348)^3} = 1.10$$

Where

k_e = energy pattern factor of a location

$\overline{V^3}$ = mean cube of wind speeds within the period

$\overline{V}$ = mean of wind speeds within the same period

3.10.2 Power Output and Overall efficiency determination for wind air compressor

The output power as well as the overall efficiency of the wind air compressor system is determined below based on results of testing obtained for Period 3 as presented in table 4.4 on page 82.

The Overall efficiency is calculated from equation (3.6)

$$\eta_{overall} = 7105\frac{\dot{v}T_1\left(r^{0.29}-1\right)}{d_r{}^2V^3}$$

At a wind velocity of 4.90m/s,

$\dot{v} = 0.000397m^3/s$

$T_1 = 27^oC = (27+273) = 300K$

$r = \frac{2.0bara}{1.0bara} = 2.0$

$d_r = 2.50m$

$V = 4.90m/s$

Hence,

$$\eta_{overall} = 7105\times\frac{0.000397\times300\left(2^{0.29}-1\right)}{(2.50)^2(4.90)^3} = 0.2562$$

At a wind velocity of 5.70m/s,

$\dot{v} = 0.000421m^3/s$

$T_1 = 27^oC = (27+273) = 300K$

$r = \frac{2.2bara}{1.0bara} = 2.0$

$d_r = 2.50m$

$V = 5.70m/s$

Hence,

$$\eta_{overall} = 7105 \times \frac{0.000421 \times 300\left(2.2^{0.29} - 1\right)}{(2.50)^2 (5.70)^3} = 0.1992$$

Interpolating, the overall efficiency at the rated wind velocity was calculated as follows:

At $V = 4.90 m/s$, $\eta_{overall} = 0.2562$

At $V = 5.70 m/s$, $\eta_{overall} = 0.1992$

Hence,
At $V = 5.10 m/s$, $\eta_{overall}$ is calculated as

$$\eta_{overall_{v=5.10}} = 0.2545 + \left(\left(\frac{5.10 - 4.90}{5.70 - 4.90}\right) \times (0.1992 - 0.2562)\right) = 0.2420$$

The Power was calculated from equation (3.9) as:

$$\overline{P_{pn}} = 0.4504 \overline{\eta_{overall}} k_e d^2 \overline{V}^3$$

At a wind velocity of 4.90m/s,

$\eta_{overall} = 0.2562$

$k_e = 1.10$ For Zaria (Section 3.10.1)

$d_r = 2.50m$

$V = 4.90 m/s$

Hence,

$$\overline{P_{pn}} = 0.4504 \times 0.2545 \times 1.10 \times (2.50)^2 \times (4.90)^3 = 93.34W$$

At a wind velocity of 5.70m/s,

$\eta_{overall} = 0.1992$

$k_e = 1.10$ For Zaria (Section 3.10.1)

$d = 2.50m$

$V = 5.70m/s$

Hence,

$\overline{P_{pn}} = 0.4504 \times 0.1978 \times 1.39 \times (2.50)^2 \times (5.70)^3 = 114.22W$

Interpolating, the available Power at the rated wind velocity was calculated as follows:

At $V = 4.90m/s$, $P = 93.34W$

At $V = 5.70m/s$, $P = 114.22W$

Hence,
At $V = 5.10m/s$, P is calculated as

$$P_{v=5.10} = 92.72 + \left(\left(\frac{5.10 - 4.90}{5.70 - 4.90} \right) \times (114.22 - 93.34 \right) = 98.56W$$

CHAPTER FOUR

RESULTS AND DISCUSSIONS

4.1 Test results and discussions

4.1.1 Results

The air compressor system was test run and the performance observed. Test results are presented in tables 4.1, 4.2 and 4.3.

Table 4.1 Readings taken on 4th June, 2013 at 9 a.m.(Period 1)

SN	Wind velocity (m/s)	Discharge pressure (bara)	Flow rate (m^3/s)
1	1.2	1.0	0.000000
2	2.0	1.0	0.000000
3	2.4	1.0	0.000000
4	2.8	1.4	0.000151
5	3.4	1.5	0.000198
6	3.9	1.7	0.000238
7	4.6	1.9	0.000349
8	5.3	2.0	0.000397
9	5.9	2.0	0.000397
10	6.8	2.4	0.000595
11	7.5	2.5	0.000659
12	7.8	2.6	0.0006349

Inlet Temperature = 23 ^oC
Inlet Pressure = 1.0 bara

Table 4.2 Readings taken on 5th June, 2013 at 5 p.m. (Period 2)

SN	Wind velocity (m/s)	Discharge pressure (bara)	Flow rate (m^3/s)
1	2.90	1.40	0.000127
2	3.50	1.70	0.000238
3	4.30	1.90	0.000476
4	4.70	2.00	0.000476
5	5.00	2.10	0.000556
6	5.70	2.20	0.000476
7	5.90	2.30	0.000476
8	6.10	2.30	0.000476
9	6.60	2.50	0.000635
10	7.00	2.50	0.000635
11	7.10	2.70	0.000738
12	7.90	3.00	0.000833

Inlet Temperature = 25 ^oC
Inlet Pressure = 1.0 bara

Table 4.3 Readings taken on 6th June, 2013 at 12 p.m. (Period 3)

SN	Wind velocity (m/s)	Discharge pressure (bara)	Flow rate (m^3/s)
1	0.7	1.0	0.000000
2	1.8	1.0	0.000000
3	2.7	1.2	0.000127
4	3.3	1.4	0.000159
5	3.7	1.5	0.000286
6	4.3	1.9	0.000317
7	4.8	2.0	0.000381
8	4.9	2.0	0.000397
9	5.7	2.2	0.000421
10	6.7	2.2	0.000437
11	8.1	2.5	0.000484
12	9.2	2.8	0.000500

Inlet Temperature = 27 ^oC
Inlet Pressure = 1.0 bara

Table 4.4 Power and Overall efficiency for wind air compressor for period 3

SN	Wind velocity (m/s)	Discharge pressure (barg)	Discharge pressure (bara)	Flow rate (m^3/s)	Efficiency (%)	Power(W)
1	0.7	0.0	1.0	0.000000	0.00	0.00
2	1.8	0.0	1.0	0.000000	0.00	0.00
3	2.7	0.2	1.2	0.000127	11.87	7.23
4	3.3	0.4	1.4	0.000159	15.36	17.10
5	3.7	0.5	1.5	0.000286	23.87	37.43
6	4.3	0.9	1.9	0.000317	27.63	68.03
7	4.8	1.0	2.0	0.000381	25.98	88.98
8	4.9	1.0	2.0	0.000397	25.45	92.72
9	5.7	1.2	2.2	0.000421	19.78	113.46
10	6.7	1.2	2.2	0.000437	12.65	117.77
11	8.1	1.5	2.5	0.000484	9.39	154.53
12	9.2	1.8	2.8	0.000500	7.57	182.50

From the results, graphs of compressor flow rate and discharge pressure were plotted against the instantaneous wind velocity and are shown in figure 4.1 and figure 4.2.

Fig 4.3 shows a plot of the compressor flow rate and discharge pressure versus wind air velocity for period 3.

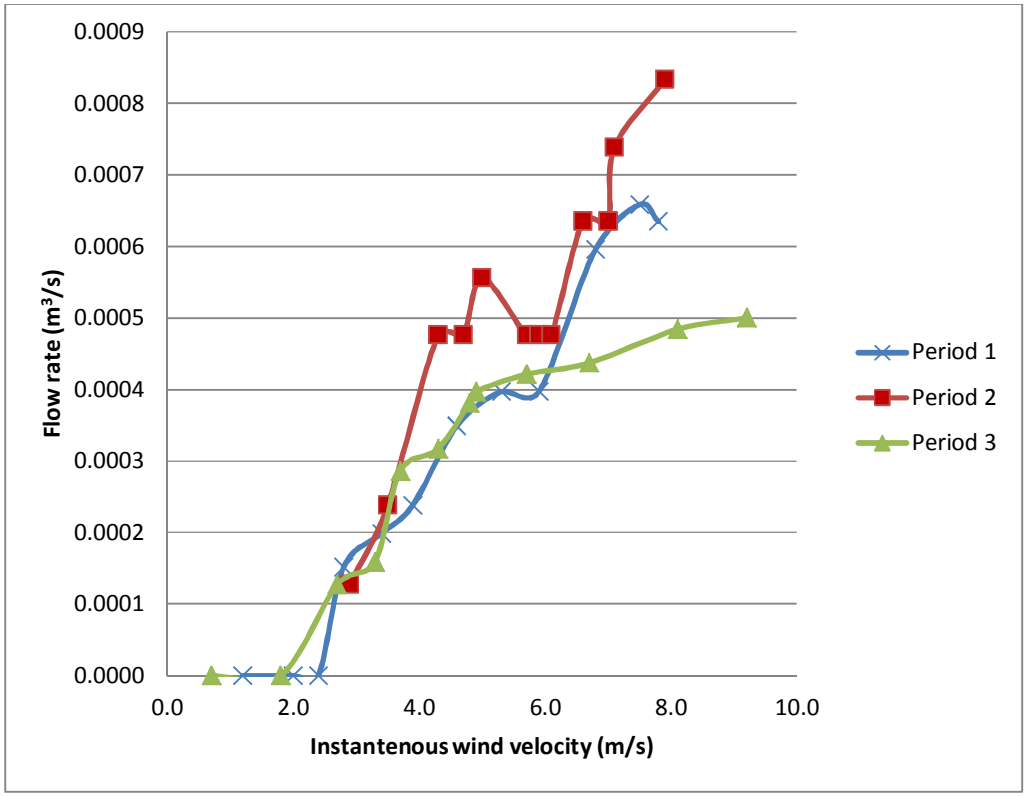


Fig 4.1: Flow rate profile for wind air compressor system at various test periods

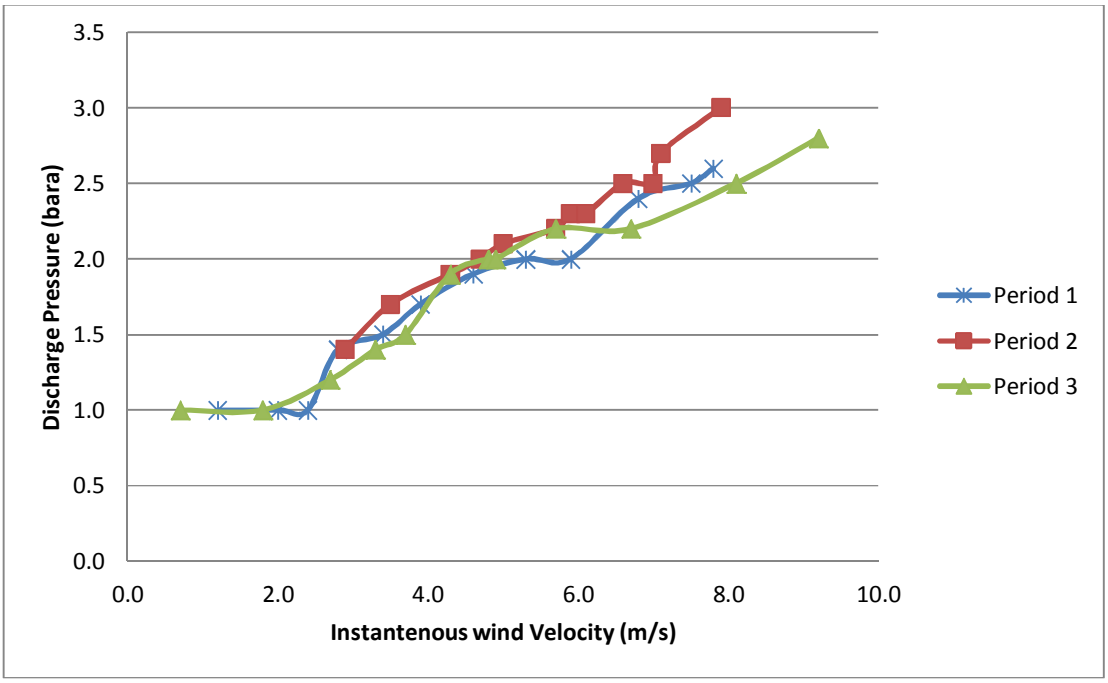


Fig 4.2: Discharge pressure profile for wind air compressor system at various test periods

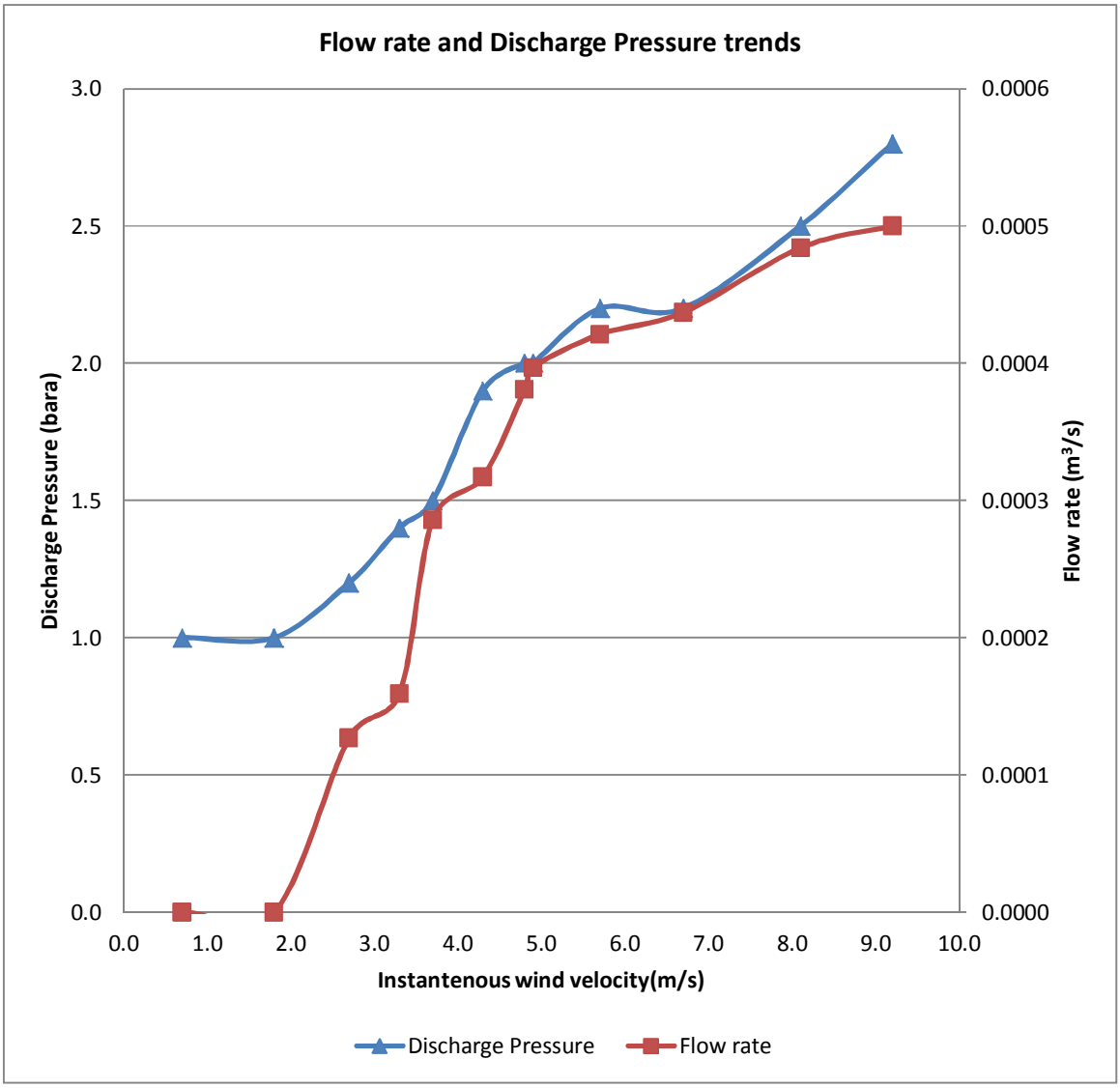


Fig. 4.3: Graph showing trends of discharge pressure and flow rates versus the wind velocity

The actual power available at the compressor as well as the overall efficiency for period 3 is calculated as in section 3.10.2.

A graph of the power output and the overall efficiency against the instantaneous velocity, and the flow rates for period 3 are plotted for the calculated rotor power and the overall efficiency and are shown in figure 4.4 and figure 4.5.

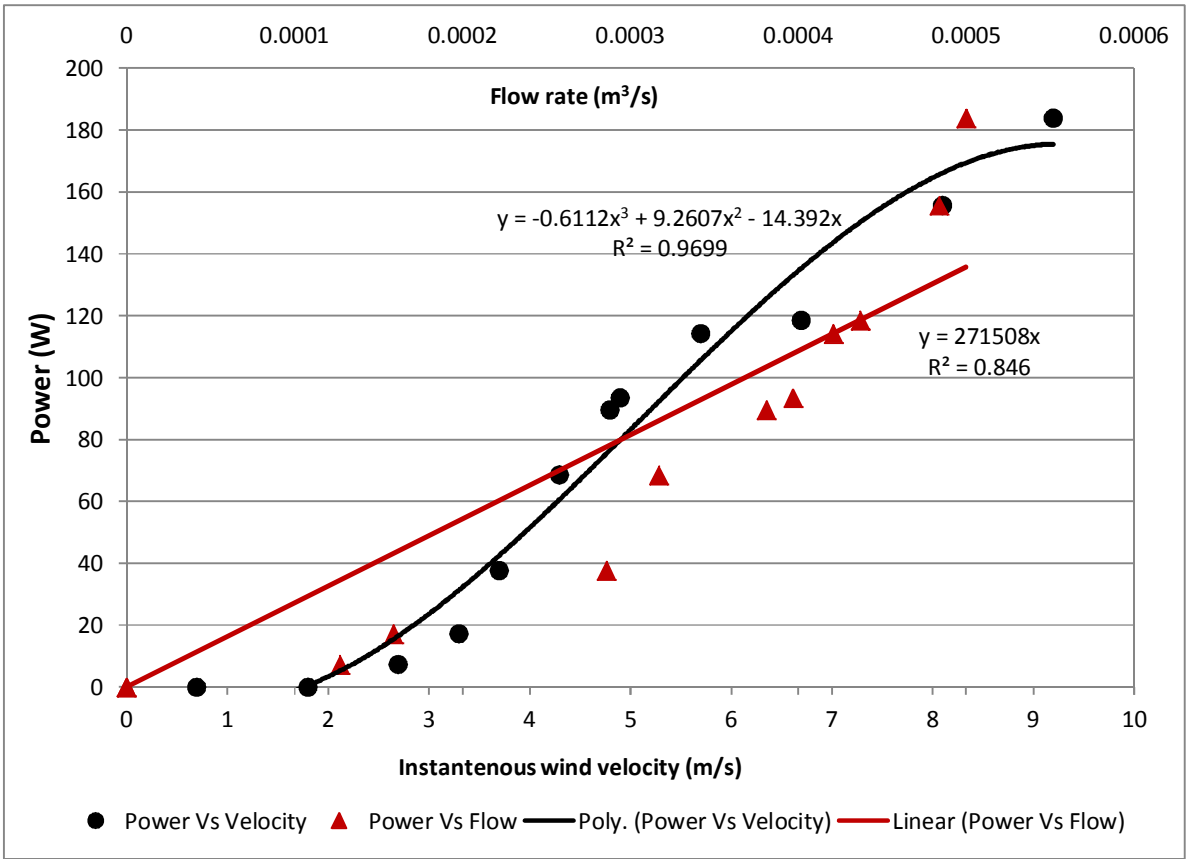


Fig 4.4: Variation of power output with instantaneous wind velocity and air flow rates

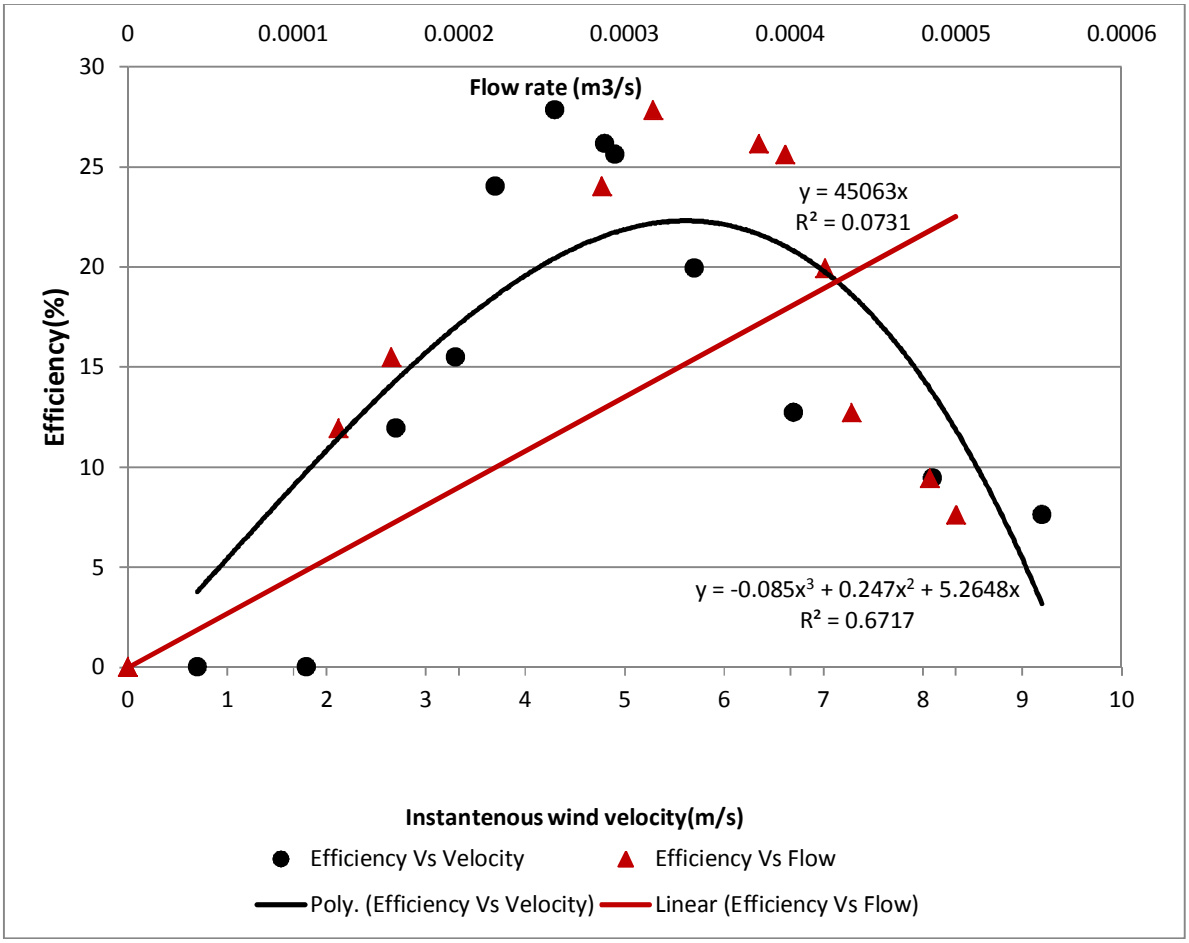


Fig 4.5: Variation of Overall efficiency with instantaneous wind velocity and air flow rates

4.1.2 Discussion of results

From figure 4.1, the flow rate profile for the wind air compressor system, it can be seen that the compressor flow rate increased with an increase in the wind velocity. This is in agreement with established theories for wind pumps as presented by Abed, 1997 for example. This is particularly true because higher wind velocities result in more power at the compressor shaft which leads to more air being compressed hence a higher flow rate.

From figure 4.2, the discharge pressure profile for the wind air compressor system, it can be seen that the compressor discharge pressure increases as the wind velocity increases. At the rated wind velocity of 5.10 m/s, an average discharge pressure of 1.05 barg was obtained.

From figure 4.3, a graph showing the flow rate and discharge pressure versus the wind velocity, it is seen that at an instantaneous wind velocity of 9.2 m/s, a flow rate of 0.0005 m^3/s is obtained at a discharge pressure of 2.8 barg.

Figure 4.4 show a graph of power output from the compressor based on differing wind velocities and compressor flow rates. The graph follows the cubic trend of the extractable wind power for Power against Instantaneous wind velocity; it shows that the power output from the wind rotor increased as the wind velocity increased. At the rated wind velocity of 5.10 m/s, the power output was calculated as 98.56 W for period 3. This was less than the expected 130 W most probably due to unconsidered losses. For the graph of Power against Compressor flow rates; it shows that the power output from the wind rotor increased linearly as the flow rates increased. This is particularly true as more power equals more production.

Figure 4.5 show a graph of calculated overall efficiency against the wind velocity and compressor flow rates. As seen the overall efficiency increased as the wind velocity increased until it reached a maximum and then decreased gradually. An overall efficiency range of 7.6 to 27.6 % was obtained from the test data for period 3. This is comparable to an overall efficiency range of 16 to 22 % as obtained by Abed (1997) for a system consisting of a wind turbine, compressor, storage tank, and air-lift pump.

At the nominal power output, the overall efficiency of the wind air compressor for period 3 was found to be 0.242 or 24.2 % as opposed to the design efficiency of 35 %.

CHAPTER FIVE

SUMMARY, CONCLUSIONS, LIMITATIONS AND RECOMMENDATIONS

5.1 Summary

i. A wind air compressor system to power a 130W compressor was designed, constructed and tested at Zaria, Nigeria.

ii. A discharge pressure of 1.0 barg (2.0bara) was obtained during the testing period at the rated wind velocity of 5.10 m/s.

iii. A maximum compressor capacity of 0.000833 m^3/s at a discharge pressure of 2.0 barg (3.0bara) was obtained at an instantaneous velocity of 7.90 m/s.

iv. At the rated wind velocity of 5.10 m/s, the power output was calculated as 97.91 W.

v. The overall efficiency increased as the wind velocity increased until it reached a maximum at the rated wind velocity and then decreased gradually. An overall efficiency range of 7.6 to 27.6 % was obtained from the test data.

vi. The mean overall system efficiency at the rated wind velocity was obtained as 24.2 % as opposed to the design efficiency of 35 %.

5.2 Conclusions

i. A design analysis was carried out for the wind rotor and transmission mechanism for the wind- air compressor system.

ii. Appropriate materials were selected for the wind air compressor system.

iii. Components for the wind air compressor system were constructed or procured as detailed in construction report.

iv. The compressor discharge pressure and flow rate per unit time increased with an increase in the wind velocity.

v. The overall efficiency increased as the wind velocity increased until it reached a maximum of 27.6 % and then started to decrease gradually thereafter to a minimum of 7.6 %.

vi. The installed capital cost of the system was evaluated as sixty one thousand, one hundred and fifty naira only. (61, 150).

5.3 Recommendations

i. The use of a temperature, pressure and flow transmitters in conjunction with a data logger to obtain concurrent readings to avoid synchronisation issues as well avoid difficulties in taking readings during odd times of the day/night (Example at 3am in the night) is recommended.

ii. A control mechanism should be designed for the system to withstand speeds above the cut out speed.

iii. Further research on the wind air compressor, in particular the effect of the number on blades on the compressor output should be carried out.

5.4 Contributions to knowledge

i. The use of a belt and pulley system transmission system was adopted for the design in order to reduce the outage time during maintenance. A time saving of about 12 hours (1.5 work days) is therefore archived compared to the use of traditional piston and cylinder transmission system.

ii. Designed and constructed a 130W wind powered air compressor system that couples the compressor directly to the wind rotor.

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APPENDIX A

Wind data for Zaria

Month	Average wind velocity at 10m height(m/s)	Average wind velocity at 30m height	Average wind velocity at 50m height	Average wind velocity at 100m height
January	3.760443034	4.3994422	4.73249626	5.22510099
February	3.55264449	4.1563468	4.47099781	4.93638039
March	3.00874209	3.52001834	3.78649831	4.180663307
April	2.96671629	3.47085118	3.7336092	4.12223959
May	3.18363881	3.72463417	4.00660563	4.42365122
June	3.08640027	3.61087251	3.88423228	4.28853989
July	2.93207908	3.43032885	3.690001985	4.07411337
August	2.32541966	2.72058105	2.92653871	3.23116016
September	2.01541901	2.35790133	2.53640246	2.80041552
October	2.09609723	2.4522891	2.6379354	2.91251731
November	2.58537412	3.02470756	3.25369048	3.59236622
December	3.35422059	3.92420411	4.22128296	4.66067505
Annual	2.90103245	3.39400506	3.65094399	4.03097153

Source: (Pam, 2008)

APPENDIX B

DESIGN DRAWINGS

APPENDIX C

PLATES



Plate C1: Rotor Blade



Plate C2: Rotor Hub

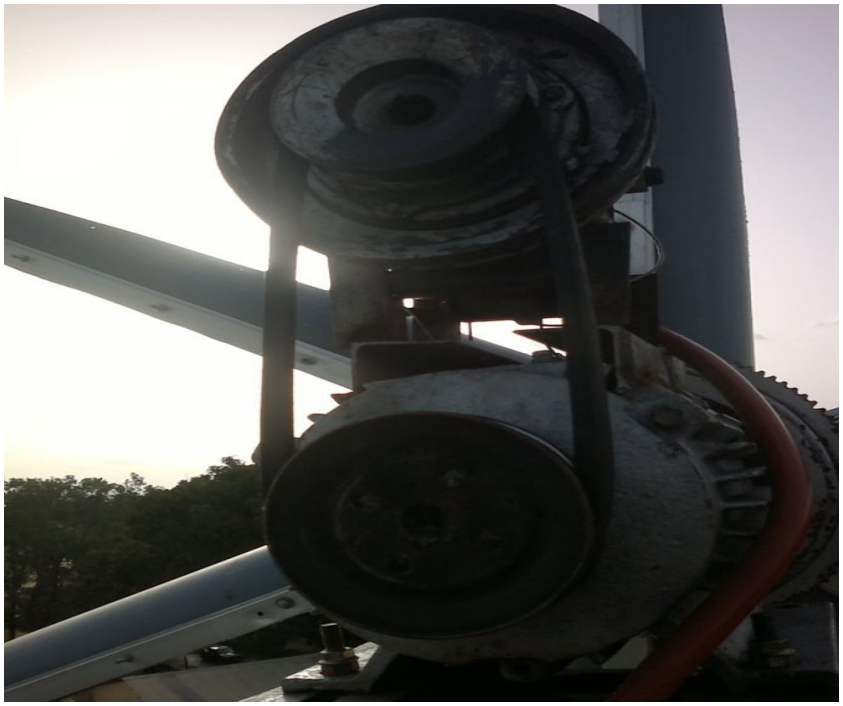


Plate C3: Belt and pulley system



Plate C4: Tail Vane



Plate C5: The complete wind air compressor system at mechanical engineering department roof (View 1)



Plate C6: The complete wind air compressor system at mechanical engineering department roof (View 2)



Plate C7: The complete wind air compressor system at mechanical engineering department roof (View 3)



Plate C8: Testing in Progress



Plate C9: Test kit (Pressure Gauge and anemometer)

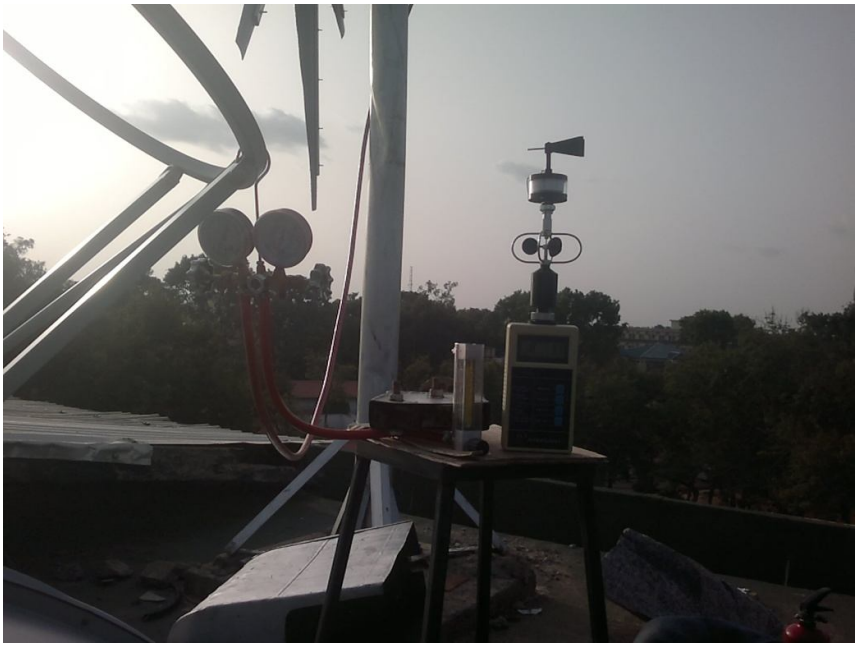


Plate C10: Anemometer for measuring wind speed