Declaration

Unless where stated and acknowledged, we declare that the project report which includes research work, findings and discussions and conclusions is purely of our effort and is original. The project details entailed in our report have not been presented before to the best of our knowledge.

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Dedication

We dedicate this project firstly to our parents for the continuous support, both financially and emotionally. Their support has come a long way in giving us the motivation to complete this project in time and to the best of our abilities. We would also like to dedicate this project to our close friends for their inspiration and support, pushing us on through the tough times.
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We would like to thank the following parties:

First and foremost we would like to express our gratitude to God almighty for giving us the strength, hope, courage and inspiration to successfully complete this project and without Whom we would not have made it this far.

We would also like to express our sincere appreciation to our project supervisor and lecturer Mr. M. Mwaka for his guidance and advice throughout the project.

We also recognize the assistance of Mr. James Wafula, from the Nuclear Science Department, and our fellow colleague Mr. Kamau King’ora in attaining the laboratory apparatus and running the wind tunnel tests.
List of Symbols

A = area swept by the rotor (m$^2$)

$C_t$ = the co-efficient of torque

$C_p$ = the co-efficient of power

$g$ = gravitational acceleration (9.81 m/s$^2$)

$H$ = head (m)

$H_g$ = gross head (m)

$m$ = mass of flow rate (kg/s)

$N$ = rotational speed (rpm)

$P_t$ = power in turbine shaft (W)

$P_w$ = wind power due to its kinetic energy (W)

$Q$ = flow rate (m$^3$/s)

$R = rotor radius = \frac{D}{2}$ in m

$T$ = Torque (Nm)

$F$ = force (N)

$U$ = wind velocity (m/s)

$W_p$ = weight of the piston and pump rods (N)

$\eta$ = total system efficiency at max flow rate

$\rho_a$ = density of air in kg/m$^3$ (1.2kg/m$^3$)

$\rho_w$ = density of water (1000 kg/m$^3$)
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Abstract

Research and development activities in the field of renewable energy have been considerably increased in many countries recently, due to the worldwide energy crisis. Wind energy is becoming particularly important. Although considerable progress have already been achieved, the available technical design is not yet adequate to develop reliable wind energy converters for conditions corresponding to low wind speeds and urban areas. The Savonius turbine appears to be particularly promising for such conditions, but suffers from a poor efficiency. The present study considers improved designs in order to increase the output power of a classical Savonius turbine. It aims at improving the output power of the Savonius turbine as well as its static torque, which measures the self-starting capability of the turbine.

Our objective was to investigate the characteristics of Savonius turbine under varying wind speeds, and obtain the co-efficient of power that can be produced from it, and thus its relevant applications. An experiment carried out in the Fluids lab on a model version of the Savonius turbine, during which measurements of rotor speed, current and voltage were taken at each wind speed, enabling the calculation of power and tip speed ratio. After obtaining experimental values of power, tip speed ratio and coefficient of power, they were compared to theoretical values and we were able to determine the relationships between power and wind speed, co-efficient of power vs. tip speed ratio, (in the two cases with and without air gaps) among others. From the resulting graphs the maximum amount of coefficient of power, $C_p$ was obtained as 0.14 deeming the experiment a success since the accepted values are between 0.1 and 0.37.

From the results obtained, dimensional analysis was done, up-scaling to estimate design parameters for a possible prototype to be made in the future. It was seen that the most viable application would be in the pumping of water, especially in arid and semi-arid areas which have relatively high average wind speeds over 6m/s. Various recommendations were then made on improving on the Savonius coefficient of power and static torque, hence its overall efficiency.

On an aerodynamic performance basis, the Savonius rotor cannot generally compete with other types of wind turbines. This is entirely due to its mode of operation. Unlike its counterparts that operate by rotating around a horizontal axis, it rotates around a vertical axis. This has the unfortunate effect of lowering its efficiency, but it has several compensating factors. Its main advantages are that it has better starting torque performance with operating characteristics independent of the wind direction. In addition, it is simple in structure and the fabrication technology required is less sophisticated when compared to similar types of windmills. This makes it a suitable system for small scale applications in wind energy conversion; especially in remote rural regions in developing countries.
Chapter 1

1 Introduction
Wind has been used as a source of energy for many centuries. It has always provided sufficient propulsive power to ships and to run windmills; for example, the American farm wind pump. However, with the invention and spread of other sources of power, especially electricity and fossil fuels, the use of wind as a source of energy declined.

Recently, with attention drawn to climate change, demand to increase green energy production and manage finite fossil fuel supplies, there has been renewed interest in alternative renewable energy sources. Wind energy requires relatively lower investment and uses a natural energy resource that is widely available lowering the entry barrier for application. In fact, the exploration of wind power could lead to the development of the arid and semi-arid areas which, by virtue of low plant cover, experience high maximum and average wind speeds. These areas could become fertile for deployment of wind farms for the generation of electricity or the pumping of borehole water.

Water has always been a primary human need and its access has affected the settlement of population. In Africa, in particular, most of the population resides in the rural areas where piped water is non-existent. This means that for the most part, it is women and children who take over the task of searching for water for domestic purposes. This chore can take hours of time and involve trekking over large distances. Water scarcity affects the local economy in the following ways:

1. **Agriculture** – water is important for the cultivation of crops and keeping of livestock. Scarcity of water results in poor returns from both because of low crop yields with low nutritional value, unhealthy livestock and the large amounts of time taken in searching for the resource subtracts from time spent working on farms.

2. **Health**. It is also common to find that the human and livestock population share a single source of water in rural areas. In pastoral communities this means that such water sources are defiled by animals, which puts the local community at a serious health risk from water borne diseases. Poor crop and livestock yields also caused by a lack of water have a direct influence on the health of the local population.

3. **Education** – Time spent by children searching for water is time spent away from schools. For most African communities, it is especially the girls who will help their mothers in this task in favour of attending school. This leads to a gender educational imbalance which is a waste of human resource. This has the larger impact of affecting the social-economic development of a country and the ability of its government to adequately plan for the future.

The water crisis in Kenya is also disrupting social and economic activities throughout the country. Unfortunately, the current wave of droughts and water shortages in Kenya and the rest of East Africa are only expected to continue. The water crisis is due not only to the wave of droughts, but
also to poor management of the water supply, under-investment, unfair allocation of water, rampant deforestation, pollution of water supplies by untreated sewage, and a huge population explosion (thirty-fold increase since 1900).

Kenya is limited by an annual renewable fresh water supply of only 647 m$^3$ per capita, and is classified as a water scarce country. The total yearly water withdrawal is estimated to be over 2.7 km$^3$, or less than 14% of resources. [1]

However, water resources availability varies significantly in time and between regions. Most parts of the country have two rainy seasons. The long rains are typically from March to May while short rains are typically from October to November. In addition, the country experiences every three to four years droughts and floods, which affect a large number of the population. The latest severe drought was from 2007 to the end of 2009, which had impacts on all sectors of the economy. The average annual rainfall is 630 mm, but it varies between less than 200 mm in northern Kenya to over 1,800 mm on the slopes of Mount Kenya.

Only 57% of the rural population has access to an improved drinking water source, and the time-intensive pursuit of water collection often prevents women from taking up income generating activities, or in the case of girls, prevents them from attending school.

Estimates from the Joint Monitoring Programme for Water Supply and Sanitation show that in 2008 59% of Kenyans (83% in urban areas and 52% in rural areas) had access to improved drinking water sources. 19% of Kenyans (44% in urban areas and 12% in rural areas) are reported as having access to piped water through a house or yard connection. According to the JMP estimates, access to improved water sources in urban areas decreased from 91% in 1990 to 83% in 2008. In rural areas, however, access increased from 32% to 52% during the same period [2]. According to a different definition called "weighted access", the 2009 Impact Report estimates that in 2006-2007 only 37% of Kenyans had access to sufficient and safe drinking water close to their homes at an affordable price. Significant regional differences in access were reported: the highest level was registered in the area served by Tetu Aberdare Water and Sanitation Company (72%) whereas the lowest was recorded in Muthambi in Meru South District (4%). In the capital Nairobi access for the same period was reported at 35%, as opposed to a less realistic figure of 46% reported for 2005-2006. [3]

From these figures, it is clear that water supply is an important issue affecting the development of the country, especially in the rural areas. Our project is based on testing the applicability of the Savonius wind pump in providing sufficient power for pumping water in arid and semi-arid rural areas.
Chapter 2

2 Literature Review

2.1 Wind Power

2.1.1 Background
The wind has been used for many centuries; it was in fact the primary method used for dewatering large areas of the Netherlands from the 13th century onwards. Smaller wind pumps, generally made from wood, for use to dewater polders, (in Holland) and for pumping sea water in salt workings, (France, Spain and Portugal), were also widely used in Europe and are still used in places like Cape Verde.

![Figure 1 Basic Wind Turbine](image)

The most common wind pump, however, is the American Farm Wind pump (Figure 2). This normally has a steel, multi-bladed, fan-like rotor, which drives a reciprocating pump linkage.
usually via reduction gearing that connects directly with a piston pump located in a borehole directly below. This pump arose from vast demand for water-lifting machinery brought about by the introduction of cattle ranches on the North American Great Plains which had scant surface water. This wind pump evolved with the backing of the US government sponsorship \textsuperscript{[4]} to develop better wind pumps for irrigation and water supply. The American Farm Wind pump was also deployed in Argentina and Australia with great success.

![American Farm Wind Pump](image)

**Figure 2 the American Farm Wind Pump**

Wind pumps have also been used in Europe and Asia, e.g. the Chinese sail wind pump that has been used for hundreds of years and is still use in rural parts of China, the Persian panemone and the Cretan Windmill from the Mediterranean.
2.1.2 Modern Wind Pumps

There are 2 distinct end-uses of any wind pump: irrigation or water supply. This is because the two applications require different technical, operational and economic requirements. However, a water supply wind pump can be used for irrigation duties but the reverse is not applicable.

A water supply pump needs to be reliable (run approximately 20 years), requires minimal maintenance (only annual service), pump water over relatively large head (sometimes down to 100m) and run unattended. This means we must use very robust construction materials, drive reciprocating piston pumps and the windmill should be self-starting. They must also be capable of supply from a distant water source.

Irrigation wind pumps, on the other hand, are only for supervised seasonal operation, low head, and large volumes of water whose intrinsic value is low. They are mostly sited right next to water sources. They therefore mostly need to be cheap and can generally be indigenous designs e.g. screw type pumps.

2.2 Principles of Wind Turbines

2.2.1 Power Available in Wind

The general formula for the power available in wind is given by:

\[ P_w = \frac{1}{2} m U^2 \]

Equation 2-1

Where: \( P_w \) = Power available in Watts

\( m \) = mass of flow rate in kg/s

\( U \) = wind velocity

But \( m = \rho a A \), therefore

\[ P_w = \frac{1}{2} \rho a A U^3 \]

Equation 2-2

Where: \( \rho_a \) = density of air in kg/m³

\( A \) = area swept by the rotor in m²

It can be seen that the power available in wind is proportional to the cube of the wind velocity. This means that the power obtainable is greatly influenced by the wind speed available; doubling the wind speed increases the power by a factor of eight. For example, the power obtainable from wind at sea level (\( \rho_a = 1.2 \text{ kg/m}^3 \)) is:
\[ P_w = 0.6U^3 \text{ watts/m}^2 \text{ of rotor area} \]

<table>
<thead>
<tr>
<th>Wind Speed m/s</th>
<th>2.5</th>
<th>5</th>
<th>7.5</th>
<th>10</th>
<th>15</th>
<th>20</th>
<th>30</th>
<th>40</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power Density kW/m²</td>
<td>0.01</td>
<td>0.08</td>
<td>0.27</td>
<td>0.64</td>
<td>2.2</td>
<td>5.1</td>
<td>17</td>
<td>41</td>
</tr>
</tbody>
</table>

**Table 1 Wind speed and power density relationship**

Table 1 shows the variability of power with wind speed. This will have an effect on the design e.g. it is impractical to use wind speed less than 2.5m/s while it will be necessary to provide control systems in case of high speeds as they would damage the wind pump system.

Wind power is also a function of air density, which changes with altitude; i.e. increase in altitude decreases the density and in turn power density. This is shown in Table 2 Relationship between altitude and density correction factor:

<table>
<thead>
<tr>
<th>a.s.l. (m)</th>
<th>0</th>
<th>760</th>
<th>1520</th>
<th>2290</th>
<th>3050</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density Correction Factor</td>
<td>1.00</td>
<td>0.91</td>
<td>0.83</td>
<td>0.76</td>
<td>0.69</td>
</tr>
</tbody>
</table>

**Table 2 Relationship between altitude and density correction factor**

However, the effect of altitude is relatively small in comparison to that of wind velocity. Any increase in altitude can be mitigated by a marginal increase in wind speed e.g. the power density of a 5m/s wind at sea level is about 75 Watts/m²; however, due to the cube law, it only needs a wind speed of 5.64m/s at 3000m above sea level (a.s.l.) to obtain exactly the same power of 75 Watts/m².

### 2.2.2 Operating Principle of a Wind Turbine

Wind turbines transform the kinetic energy available in wind into mechanical rotation energy by use of blades. As air flows across a turbine blade (aerofoil) an aerodynamic force is generated in a rearward direction and at an angle with the direction of relative motion. This aerodynamic force can be resolved into two components:

- **Lift** (L) is the force component perpendicular to the direction of relative motion, while
- **Drag** (D) is the force component parallel to the direction of relative motion.
Turbines can be divided into “lift” or “drag” machines according to whatever force is generated by the wind and exploited as the motive force. In “lift” turbines, the wind flows on both sides of the blade surfaces which have different profiles. This creates a region of low pressure at the upper surface as air is forced to flow faster relative to that at the lower surface. This produces a force called aerodynamic lift which determines the rotation of the turbine about its axis. Simultaneously, a drag force is generated that opposes this motion but in these lift machines is relatively small compared to the lift produced. In drag type machines, the primary motive force is drag produced by the relative motion between the blade and air.

2.3 Types of Wind Turbines
There are two main types of wind turbines, categorized according to their construction:

- Horizontal Axis Wind Turbines (HAWT) – which rotate about a horizontal axis
- Vertical Axis Wind Turbines (VAWT) – which rotate about a vertical axis

2.3.1 Horizontal Axis Turbines
These are the most common type of wind turbine, accounting for above 90% of all installed wind pumps. They can be further sub-divided into:

- Upwind turbines
- Downwind turbines

Upwind turbines are named so because the wind meets the rotor before the tower while downwind turbines have wind meeting the tower before the rotor. Due to these arrangements we achieve higher efficiency in upwind turbines than downwind turbines because they have less aerodynamic interference from the tower which produces turbulence behind it. Downwind turbines also suffer from mast wake from tower turbulence which means they are very susceptible to fatigue failures.
HAWTs are generally used in electricity generation. They have the main rotor shaft and electrical generator at the top of the tower and must be pointed into the wind. They may also include a gear box. Three-blade HAWTs are the most common model but we also have two-blade (Figure 5), single-blade with counterweight (Figure 6) and multi-blade models (such as the local Kijito Wind pump).
HAWTs require a yaw system or tail vane because they are not self-aligning. This is a major drawback and means that citing them requires major consideration of prevailing wind direction. Modern HAWTs have a high efficiency but their capital cost is high.

2.3.2 Vertical Axis Wind Turbines
There are two main types of VAWTs:

- Darrieus turbines
- Savonius turbines

VAWTs allow the mounting of equipment at ground level, unlike HAWTs which eases access and reduces cost. However, they are generally less efficient than traditional HAWTs. Improvements in blade design and the introduction of control mechanisms to allow the rotor speed to vary with wind speed have increased their efficiency.

2.3.2.1 Darrieus VAWT
In 1931 a French engineer called George J. M Darrieus invented a new type of VAWT. Its principle of operation depends on the fact that its blade speed is a multiple of the wind speed, resulting in an apparent wind throughout the whole revolution coming in as a head wind with only a limited variation in angle. They are lift type machines which tend to run very fast.

The Darrieus turbine \[^5\] consists of two or more aerofoil blades attached to the top and bottom of a rotating vertical shaft. These could be straight, giving the machine an "H"-shaped profile, but in
practice most machines have the curved "egg-beater" or troposkien profile. The main reason for this shape is because the centrifugal force caused by rotation would tend to bend straight blades, but the skipping rope or troposkien shape taken up by the curved blades can resist the bending forces effectively. The wind blowing over the aerofoil contours of the blade creates an aerodynamic lift and actually pulls the blades along.

![Figure 7 the Darrieus VAWT](image)

A symmetrical aerofoil isn’t the most efficient approach in providing lift. A design goal was to maximize lift at the same time as utilizing the inevitable drag created by the opposing blade. The wind flowing over the high lift and low drag blade profile, coupled with the blade areas’ solidity gives the machine the initial power to overcome its inertia. Once momentum has been established the forward movement of the blade through the air creates its own local wind flow over its contours creating a varying amount of lift throughout the full 360° of its rotation. In its downwind stroke, the curved underside of the blade acts like the sail of a yacht therefore creating usable torque throughout its rotation.

The stalling of the blades at the top and bottom of their arc rotation is also useful as it regulates the speed of the blades rotation, implying that the blades accelerate up to a point of equilibrium at which they won’t increase their speed no matter how hard the wind blows.

In comparison with the “drag-type” Savonius turbines, Darrieus type turbines offer higher efficiency since they reduce the losses due to friction. However, they are incapable of self-starting
since independently of wind they have a very low starting torque. They therefore require an auxiliary device to start, such as a motor or use in combination with a Savonius turbine. For the combined type Darrieus-Savonius, the starting torque is represented by the Savonius turbine coaxial and internal to the Darrieus turbine.

A subtype of Darrieus turbine with straight, as opposed to curved, blades is known as the Giromill or Cycloturbine. The Cycloturbine variety has variable pitch to reduce the torque pulsation and is self-starting. The advantages of variable pitch are: high starting torque; a wide, relatively flat torque curve; a lower blade speed ratio; a higher coefficient of performance; more efficient operation in turbulent winds; and a lower blade speed ratio which lowers blade bending stresses. Straight, V, or curved blades may be used.

The main characteristics of the Darrieus-type turbine are:

- “fast” turbine
- reduced efficiency in comparison with horizontal axis turbines, also because a great part of the blade surface rotates very close to the axis at a low speed
- adaptability to variations in the direction of the wind
- effective for winds with an important vertical component of speed (sites on slopes or installation on the roof of the buildings “corner effect”)
- suitable for low values of wind speed and for a limited range
- necessity of an adequate speed control to keep the efficiency within acceptable values
- impossibility of reducing the aerodynamic surface in case of speed exceeding the rated one because of the fixed blades
- necessity of a mechanical break for stopping the turbine
- necessity of a structure not extremely robust to withstand extreme winds (given the smaller surface of the blades exposed to the wind in comparison with Savonius turbines)
- suitable for large power applications
- low noise and with vibrations limited to the foundations, therefore suitable to be installed on buildings
- able to operate also under turbulent wind conditions gearbox and electric generator may be positioned at ground level
- high fluctuations of the motive mechanical torque.

### 2.3.2.2 The Savonius Rotor

The Savonius turbine is a vertical axis machine which uses a rotor that was introduced by a Finnish Engineer, S. J. Savonius in 1922 [5]. In its simplest form it is two cups or half drums fixed onto a central shaft in opposing directions. Each cup/drum catches the wind and so turns the shaft, bringing the opposing cup/drum into the flow of the wind. The cup/drum then repeats the process causing the shaft to rotate further and completing a full rotation. This process continues all the time the wind blows and the turning of the shaft is used to drive a pump or small generator.

The Savonius rotor is simple in structure and has good starting characteristics, relatively low operating speeds and an ability to accept wind from any direction. It has a lower aerodynamic efficiency compared to the Darrieus and H-rotor.

Savonius claimed a maximum coefficient of power of 0.31. However field trials done by him on the Savonius rotors gave a maximum coefficient of 0.37. However the same hasn’t been achieved by subsequent researchers. This means that the main driving force is drag force of wind acting on its blade. However at low angles of attack, lift force also contributes to torque production. Hence Savonius rotor is not a purely drag machine but a compound machine.

Although conventional Savonius rotors have low aerodynamic efficiency, they have a high starting torque or high coefficient of static torque. Due to this they can be used as starters for other types of...
wind turbines that have lower starting torques and for applications that require high starting torque such as water pumping.

When turning, the Savonius rotor presents the wind with a concave and convex section, and derives most of its power from drag. The drag coefficient for a flow perpendicular to the convex face of half a drum/pipe is 1.2 while the drag coefficient for the concave section is nearly twice as high at 2.3. Therefore the force on the concave side of the rotor is higher inducing a torque that turns the rotor. In addition to the torque due to the drag flow through the gap between the two rotor blades causes lift with thrust out the back face of the rotor helping it turn in the desired direction. Drag based rotors generally have a higher starting torque but lower efficiency when compared to modern lift based rotors.

The main characteristics of Savonius turbine are:

- "slow" turbine
- low efficiency value
- suitability for low values of wind speed and within a limited range
- necessity of adequate speed control to keep the efficiency within acceptable values
- impossibility of reducing the aerodynamic surface in case of speed exceeding the rated one because of the fixed blades
- necessity of a mechanical break for stopping the turbine
- necessity of a robust structure to withstand extreme winds (the high exposed surface of the blades)
- suitable for small power applications only
- high starting torque
- low noise.

These characteristics make the Savonius turbine particularly suited for applications such as a source of power for water supply & irrigation in rural areas. The Darrieus turbine on the other hand, whilst having high efficiency and tip speed ratios exceeding unity, needs to be coupled to a generator that converts its high speed into electric power which in turn runs a pump. This increases the cost of the installation as well as its complexity. The high starting torque also makes the Savonius turbine suited to matching to applications with reciprocating pumps which require three times as much torque to start them as compared to that used to run them.

2.4 Efficiency, Power and Torque Characteristics

The maximum efficiency coincides with the maximum power output in a given wind speed. Efficiency is usually presented as a non-dimensional ratio of shaft-power divided by wind-power passing through a disc or shape having the same area as the vertical profile of the windmill rotor; this ratio is known as the "Power Coefficient" or $C_p$: 
Where: $P_t =$ power in turbine shaft

$P_w =$ wind power due to its kinetic energy

However, $C_p$ has a maximum value of 0.593, the **Betz Limit**, because it is impossible to stop the wind completely and obtain all its kinetic energy. This limit was discovered by Albert Betz. Between 1922 and 1925, Betz published writings in which he was able to show that, by applying elementary physical laws, the mechanical energy extractable from an air stream passing through a given cross-sectional area is restricted to a certain fixed proportion of the energy or power contained in the air stream. Moreover, he found that optimal power extraction could only be realized at a certain ratio between the flow velocity of air in front of the energy converter and the flow velocity behind the converter. Although Betz's "momentum theory", which assumes an energy converter working without losses in a frictionless airflow, contains simplifications, its results are still used for performing first calculations in practical engineering. But its true significance is founded in the fact that it provides a common physical basis for the understanding and operation of wind energy converters of various designs.

The essential findings derived from the momentum theory can be summarized in words as follows:

- The mechanical power which can be extracted from free-stream airflow by an energy converter increases with the third power of the wind velocity.
- The power increases linearly with the cross-sectional area of the converter traversed; it thus increases with the square of its diameter.
- Even with an ideal airflow and lossless conversion, the ratio of extractable mechanical work to the power contained in the wind is limited to a maximum value of 0.593. Hence, only less than 60% of the wind energy of a certain cross-section can be converted into mechanical power.
- When the ideal power coefficient achieves its maximum value ($C_p = 0.593$), the wind velocity in the plane of flow of the converter amounts to two thirds of the undisturbed wind velocity and is reduced to one third behind the converter.

Speed is conventionally given as a non-dimensional ratio known as the tip-speed ratio; the ratio of the speed of the windmill rotor tip, at radius $R$ when rotating at $\omega$ radians/second, to the speed of the wind, $U$.

$$\lambda = \frac{\omega R}{U} = \frac{\pi ND}{60U}$$

Equation 2-4

Where: $R =$ rotor radius $= \frac{D}{2}$ in m
The torque coefficient, which is a non-dimensional measure of the torque produced by a given size of rotor in a given wind speed (torque is the twisting force on the drive shaft). The torque coefficient, $C_t$, is defined as:

$$C_t = \frac{T}{\frac{1}{2} \rho_a A U^2}$$

Equation 2-5

Where $T = \text{Torque}$

### 2.5 Pumps and Water Lifting Techniques

Water may be moved by the application of any one (or any combination) of five different mechanical principles, which are largely independent, i.e. by:

1) **direct lift** - this involves physically lifting water in a container
2) **displacement** - this involves utilizing the fact that water is (effectively) incompressible and can therefore be "pushed" or displaced
3) creating a **velocity head** - when water is propelled to a high speed, the momentum can be used either to create a flow or to create a pressure
4) using the **buoyancy of a gas** - air (or other gas) bubbled through water will lift a proportion of the water
5) **gravity** - water flows downward under the influence of gravity

Pumps and lifting/propelling devices may be classified according to which of the above principles they depend on. These major categories can then be further sub-divided into two:

- **Reciprocating/Cyclic** - relates to devices that are cycled through a water-lifting operation (for example a bucket on a rope is lowered into the water, dipped to make it fill, lifted, emptied and then the cycle is repeated); in such cases the water output is usually intermittent, or at best pulsating rather than continuous
- **Rotary** - are generally developed to allow a greater throughput of water, and they also are easier to couple to engines or other types of mechanical drive. Therefore, by definition, a rotary pump will generally operate without any reversal or cessation of flow, although in some cases the output may appear in spurts or pulsations.

The fluid quantities involved in all hydraulic machines are the flow rate ($Q$) and the head ($H$), whereas the mechanical quantities associated with the machine itself are the power ($P$), speed ($N$), size ($D$) and efficiency ($\eta$). For a pump running at a given speed, its output is the flow rate it delivers and the head developed. Therefore, fundamental pump characteristics are expressed in graphs of head against flow rate at constant speed. To achieve this performance, a power input is required which involves efficiency of energy transfer. It is thus useful to plot the power and
efficiency against flow rate wherever possible to show a complete set of performance characteristic of pumps as shown in Figure 10

![Figure 10 Typical curves showing relationship between head, flow, speed and efficiency of pumps][6]

The power ($P_{hyd}$ in Watts) required to lift any water is a function of both the apparent vertical height lifted and the flow rate at which water is lifted:

$$P_{hyd} = \rho_w g H_s Q$$

Where: $\rho_w = \text{density of water (1000 kg/m}^3\text{)}$

$g = \text{gravitational acceleration (9.81 m/s}^2\text{)}$

$Q = \text{flow rate (m}^3\text{/s)}$

$H_s = \text{vertical height (m)}$
2.5.1 Head

**Head**[^6] is the resistance of the system to fluid flow. The actual pumping head imposed on a pump, or "*gross working head*", will be somewhat greater than the actual vertical distance, or "*static head*", water has to be raised from the source to its destination. Figure 11 indicates a typical pump installation, and it can be seen that the gross pumping head, (which determines the actual power need), consists of the sum of the friction head, the velocity head and the actual static head (or lift) on both the suction side of the pump (in the case of a pump that sucks water) and on the delivery side. The gross head \( H_g \) can also be simplified to the sum of the **delivery head** \( H_d \) and **suction head** \( H_s \).

\[
H_g = H_s + H_d
\]

The **friction head** consists of a resistance to flow caused by viscosity of the water, turbulence in the pump or pipes, etc. It can be a considerable source of inefficiency in badly implemented water distribution systems, as it is a function which is highly sensitive to flow rate, and particularly to pipe diameter, etc.

The **velocity head** is the apparent resistance to flow caused by accelerating the water from rest to a given velocity through the system; any object or material with mass resists any attempt to change its state of motion so that a force is needed to accelerate it from rest to its travelling velocity. This force is "felt" by the pump or lifting device as extra resistance or head. The higher the velocity at which water is propelled through the system, the greater the acceleration required and the greater the velocity head. The velocity head is proportional to the square of the velocity of the water. Therefore, if the water is pumped out of the system as a jet, with high velocity, then the velocity head can represent a sizeable proportion of the power need and hence of the running costs. But in most cases where water emerges from a pipe at low velocity, the velocity head is relatively small.

[^6]: Head is the resistance of the system to fluid flow. The actual pumping head imposed on a pump, or "*gross working head*", will be somewhat greater than the actual vertical distance, or "*static head*", water has to be raised from the source to its destination. Figure 11 indicates a typical pump installation, and it can be seen that the gross pumping head, (which determines the actual power need), consists of the sum of the friction head, the velocity head and the actual static head (or lift) on both the suction side of the pump (in the case of a pump that sucks water) and on the delivery side. The gross head \( H_g \) can also be simplified to the sum of the **delivery head** \( H_d \) and **suction head** \( H_s \).

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2.5.2 Suction Lift
Some pipes are capable of sucking water from a source; i.e. the pump can be located above the water level and will literally pull water up by creating a vacuum in the suction pipe. Drawing water by suction depends on the difference between the atmospheric pressure on the free surface of the water and the reduced pressure in the suction pipe developed by the pump. The greater the difference in pressure, the higher the water will rise in the pipe. However, the maximum pressure difference that can be created is between sea level atmospheric pressure on the free surface and a pure vacuum, which theoretically will cause a difference of level of water of 10.4m. However, before a drop in pressure even approaching a pure vacuum can be produced, the water will start gassing due to release of air held in solution (just like soda water gasses when released from a pressurized container); if the pressure is reduced further, the water can boil at ambient temperature. As soon as this happens, the pump loses its prime and the discharge will cease (due to loss of prime) or at least be severely reduced. In addition, boiling and gassing within the pump (known as cavitation) can cause damage if allowed to continue for any length of time.

The suction lifts that can be achieved in practice are therefore much less than 10.4m. For example, centrifugal pumps, which are prone to cavitation due to the high speed of the water through the impeller, are generally limited to a suction lift of around 4.5m even at sea level with a short suction pipe. Reciprocating pumps generally impose lower velocities on the water and can therefore pull a higher suction lift, but again, for practical applications, this should never normally exceed about 6.5m even under cool sea level conditions with a short suction pipe.

At higher altitudes, or if the water is warmer than normal, the suction lift will be reduced further. For example, at an altitude of 3 000m (10 000ft) above sea level, due to reduced atmospheric pressure, the practical suction lift will be reduced by about 3m compared with sea level, (and proportionately for intermediate altitudes, so that 1 500m above sea level will reduce suction lift by about 1.5m). Higher water temperatures also cause a reduction in practical suction head; for example, if the water is at say 30°C, the reduction in suction head compared with water at a more normal 20°C will be about 7%.

Extending the length of the suction pipe also reduces the suction head that is permissible, because pipe friction adds to the suction required; this effect depends on the pipe diameter, but typically a suction pipe of say 80m length will only function satisfactorily on half the above suction head.

2.5.3 Practical Power Requirements
Calculating the power requirement for water lifting is fundamental to determining the type and size of equipment that should be used, so it is worth detailing the principles for calculating it. In general the maximum power required will simply be:

\[ P_{hyd} = \frac{\rho_w g H_d Q}{\eta} \]

Equation 2-7
Where: $\eta =$ total system efficiency at max flow rate

The daily energy requirement will similarly be:

\[
\frac{\text{volume delivered per day} \times \rho g H_g}{\eta}
\]

### 2.5.4 Rotary Positive Displacement Pumps

This is a group of devices which utilizes the displacement principle for lifting or moving water by using a rotating form of displacer. These generally produce a continuous, or sometimes a slightly pulsed, water output. The main advantage of rotary devices is that they lend themselves readily to mechanization and to high speed operation. The faster a device can be operated the larger the output in relation to its size and the better its productivity and cost-effectiveness. Also, steady drive conditions tend to avoid some of the problems of water hammer and cavitation that can affect reciprocating devices.

All rotary pumps have the following features:

1. They typically have more than one rotating chamber, so that inertia effects are minimized and the fluctuations in flow rate are made negligible.
2. The chambers rotate so that, in turn, they come in direct contact with the inlet and outlet ports, making valves unnecessary.
3. The rotational speed is usually quite high, so they can be directly coupled to high-speed prime movers.

They are suitable for applications calling for large heads and small volumes, or specific speeds less than 0.2 [7]

They can be further split into 2 types:

a) Fixed capacity, in which the swept volume per revolution is fixed.

b) Variable capacity where the swept volume per revolution may be varied within limits determined by the type of pump.

### 2.5.5 Reciprocating Displacement Pumps

Water is for most practical purposes incompressible. Consequently, if a close fitting piston is drawn through a pipe full of water (Figure 12a), it will displace water along the pipe. Similarly, raising a piston in a submerged pipe will draw water up behind it to fill the vacuum which would otherwise occur (Figure 12b); this applies of course only up to a certain limit of the height water can be pulled by a vacuum, known as the suction lift (atmospheric limit). In the first case water is displaced by the piston, but in the second case, the piston serves to create a vacuum and the water is actually displaced by atmospheric pressure pressing on its external surface, as indicated in the figure. So water car. be displaced either by "pushing" or by "pulling", but it can also be "displaced" by a solid
object being pushed into water so that the level around it rises when there is nowhere else for the water to go, as indicated (Figure 12c).

![Figure 12 Reciprocating Displacement Pump Principles](image)

### 2.5.6 Basic Principles of Reciprocating Displacement Pumps

There are various basic relationships between the output or discharge rate \( Q \), piston diameter \( D_p \), stroke or length of piston travel \( s \), number of strokes per minute \( n \), and the volumetric efficiency, which is the percentage of the swept volume that is actually pumped per stroke \( \eta_{vol} \):

The swept area \( A_s \), in \( m^2 \), of the piston is:

\[
A_s = \frac{\pi D_p^2}{4}
\]

The swept volume per stroke \( V \), in \( m^3 \), will be:

\[
V = A_s
\]

The discharge per stroke will be \( q \) is:

\[
q = \eta_{vol} V
\]

The pumping rate \( Q \), \( m^3 \) per min, is:

\[
Q = nq
\]

\[
Q = n \times \eta_{vol} \times s \times \frac{\pi D_p^2}{4}
\]
Thus the pump discharge is entirely independent of the pressure against which the pump is delivering.

The force \( F \) required to lift the piston, will be the weight of the piston and pump rods \( (W_p) \), plus the weight of the column of water having a cross section equal to the piston area and a height equal to the head \( (H) \). There is also a dynamic load which is the force needed to accelerate these masses. If the acceleration is small, we can ignore the dynamic forces, but in many cases the dynamic forces can be large. In principle, the dynamic force, to be added to the static force, will be the summed product of the mass and acceleration of all moving components (i.e. water, plus piston, plus pump rod). However, if we ignore the practically negligible effects of upward buoyancy force due to displacing water and dynamic forces, we arrive at the following formula:

\[
F = W_p + A_s \rho_w g H
\]

Equation 2-8

The conventional way of mechanizing a reciprocating piston pump is by connecting the pump rod to a flywheel via a crank as shown in Figure 13:

![Crank Operated Piston Pump](image-url)
The torque (or rotational couple) needed to make the crank or flywheel turn will vary depending on the position of the crank. When the piston is at the bottom of its travel (bottom dead centre or b.d.c), marked as "a" on the figure, the torque will be zero as the pump rod pull is acting at right angles to the direction of movement of the crank and simply hangs on the crank; as it rotates to the horizontal position marked "b", the torque will increase sinusoidally to a maximum value of Fs/2 (force F, times the leverage, which is s/2); the resisting force will then decrease sinusoidally to zero at top dead centre (t.d.c.) marked "e". Beyond t.d.c. the weight of the pump rod and piston will actually help to pull the crank around and while the piston is moving down the water imposes no significant force on it other than friction. If, for convenience, we assume the weight of the piston and pump rod is more or less cancelled out by friction and dynamic effects, the torque is effectively zero for the half cycle from t.d.c. at "e" through "d" to "a" at b.d.c. where the cycle restarts. The small graph alongside illustrates the variation of torque with crank, position through two complete revolutions.

If the crank has a flywheel attached to it, as it normally will, then the momentum of the flywheel will smooth out these cyclic fluctuations by slowing down very slightly (too little to be noticeable) during the "a-b-c" part of the cycle and speeding up during the "c-d-a" part, as illustrated by the broken line following the first revolution in the graph in Figure 13. If the flywheel is large, then it will smooth the fluctuations in cyclic torque to an almost steady level approximating to the mean value of the notched curve in the figure. The mean value of half a sine wave, to which this curve approximates, is the peak value divided by π:

The peak torque, $T_p$, felt by the crank drive:

$$T_p = \frac{Fs}{2}$$

Equation 2-9

Mean torque, $T_m$, with the flywheel:

$$T_m = \frac{Fs}{2\pi}$$

Equation 2-10

Therefore, the torque necessary to turn a crank through its first revolution will be about 3 times greater than the mean torque which is needed to maintain steady running. Power can be calculated as the product of speed and torque:

$$P = T_m \omega = \frac{Fs}{2\pi} x \frac{2\pi N}{60}$$

$$P = \frac{FsN}{60}$$

Equation 2-11
2.6 Power Transmission

Power transmission from the rotor to the pump can be represented as shown above. This was taken from an installation done in Honduras [8] of a Savonius wind powered water pumping system. The reducer is threaded on a rotor shaft and bolted through the flanges on the Savonius rotor. The reducer is supported in the oil impregnated, flanged bronze brushing which bears the support of the rotor through the support pipe welded to the tower structure. At the bottom of the rotor shaft is an automotive rear differential from a 1982 Mazda. The torque of a Savonius rotor enters the differential at the drive shaft input and with a 2.5:1 increase in torque turns the shaft attached to the right wheel hub. The pump wheel is bolted to the pump shaft and supported on the other end in an oil impregnated bronze bushing. The brake drum is on the left. When the pump is supposed to operate under wind power, the brake is locked with a handbrake lever not shown. To disconnect the
pump from the rotor the handbrake is released and that side of the differential becomes the lower resistance path. Since the differential is non-locking, the pump as the higher resistance path doesn’t turn in the same way as a car wheel on solid ground stays still when the wheel on the other side slips in the mud. With the brake locked the pump wheel turn pulling the rope with the threaded washers through a pipe in the well not shown and up to the outlet. The water then flows in to the storage tank where the pressure head due to gravity is used to distribute water to the fields.

2.7 Dimensional Analysis
Dimensional analysis is a method for reducing complex physical problems to the simplest (that is, most economical) form prior to obtaining a quantitative answer. It offers a route by which one can determine the form of the dependence of one variable upon a range of other controlling parameters in the absence of an analytical solution by allowing the identification of groups of variables whose interrelationships may be determined experimentally.

Bridgman\[9\] explains it thus: "The principal use of dimensional analysis is to deduce from a study of the dimensions of the variables in any physical system certain limitations on the form of any possible relationship between those variables. The method is of great generality and mathematical simplicity". Therefore, it provides a qualitative route to the understanding of fluid flow mechanisms; the quantitative understanding is provided experimentally.

Any physical situation involving an object or system may be described in terms of its fundamental properties, e.g. mass, length, velocity, acceleration, density or the forces involved. These fundamental properties of a system are universal and known as its dimensions. For Fluid Mechanics, the four basic dimensions are usually taken to be mass M, length L, time T, and temperature θ, or an MLTθ system.

While dimensions are universal, units are chosen by convenience. Contemporary units; such as mass in kg or velocity in m/s; provide a standardized measure of the dimension under consideration so that conversion between different units of the same dimension is possible.

Dimensional analysis relies on the concept of similarity, which refers to some equivalence between two phenomena that are actually different. Using similarity, one can clarify the functional form of physical relationships and analyzing its dimensions by group. This involves a generalized methodology that leads directly to a set of dimensionless groupings whose number could be determined in advance by a scrutiny of the matrix formed from the variables (and their dimensions) considered relevant and necessary to describe the physical relationships. Such a technique is the Buckingham π method.

The initial step in the Buckingham π method is to list the variables considered significant and form a matrix with their dimensions. The number of dimensionless groups arising from a particular matrix formed from n variables in m dimensions is n – r, where r is the largest non-zero determinant that can be formed from the matrix, and therefore the equation relating the variables will be of the form:
In the treatment of fluid conditions that only involve the dimensions MLT, it is simpler to state that the number of dimensionless groups arising from a particular matrix formed from $n$ variables in $m$ dimensions is $n - m$.

Whenever a design engineer needs to take decisions at the design stage of a project, he will probably need to initiate some form of model test programme. The basis of such a test series depends on the accurate use of instrumentation systems and the correct application of the theories of similarity. This in turn involves application of dimensional analysis and utilization of dimensionless groups such as the Reynolds, Froude or Mach numbers.

Model testing occurs in all areas of engineering based on fluid mechanics. Wind tunnel tests are carried out using models of high levels of intricacy to simulate tidal flow. Model tests depend on two basic types of similarity which may be considered separately: geometric and dynamic similarity.

2.7.1 Geometric Similarity
The first requirement for model testing is a strict adherence to the principle of geometric similarity. This means the model must be an exact geometric replica of the prototype. Therefore a model and prototype are geometrically similar if all body dimensions in all three coordinates have the same linear scale ratio. Scale factor refers to the characteristic property that the ratio of any length in one system to the corresponding length in the other system is everywhere the same.

2.7.2 Kinematic Similarity
This is similarity of motion and implies similarity of lengths and in addition similarity of time intervals, i.e. as the corresponding lengths in the two systems is in a fixed ratio, so are those of the velocities of corresponding particles.

2.7.3 Dynamic Similarity
This is the similarity of forces and can be defined as stated: Forces acting on corresponding masses in the model and prototype shall be in the same ratio throughout the area of flow modelled. If this similarity is achieved then it follows that the flow pattern will be identical for the model and the prototype flow fields. For dynamic similarity to be realised, the model and prototype must be kinematically and geometrically similar. However, kinematically similar systems aren’t necessarily dynamically similar. This means dynamic similarity exists when the model and prototype have the same length-scale ratio, time-scale ratio and force scale ratio.

2.7.4 Incomplete Similarity
Many instances involve more than one force ratio in a flow system. For such systems complete dynamic similarity is possible only when a full sized model is built. Since this is impractical, incomplete similarity results where significant forces are compared and effects of other forces neglected.
Fluid flow encounters the following forces:

- Inertia Forces = \( Ma = (\rho l^3)(v^2/l) = \rho l^2 v^2 \)
- Viscous forces = \( \mu (du/dy)A = \mu (v/l)^2 = \mu v l \)
- Gravity force = \( Mg = \rho l^3 g \)
- Pressure force = \( \Delta P A = \Delta P l^2 \)
- Surface tension force = \( \tau l \)
- Reynolds number (Re) = \( \rho l^2 u^2/\mu vl = \rho ul/\mu \)

For a windmill, the Power (\( P \)) depends on the following factors: the fluid density (\( \rho \)), the fluid velocity (\( U \)), the fluid dynamic viscosity (\( \mu \)), the rotor diameter (\( D \)) and the rotor height (\( H \)). This may be expressed as the following functional expression:

\[
P = f(\rho, U, \mu, D, H)
\]

Equation 2.12

With the following dimensions:

<table>
<thead>
<tr>
<th>Quantity Symbol</th>
<th>Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>( P )</td>
<td>ML(^2)T(^{-3})</td>
</tr>
<tr>
<td>( \rho )</td>
<td>ML(^{-3})</td>
</tr>
<tr>
<td>( U )</td>
<td>LT(^{-1})</td>
</tr>
<tr>
<td>( M )</td>
<td>ML(^{-3})T(^{-1})</td>
</tr>
<tr>
<td>( D )</td>
<td>L</td>
</tr>
<tr>
<td>( H )</td>
<td>L</td>
</tr>
</tbody>
</table>

Table 3 Quantities and their Dimensions

There are 6 variables (\( n \)) and 3 common dimensions (\( m \)) therefore the number of \( \pi \) groups is:

\[
n - m = 6 - 3 = 3
\]

Equation 2.13

We select \( \rho \), \( D \) and \( U \) as our repeating variables. Thereafter we can express the functional expression as:

\[
\pi_0 = f(\pi_1, \pi_2)
\]

Equation 2.14

2.7.4.1 \( \pi_0 \)

\[
M^0L^0T^0 = P^{a}U^{b}D^{c} = [ML^2T^{-3}] [ML^{-3}]^{a}[LT^{-1}]^{b}[L]^{c}
\]

For \( M \):

\[
0 = 1 + a
\]

\[
a = -1
\]
For T:

\[ 0 = -3 + b \]
\[ b = -3 \]

For L:

\[ 0 = 2 - 3a + b + c = 2 + 3 - 3 + c \]
\[ c = -2 \]

Therefore,

\[ \pi_0 = \frac{P}{\rho U^3 D^2} \]

Equation 2.15

\[ 2.7.4.2 \quad \pi_1 \]

\[ M^0 L^0 T^0 = H \rho^a U^b D^c = [L][ML^{-3}][LT^{-1}]^b[L]^c \]

For M:

\[ 0 = a \]

For T:

\[ 0 = -b \]
\[ b = 0 \]

For L:

\[ 0 = 1 - 3a + b + c = 1 + c \]
\[ c = -1 \]

Therefore,

\[ \pi_1 = \frac{H}{D} \]

Equation 2.16

\[ 2.7.4.3 \quad \pi_2 \]

\[ M^0 L^0 T^0 = \mu^a U^b D^c = [ML^{-1}T^{-2}][ML^{-3}][LT^{-1}]^b[L]^c \]

For M:

\[ 0 = 1 + a \]
For T:

\[ a = -1 \]

\[ 0 = -1 + b \]

\[ b = -1 \]

For L:

\[ 0 = -1 - 3a + b + c = -1 + 3 - 1 + c \]

\[ c = -1 \]

Therefore,

\[ \pi_2 = \frac{\mu}{\rho UD} \]

Then Equation 2-14

\[ \pi_0 = f(\pi_1, \pi_2) \]

Can be rewritten as:

\[ \frac{P}{\rho U^3 D^2} = f\left(\frac{H}{D}, \frac{\mu}{\rho UD}\right) \]

\[ \frac{P}{\rho U^3 D^2} = f\left(\frac{H}{D}, \rho UD \right) \frac{1}{\mu} \]

\[ \frac{P}{\rho U^3 D^2} = f\left(\frac{H}{D}, Re\right) \]

Equation 2-18

This means that under particular conditions where the parameters of the two events are such that \( \pi_1 \) and \( \pi_2 \) have the same values, that is, where:

\[ \frac{H_1}{D_1} = \frac{H_2}{D_2} \]

\[ \frac{\rho_1 U_1 D_1}{\mu_1} = \frac{\rho_2 U_2 D_2}{\mu_2} \]

Then the following similarity will hold true:

\[ \frac{P_1}{\rho_1 U_1^3 D_1^2} = \frac{P_2}{\rho_2 U_2^3 D_2^2} \]
Chapter 3

3 Laboratory Test

3.1 Objective
To determine the characteristics of the Savonius Rotor under varying wind speeds in the wind tunnel.

3.2 Introduction

From the diagram we define the rotor geometrical parameters:

\[ e = \text{primary overlap} \]
\[ a = \text{secondary overlap} \]
\[ R = \text{rotor radius} \]
\[ d = \text{diameter of blade} \]

The goal of the experiment is to obtain the relationships between the motive fluid and the Savonius rotor while also noting the effect of having a primary overlap and not having a primary overlap (or air gap henceforth).
The wind tunnel employed has a square cross-section and is powered by large motor that drives fans which propel the wind. It is a variable speed wind tunnel with a minimum wind speed of about 3.5m/s.

The Savonius model employed was obtained from a kit used to demonstrate wind turbine principles. Although it lacked in design complexity (we could not alter the overlap ratio e/D, the blade shape factor e.t.c.) it was allowed for some simple tests to prove certain important characteristics of the Savonius rotor e.g. high starting torque, maximum efficiency (otherwise known as the co-efficiency of power) and allow us to compare them with established results of VAWT types.

3.3 Procedure
The Savonius model used has an air gap that can be sealed. Initially the model was set up on the base plate with its air gap open and placed in the wind tunnel section. The ammeter and the voltmeter were then connected in series and parallel, respectively, in the circuit with the Savonius model as the power source.

The wind tunnel was switched on to provide our motive fluid. Various wind speeds were applied on the model, read off the anemometer, with corresponding instantaneous values for voltage, current and rotor speed in revolutions per minute (measured by the tachometer) for each value of wind speed. These values were recorded in a table.

The procedure was repeated for the Savonius model with its air gap sealed. Thereafter analysis was drawn from the tabulated results.
3.4 Apparatus

Figure 16 Test Apparatus: Savonius model, Tachometer, Anemometer, Voltmeter and Ammeter

1. A wind tunnel
2. Tachometer
3. Digital anemometer
4. Voltmeter and ammeter
5. Savonius model
6. Base plate
7. Connecting wires
Figure 17 Wind Tunnel
3.5 Results, Tables and Graphs

Theoretical Power, $P_{th}$

$$P_{th} = \frac{1}{2} \rho_a (2RH) U^3 = \frac{1}{2} \rho_a DH U^3$$

Equation 3-1

Where: $\rho_a = 1.20 \text{ kg/m}^3$

The experimental power, $P_{exp}$:

$$P_{exp}(\text{Watts}) = V(\text{Volts}) \times I(\text{Amps})$$

Equation 3-2

Therefore the co-efficient of Power, $C_p$:

$$C_p = \frac{P_{exp}}{P_{th}}$$

Equation 3-3

And the velocity co-efficient (tip speed ratio), $\lambda$ according to Equation 2-4

$$\lambda = \frac{\omega R}{U} = \frac{\pi N D}{60 U} = \frac{\pi N R}{30 U}$$

For the theoretical Torque; $T_{th}$

$$T_{th} = \frac{1}{2} \rho R^2 H U^2$$

Equation 3-4

The experimental Torque, $T_{exp}$:

$$T_{exp} = \frac{P_{exp}}{\omega} = \frac{60P_{exp}}{2\pi N}$$

Equation 3-5

Thus, the co-efficient of torque, $C_t$

$$C_t = C_p \times \lambda$$

$$C_t = \frac{C_p}{\lambda}$$

Equation 3-6
For the model employed:

\( H = 7.8\text{cm} \)

\( D = 2R = 8.84\text{ cm} \)

\( a = 0\text{ cm} \)

\( e = 3.6\text{ cm} \)

\( d = 6.16\text{ cm} \)
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Table 4 Calculated values of Savonius with air gap
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<th>Power Coefficient, ( C_p )</th>
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Table 5 Calculated values of Savonius without air gap
From the data obtained, we can draw the following characteristic graphs:

**Figure 18** Graph of Power vs. wind speed of the Savonius rotor
Comparison Graphs of $C_p$ Values For a Savonius Rotor With and Without an Air Gap

Figure 19 Comparison graph of $C_p$ values for a Savonius rotor with & without an air gap
Aerodynamic Co-efficients ($C_p$ & $C_t$) vs. Tip Speed Ratio for A Savonius Rotor With an Air Gap

Figure 20 Aerodynamic co-efficients vs. tip speed ratio with an air gap
Figure 21 Aerodynamic co-efficients vs. tip speed ratio without an air gap
Chapter 4

4 Analysis
It was observed that with or without the air gap sealed, the Savonius model on occasion would not rotate even in sufficient speeds for self-start ability. This was due to the orientation of the semi-circular blades in relation to the wind. If they were parallel to the wind direction, they would not rotate. This is because the fluid would flow right past them without imparting any kinetic energy.

Once the blades were orientated perpendicular to the flow, the rotor operated. From this we can infer that to increase the self-start ability of the Savonius Rotor, we should add a second pair of blades at 90° angles to the other pair. Work done by Menet, J.L \cite{menet10,menet11} agrees with our findings. He recommended that to optimize the Savonius rotor, it should be double stepped (each step staggered from the other by a 90° angle). This ensures sufficient stability of mechanical torque.

Figure 22 Savonius blades parallel to the flow direction

Figure 23 Savonius blades normal to the flow direction
It was also observed that above wind speeds of 10-11 m/s the model experienced significant vibrations and instability. At speeds below 4 m/s, the rotor would not start, and even when given a slight push, nor be able to maintain its rotation. This could be due to significant frictional forces present in the miniature generator system used to produce electrical power.

From the graphs of power vs. wind speed it is observed that the Savonius rotor with an air gap had a higher output of power at the same wind speed than that without the air gap. This is because sealing the air gap interferes with the fluid flow around the rotor as fluid towards the centre of the rotor cannot pass through the seal. This fluid therefore has to flow back out of the rotor, interfering with that fluid now entering the rotor. This could also cause turbulent flow of the fluid which in turn reduces the turbines efficiency, especially at higher speeds.

From the graph drawn of the two co-efficiency of power curve vs. tip speed ratio on a single axis we can make the following observations:

1. The $C_p$ curve of the rotor with an air gap is much broader than when the air gap is sealed. This means that the rotor can be used in higher speeds and achieve higher sustainable efficiencies. However, at low speeds, the rotor with its air gap sealed had better $C_p$ characteristics. This is because it was able to maximize on the little kinetic power available from the wind. It would therefore be beneficial to reduce the air gap to some optimal amount to obtain characteristics of both curves.

2. The Savonius with the air gap is able to function at tip speed ratios slightly above 1. This proves that it is not a purely drag-type rotor, whose main characteristic is to have a maximum tip-speed ratio of 1 because the tip of the rotor blade cannot move faster than the wind. It therefore has a lift component that enables it to achieve this effect. Without the air gap, this is not possible as the blade is not “submerged” in the fluid i.e. it is not completely surrounded by the motive fluid. It is thus a purely drag-type machine.

The maximum value of $C_p$ obtained is around 0.14. This is almost similar to those found in the general body of available literature which claims the efficiency of a Savonius rotor to be anywhere between 10 – 37% \cite{12}\cite{13} and more commonly to be around 0.15 which means our results are acceptable.

From the curves of the co-efficiency of torque, we observe that the torque rises sharply before reaching a maximum. Thereafter, it drops. This shows that the Savonius rotor does indeed have high starting torque.

### 4.1 Prototype Design

We employ the previously found dimensionless relationship Equation 2-18 to determine the design parameters of a Savonius Windmill to be used in the pumping of borehole water:

$$\frac{P}{\rho U^3 D^2} = f\left(\frac{H}{D}, Re\right)$$
Using subscripts \( m \) & \( p \) to denote our model and prototype respectively:

\[
\frac{H_m}{D_m} = \frac{H_p}{D_p}
\]

\[
Re_m = Re_p \equiv \frac{\rho_m U_m D_m}{\mu_m} = \frac{\rho_p U_p D_p}{\mu_p}
\]

\[
\frac{P_m}{\rho_m U_m^3 D_m^2} = \frac{P_p}{\rho_p U_p^3 D_p^2}
\]

Equation 4-1

Therefore, assuming that the change in density and dynamic viscosity of air between the model and prototype conditions are negligible i.e. \( \rho_m = \rho_p \) and \( \mu_m = \mu_p \), then we can simplify the above relations to:

\[
\frac{H_m}{D_m} = \frac{H_p}{D_p}
\]

\[
U_m D_m = U_p D_p
\]

If we use a prototype wind velocity of about 4m/s as can be seen from a wind atlas \(^{14}\) available from Ministry of Energy 2004, then we can obtain our prototype rotor diameter from the dynamic similarity of \( Re \):

\[
D_p = \frac{U_m D_m}{U_p}
\]

Equation 4-2

For maximum performance, we select characteristics of our rotor as \( C_p = 0.14, D_m = 8.84\text{cm}, U_m = 10 \text{ m/s} \):

\[
D_p = \frac{10 \times 0.0884}{4} = 0.221\text{m}
\]

Then the height of the prototype, \( H_p \) rotor will be found using the linear scale factor relating \( H \) to \( D \):

\[
H_p = \frac{H_m D_p}{D_m} = \frac{0.078 \times 0.221}{0.0884} = 0.195\text{m}
\]

From this we can obtain the power, \( P_p \), obtainable from the prototype as:

\[
P_p = \frac{1}{4} C_p \rho_p D_p H_p U_p^3 = 0.25 \times 0.14 \times 1.2 \times 0.221 \times 0.195 \times 4^3 = 0.11584\text{ W}
\]
This provides the power used to pump water, \( P_{\text{hyd}} \) in Equation 2-6:

\[
P_{\text{hyd}} = \rho_w g H Q
\]

\( \rho_w = 1000 \text{ kg/m}^3 \) and \( g = 9.81 \text{ m/s}^2 \). This means we can have a potential output at any particular head of:

\[
H Q = \frac{P_{\text{hyd}}}{\rho_w g} = \frac{0.11584}{9.81 \times 1000} = 1.18 \times 10^{-5} \text{ m}^3 \cdot \text{m/s}
\]

Therefore if we have a gross head of, \( H_g \) we can determine our potential flow rate as follows:

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<th>( Q ) (m(^3)/h)*10(^3)</th>
<th>( Q ) (m(^3)/day)</th>
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<td>0.4248</td>
<td>0.01020</td>
<td>10.2</td>
</tr>
</tbody>
</table>

Table 6 Potential flow rates per gross head

These values are comparable with those found by K-V.O. Rabah \(^{[15]}\) who used two similar double stack Savonius rotors having a bucket diameter of 475mm diameter rotor with a height of 2m. These were constructed from 12.5 mm thick plywood sheets and 26 gauge plain galvanised iron sheets. The Bach type buckets (preferred due to its superiority to the usual semi-circular type in both torque and power characteristics at tow tip speed ratio range) were attached to the circular end and middle plywood discs using small bolts and nuts. This configuration was attached to the rotor shaft (made from 37.5mm mild steel tubes) using flanges at both ends. To minimise friction, the rotor shaft was supported with two self-aligning bearings of very low friction and the assembly mounted on a low tower (2m) high. With an overlap ratio (e/d) of approximately 0.14 was set for both rotors with and the aspect ratio (H/d) set at 4. (Note that the recommended values from other wind tunnel experiments are: overlap ratio of 0.1 - 0.15, and an aspect ratio of 4.29). The double-stack rotor is slightly superior to the corresponding single-stack in both power and torque characteristics, while that with end plates is greatly superior to that without end plates in power output and the width of the operating tip speed ratio range.

A 30-litre water tank was used for measuring the amount of water produced. The pump head was located at 10m. This was done on continuous basis throughout the day, for the purpose of determining the water output versus the wind speed characteristics of the rotor. The initial field test lasted six months. These were spread over to cover the period of low and high wind regimes. The Cp for the rotor was found to be 0.24. At a relatively moderate speed of 3.5m/s, the pump was able to produce 6m\(^3\)/day. For a locality like Kathiani (where the tests were carried out),
which is a typical rural setting in Kenya, this output corresponds to a daily domestic water requirement for about 80-120 people on the average.

4.2 Conclusion
The conventional Savonius turbine is a promising concept for small scale wind energy systems, but suffers from a poor efficiency. Therefore one of the objectives of the experiment was to identify an optimal design that would lead to higher values of efficiency and better self-starting capabilities as mentioned in the recommendations below.

In conclusion, we can say that with further slight modifications of some components, this type of rotor system can be said to be suitable for small scale wind energy conversion in regions with better wind regimes which is reasonable for most parts of the country as indicated in the wind atlas.

However, further research and field testing with improved designs in regions with better wind regimes should be encouraged. The beauty of Savonius (or any similar kind of windmills) is that they can also be coupled into a hybrid system to alternate between electricity generation and water pumping. This is very important, because it further enhances the energy resources of the rural-based communities. Moreover, the ease of construction and design modification meant that the system is well suited for technological transfer to rural-based community groups or organizations working in developing countries, and that at the end of the learning period, they (i.e., the rural people) would have gained sufficient skills to enable them to continue with the maintenance and further innovation upon the design.

Finally as per the results obtained in the conducted experiment, we can deduce that The Savonius turbine is applicable for small scale use in Kenya as per the operating winds available, as shown in the wind atlas, with the best application in the arid and semi-arid areas with mean wind speeds above 6m/s.

4.3 Recommendations
From literature \[^{16}\] to achieve better performance, the position of an obstacle shielding the returning blade of the Savonius turbine and possibly leading to a better flow orientation toward the advancing blade is first optimized (Figure 26). Adding a shielding obstacle should in principle reduce the reverse moment, and as a consequence the total moment of the turbine will be increased.
This recommendation goes a long way in improving the self-starting properties of the classical configuration. This was investigated and proved in the referenced article. It demonstrated that an obstacle plate had a considerable and positive effect on the static torque coefficient for the classical configuration. This can be clarified by doing an experiment in the fluid lab on a Savonius model using the wind tunnel to investigate the relationship between the static torque co-efficient with rotation angle for three different values of $Y_1$, compared to that of a Savonius without an obstacle.

Another recommendation would be to alter the geometry of the rotating blade shape, from a semi-circular shape, to a modified version as shown below:
This improves further the output power of the Savonius turbine as well as the static torque. An experiment done by Mohamed Hassan Ahmed Mohamed\textsuperscript{[16]} indicates an increase in $C_p$ by 38.9\% compared to the conventional Savonius design without an obstacle.

However due to the highest advantage of the Savonius turbine being in its robustness, modifications shouldn’t involve an exceedingly complex or expensive design. Therefore simple guiding plates would seem to be the best compromise between increase in efficiency and increase in cost and complexity. The effects of the number of guiding plates and various angles, together with the optimum obstacle shape employed in their setup should also be investigated. The results of the article were as shown in Figure 26:

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure26.png}
\caption{Optimum design of Savonius turbine with guiding plates\textsuperscript{[16]}}
\end{figure}
Figure 27 Instantaneous flow fields around optimum configurations (zoom) at the design point ($\lambda = 0.7$), static pressure (Pa), velocity and velocity vector magnitudes (m/s); a) classical Savonius with optimal guiding plates, b) optimal Savonius with optimal guiding plates. Note that the color scales are identical to facilitate comparisons.\textsuperscript{[16]}

The following graphs thus indicate the differences between the optimised and classical Savonius turbines:
This proves the higher performances of the optimised Savonius turbine as compared to that of the classic.

Building the entire structure from cheap and affordable steel alternatives would ensure that the wind pumps can withstand the harsh weather conditions in the tropics. This would also facilitate easier raising of the tower, thereby improving the wind catchment of the rotor, and hence, the general performance of the machine.

While modifying the components, the availability of raw materials locally and the skill to manufacture and maintain the technology must be considered. User affordability and needs as well as other complexities must also be taken into account.

4.4 Challenges and Difficulties

The following challenges and difficulties were experienced in the course of undertaking this project:

- Limited accuracy of the tachometer which may have brought about erroneous readings in the values of rotational speeds in revolutions per meter. This in turn would affect the calculations that proceeded and hence shape and accuracy of graphs obtained e.g. $C_p$ vs. Tip speed ratio.

- Vibrations of the setup Savonius model at speeds above 11m/s. This made measurements and characteristics above the speed of 11m/s unattainable, limiting the maximum wind speed we could subject our model to.

- The availability of only one type Savonius model, that lacked testing complexity, was a hindrance in its own. This meant the blade angles and gap spacing couldn’t be adjusted, in addition to the primary overlap, secondary overlap, rotor radius, diameter of blade, resulting in limited tests and results that could be carried out and obtained. This also limited the available literature useful for review as previous work has already come up with an optimum design of the Savonius rotor [17]: the optimal rotor (giving highest value of $C_p$) should be a double-stepped...
rotor with two paddles equipped with two end-plates which “canalize” the flow inside the rotor. The total height of the rotor should be twice its diameter.

- Lack of time that resulted in a restricted depth of research and experiments such as the difference between single stacked and double stacked Savonius turbines, two bladed and three bladed, and others as highlighted above. This was caused by the re-definition of the initial project undertaken over the first semester that had to be cancelled due to the unavailability of essential research material that resulted in changing the scope of the initial Tesla turbine into that of a Savonius turbine that was undertaken at the beginning of the second semester.

- A lack of co-operation from the Kijito Windmill correspondents, where we were meant to visit in a field work attempt to observe the practical applications and operations of wind turbines.
Chapter 5

5 References


[16] Mohamed Hassan Ahmed Mohamed; Design Optimization of Savonius and Wells Turbines; 2010; Magdeburg, Germany.

Appendix A

6 Appendix A [16]
The main component of a wind turbine is the energy converter, which transforms the kinetic energy contained in the moving air into mechanical energy. For an initial discussion of basic principles, the exact nature of the energy converter is irrelevant. The extraction of mechanical energy from a stream of moving air with the help of a disk-shaped, rotating wind energy converter follows its own basic rules. The credit for having recognized this principle is owed to Albert Betz. Between 1922 and 1925, Betz published writings in which he was able to show that, by applying elementary physical laws, the mechanical energy extractable from an air stream passing through a given cross-sectional area is restricted to a certain fixed proportion of the energy or power contained in the air stream. Moreover, he found that optimal power extraction could only be realized at a certain ratio between the flow velocity of air in front of the energy converter and the flow velocity behind the converter. Although Betz's "momentum theory", which assumes an energy converter working without losses in a frictionless airflow, contains simplifications, its results are still used for performing first calculations in practical engineering. But its true significance is founded in the fact that it provides a common physical basis for the understanding and operation of wind energy converters of various designs. For this reason, the following sub-topic will provide a summarized mathematical derivation of the elementary momentum theory by Betz.

6.1 Betz's Momentum Theory
The kinetic energy of an air mass \( m \) moving at a velocity \( U \) can be expressed as:

\[
E = \frac{mU^2}{2}
\]

Considering a certain cross-sectional area \( A \), through which the air passes at velocity \( U \), the volume flow rate \( Q \) (m\(^3\)/s) flowing through during a time unit, the so-called volume flow rate, is:

\[
Q = AU
\]

And the mass flow rate with the air density \( \rho_a \) is:

\[
m = \rho_a UA
\]

The equations expressing the kinetic energy of the moving air and the mass flow yield the amount of energy passing through cross-section \( A \) per unit time. This energy is physically identical to the power \( P \) in (W):

\[
P = \frac{\rho_a AU^3}{2}
\]
The question is how much mechanical energy can be extracted from the free-stream airflow by an energy converter?

As mechanical energy can only be extracted at the cost of the kinetic energy contained in the wind stream, this means that, with an unchanged mass flow, the flow velocity behind the wind energy converter must decrease. Reduced velocity, however, means at the same time a widening of the cross-section, as the same mass flow must pass through it. It is thus necessary to consider the conditions in front of and behind the converter.

![Image of flow conditions](image)

**Figure 29 Flow conditions due to the extraction of mechanical energy from a free-stream (M. H. Mohamed 2010)**

Here, $U_1$ is the undelayed free-stream velocity, the wind velocity before it reaches the converter, whereas $U_2$ is the flow velocity behind the converter. Neglecting any losses, the mechanical energy, which the disk-shaped converter extracts from the air flow, corresponds to the power difference of the air stream before and after the converter:

$$P = \frac{\rho_a A_1 U_1^3 - \rho_a A_2 U_2^3}{2}$$

Maintaining the mass flow (continuity equation) requires that:

$$\rho_a A_1 U_1 = \rho_a A_2 U_2$$

Thus,

$$P = \frac{\dot{m}}{2} (U_1^2 - U_2^2)$$

Where $\dot{m}$ = mass flow rate

From this equation it follows that, in purely formal terms, power would have to be at its maximum when $U_2$ is zero, namely when the air is brought to a complete standstill by the converter. However, this result does not make sense physically. If the outflow velocity $U_2$ behind
the converter is zero, then the inflow velocity before the converter must also become zero, implying that there would be no more flow through the converter at all. As could be expected, a physically meaningful result consists in a certain numerical ratio of $U_2/U_1$ where the extractable power reaches its maximum. This requires another equation expressing the mechanical power of the converter. Using the law of conservation of momentum, the force which the air exerts on the converter can be expressed as:

$$F = \dot{m}(U_1 - U_2)$$

According to the principle of "action equals reaction", this force, the thrust, must be counteracted by an equal force exerted by the converter on the airflow. The thrust, so to speak, pushes the air mass at air velocity $U'$, present in the plane of flow of the converter. The power required for this is:

$$P = FU' = \dot{m}(U_1 - U_2)U'$$

Thus, the mechanical power extracted from the air flow can be derived from the energy or power difference before and after the converter, on the one hand, and, on the other hand, from the thrust and the flow velocity. Equating these two expressions yields the relationship for the flow velocity, $U'$:

$$\frac{\dot{m}}{2}(U_1^2 - U_2^2) = \dot{m}(U_1 - U_2)U'$$

Thus, the flow velocity in the converter plane is equal to the arithmetic mean of $U_1$ and $U_2$.

$$U' = \frac{U_1 + U_2}{2}$$

The mass flow thus becomes:

$$\dot{m} = \rho_a AU' = \frac{\rho_a A(U_1 + U_2)}{2}$$

The mechanical power output of the converter can be expressed as:

$$P = \rho_a A(U_1^2 - U_2^2)(U_1 + U_2)$$

In order to provide a reference for this power output, it is compared with the power of the free-air stream which flows through the same cross-sectional area $A$, without mechanical power being extracted from it. This power was:

$$P_0 = \frac{\rho_a AU_1^3}{2}$$
The ratio between the mechanical power extracted by the converter and that of the undisturbed air stream is called the "power co-efficient" $C_p$:

$$C_p = \frac{P}{P_0} = \frac{\rho a A(U_1^2 - U_2^2)(U_1 + U_2)}{4 \rho A U_1^3}$$

After some re-arrangement, the power co-efficient can be specified directly as a function of the velocity ratio $U_2/U_1$:

$$C_p = \frac{P}{P_0} = \frac{1}{2} \left[ 1 - \frac{U_1}{U_2} \right] \left[ 1 + \frac{U_1}{U_2} \right]$$

The power co-efficient, i.e., the ratio of the extractable mechanical power to the power contained in the air stream, therefore, now only depends on the ratio of the air velocities before and after the converter. If this interrelationship is differentiated to get the maximum value of the power co-efficient it can be obtained that the power co-efficient reaches a maximum at a certain velocity ratio with $U_2/U_1 = 1/3$. The maximum "ideal power co-efficient" $C_p$ becomes:

$$C_{p_{(max)}} = \frac{16}{27} = 0.593$$

Betz was the first to derive this important value and it is, therefore, frequently called the "Betz factor". Knowing that the maximum, ideal power co-efficient is reached at $U_2/U_1 = 1/3$, the flow velocity $U'$ in the rotor plane becomes:

$$U' = \left( \frac{2}{3} \right) U_1$$

It is worthwhile to recall that these basic relationships were derived for an ideal, frictionless flow and that the result was obviously derived without having a close look at the wind energy converter. In real cases, the power co-efficient will always be smaller than the ideal Betz value, as shown:
The essential findings derived from the momentum theory can be summarized in words as follows:

- The mechanical power which can be extracted from a free-stream air flow by an energy converter increases with the third power of the wind velocity.
- The power increases linearly with the cross-sectional area of the converter traversed; it thus increases with the square of its diameter.
- Even with an ideal air flow and lossless conversion, the ratio of extractable mechanical work to the power contained in the wind is limited to a maximum value of 0.593. Hence, only less than 60% of the wind energy of a certain cross-section can be converted into mechanical power.
- When the ideal power co-efficient achieves its maximum value ($C_p = 0.593$), the wind velocity in the plane of flow of the converter amounts to two thirds of the undisturbed wind velocity and is reduced to one third behind the converter.
7 Appendix B