DESIGN AND ANALYSIS OF LOW-HEAD HYDROKINETIC WATER TURBINE

A Final Year Project Report

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by

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ABSTRACT

In general, there are two means to harness energy from water. One is the hydrostatic method that uses the potential head of incoming flow. The other one is a hydrokinetic method that utilizes the kinetic energy of incoming flow to rotate the incoming flow water turbine. Hydrokinetic turbines (HKTs) have become the primary source of hydrokinetic energy in the past few years because of increased global warming, urbanization, excessive use of non-renewable energies, world energy consumption, drastic climate change, etc. The most significant advantage we have that it requires less or no civil work, unlike the hydrostatic approach. This work presented here describes the design and analysis of a low head hydrokinetic turbine capable of providing 1KW of power used in open channel water flow with low flow velocities. The purpose is to make a turbine that could provide for the general public's varying power requirements capable of powering a small house or a small-scale power requirement in a forest or areas with springs or rivers passing nearby. Experiments are done to determine whether setting up such a system in open channel water flow will be successful. A small-capacity power generation system aims to discover a new way of providing electricity to needy people. To design such a system, a detailed study of different types of turbines is conducted. Vertical axis drag-type turbines are most suited for small capacity generation systems. Therefore, a Savonius type water turbine is designed using helical blades to give the best possible results. Varying some parameters give the improved power output. The mathematical modeling of the turbine is completed after the design is finalized. The turbine is portable and can be taken anywhere quickly.

ACKNOWLEDGMENTS

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ABBREVIATIONS

HKTs	Hydrokinetic Turbines
CFD	Computational Fluid Dynamics
AE	End plate area
AOA	Angle of attack
ER	End plate area ratio
AR	Aspect ratio
TSR	Tip speed ratio
AC	Turbine Cross-sectional area
DC	Direct Current
USEIA	U.S. Energy Information Administration

NOMENCLATURE

Symbol	Unit	Description
β	-	overlap ratio
F_D	Newton	Drag force
F_L	Newton	Lift force
Re	-	Reynolds number
Cp	-	Coefficient of performance
Ct	-	Coefficient of torque
Cd	-	Drag Coefficient
α	Degree	twist angle
D	Meter	Rotor Dia
e'	Meter	Shaft Dia
d	Meter	Blade dia
e	Meter	Overlap (including shaft dia)
A_s	Square meter	Swept Area
λ	-	Tip speed ratio
P	Watt	Power
P_A	Watt	Power available
ω	Rad / sec	Angular velocity
T	Newton meter	Torque
T_A	Newton meter	Available Torque

CHAPTER 1: INTRODUCTION

1.1 Background

For the past few decades, humankind has been in a constant search to catch, develop, and exploit ways of meeting the needs of the present-day generation without compromising the upcoming generations to meet their needs. Conventional non-renewable energy foundations such as natural gas, coal, and oil have proven highly efficient for energy needs. Still, on the other hand, they are damaging human health and the environment. World energy consumption will rise by 56% from 2010 to 2040. As the world population keeps increasing and the limited amount of fossil fuels starts to reduce, there is an increasing demand to exploit renewable energy sources, as stated by the U.S. Energy Information Administration (USEIA). So the energy source should be renewable and should have fewer effects on the ecosystem and environment. Due to its relative availability over a wide range of geographical terrains (water covering 71% of the planet earth), ease of access, and less civil work than the other small power-producing sources like the solar and wind, the Hydro Power reasonably became the most broadly adopted renewable source of harnessing energy [1]. Hydrokinetic energy, a comparatively new form of renewable energy, can be generated by utilizing the kinetic energy from natural sources like flowing water in rivers or oceans using a different rotor class, converting it into mechanical energy as power output at the shaft [2]. These hydrokinetic power turbines can be used to deliver power to remote areas or may be used to light a camp on the riverside. These kinds of hydrokinetic low head water turbines are more desirable in developing countries like Pakistan. There are different hydrokinetic water turbines for the production of energy from the natural sources of water streams.

Table 1: Classification of Hydrokinetic Turbines

Sr. #	Types based on the orientation of the rotor axis concerning water flow.	Types based on the pressure changes across the turbine blades.
1	Vertical axis turbines (crossflow turbines)	The impulse turbine works on the velocity of flowing water. [3]
2	Horizontal axis turbines (axial turbines).	The reaction turbine utilizes the head of the water for rotation. [3]

Horizontal axial turbines are expensive for small-scale and limited power generation. For modest power generation, vertical axis turbines are economical as they are less expensive, have less or no civil work, and require less maintenance [4]. Savonius and Darrieus helical water turbines are the most common types of vertical axis turbines. We are working on a modified savonius turbine to improve its power output. Savonius hydrokinetic turbines are drag-type turbines because drag force is the main driving or push force for the operation of savonius type turbines [5]. It has S-shaped blades, and it is simple in construction. Using savonius rotor in river and canal streams has many advantages as fixed flow direction and effective use of deflector plate.

1.2 Scope and Motivation

The biggest motivation is to save our ecosystem and the limited conventional non-renewable sources. With time, traditional energy sources are decreasing. The world is looking towards renewable energy resources to fulfill energy demands as renewable energy sources have significant environmental benefits over conventional sources. Listed below are some motivations regarding our developing country.

The common shortage of electricity in Pakistan, especially in rural areas, calls for other affordable power generation sources.

This will be particularly effective in areas with springs or rivers passing nearby since long-distance transmission at low voltages is not feasible.

1.3 Aims and objectives

The core goal of our project is to build a micro-hydrokinetic water turbine that can produce electricity up to 1000W, capable of powering up a small home. Our objective is to design such a water turbine to utilize the hydro energy of slow-moving water streams. Some primary goals of our work are as follows;

We are designing a Low Head Hydro Kinetic Turbine that is well matched to the environmental conditions of Pakistan and can easily be manufactured and installed in urban but especially in rural settings.

We are scaling up the power production capacity of the turbine to enable it to power a single household.

It saves manufacturing costs that must be substantially low to make it economically viable for domestic users and widely adaptable without compromising its durability and operation.

We are designing a Hydro Turbine to achieve maximum efficiency and co-efficient of performance by tweaking and optimizing relevant variables such as blade radius, aspect ratio, overlap ratio, tip speed ratio, swept area, etc.

The last objective includes the portable design of the turbine and its carrying mechanism to make it feasible everywhere.

1.4 Organization of thesis

This thesis is divided into five chapters. The first chapter consists of three sections, and the first section presents the background and trends in hydrokinetic energy systems. At the same time, the second section discusses the scope and motivation of low-head hydrokinetic water turbines. Finally, the last section represents the aims and objectives associated with the problem statement.

The second chapter elaborates on the studies already done on the low head hydrokinetic water turbine. In particular, the chapter highlights the research development in the field of the HKTs, presenting both experimental and simulation studies done previously. In this chapter, the impact of various arrangements and parameters are reviewed and presented.

The third chapter discusses the design and numerical modeling. Moreover, it includes solid works modeling and 2D CFD analysis, governing equations, mesh generation, and simulation procedure. Validation and convergence studies are also included in this chapter.

The fourth chapter presents the results and an extensive discussion of simulation results.

Lastly, the final chapter concludes the thesis, highlighting the findings of this research, the novelty of our work, and presenting suggestions and further study prospects.

CHAPTER 2: LITERATURE REVIEW

Electricity is considered as the most important invention of human history. Modern life and any country's economy are totally dependent on electricity. Electricity consumption is increasing day by day and to cater this we need to produce more electricity. But we have to take care of environment as well. Power plants that run on fossil fuels emit harmful gases that have significant contribution to climate change. Thus we need to focus more on renewable and green energy sources. In our case we choose hydro-power considering our country's climate and resources.

A lot of work has already been done on hydro systems but there is always room for some improvement. Many different types of hydro-turbines have been developed considering water flow and blade profiles. We have either conventional turbines that use water head (potential energy stored due to height) or Hydrokinetic turbines that use kinetic energy in flowing water. Hydro-kinetic turbines are more popular nowadays as they do not require water diversions through special channels rather uses free flow of water. Thus they do not need large civil structures and help in reducing carbon footprints.

Two main types of hydrokinetic turbines are axial turbine and cross flow turbine depending on orientation of rotor axis. As explained in previous chapter axial turbines are the one in which rotor axis is parallel to the water flow while cross-flow turbines are the one in which rotor axis is perpendicular to the water flow. Cross flow turbines are further classified as Darrieus and Savonius turbine. Darrieus type rotor uses lift force while savonius type rotor uses drag force as driving force. We will study these lift and drag forces in further detail.

Our focus is on Savonius type vertical axis turbine with a modification that design is completely portable and detachable. Here is the detailed study of different parameters affecting the performance of turbine.

2.1 Difference between Vertical and Horizontal Axis Turbine

Horizontal axis turbines are the ones in which rotating axis of blade is parallel to flow stream while in vertical axis turbines it is perpendicular to the flow stream. Horizontal axis turbines perform best under streamline flow conditions, i.e. they are not effective where the fluid flow is turbulent [6]. While Vertical axis turbines are effective for both streamline and turbulent flow, a brief comparison is shown:

Table 2: Difference between horizontal and vertical axis turbine

Sr. #	Properties	Vertical Axis Turbine	Horizontal Axis Turbine
1	Rotor adjustment Mechanism	Do not require	Yaw Control Mechanism
2	Inspection and Maintenance	Easy	Difficult
3	Efficiency	Lower	Higher
4	Power extraction from fluid	Low	High
5	Self-starting capacity	No	Yes
6	Installation process	Simple	Complex
7	Power coefficient and tip speed ratio	Low	High [7]
8	Weight and cost	Low	High
9	River stream Problems	Can deflect the debris	Usually clogged with debris [8]
10	Installation possibility in small rivers	Yes	No
11	Blades operation space	Low	High

Horizontal axis hydro turbine needs much more area of installation as compared to vertical axis turbine due to blade configuration. Moreover horizontal axis turbines are usually drag based while vertical axis turbines are usually lift based but they can be drag based too.

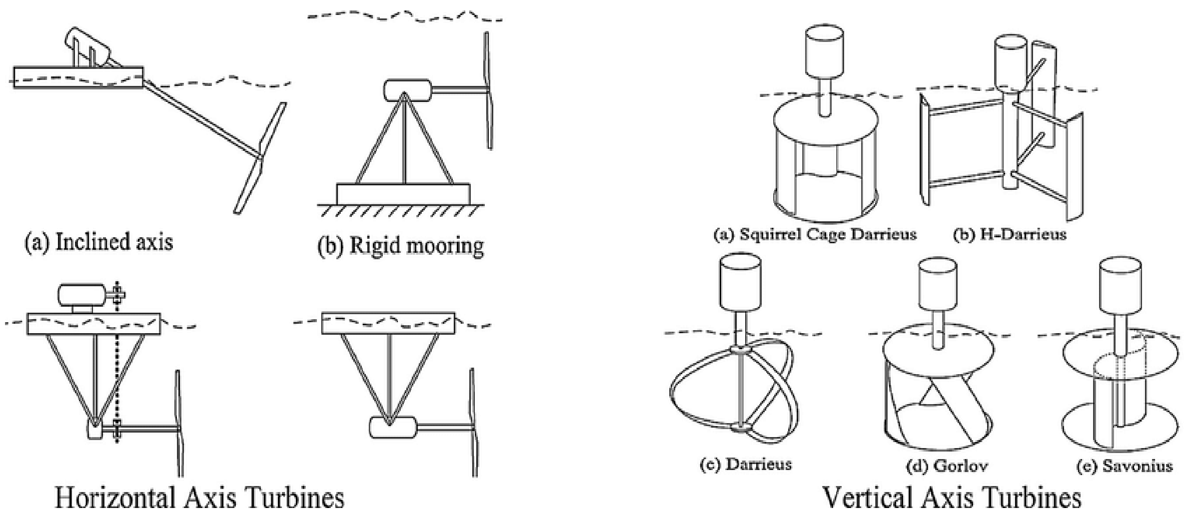

Figure 1: Horizontal and vertical axis turbine

Considering all the above properties we choose savonius vertical axis hydro kinetic turbine for our project with optimized blade profile. Savonius turbines are very cost effective and simple in design but these turbines are still less popular than conventional horizontal turbine.

2.2 Lift and Drag forces

Both lift and drag forces are mainly considered as aerodynamic forces but they play a major role in working of turbine as well. Lift forces are caused due to pressure of fluid while drag forces are due to resistance in forward motion. Lift and drag force acts orthogonally to each other and produce a combined effect on airfoil. Optimum performance airfoils are designed to maximize lift force and minimize drag force.

2.2.1 Drag force

It is a resistive force generated due tocontact of any solid body with fluid. [9] It can also be referred as a friction force as it opposes the movement of a body in fluid. Drag force is further classified as friction drag, pressure drag, lift based drag and wave drag depending on the stress causing it.

Friction drag also known as skin drag is due to surface shear stresses so it depends on the viscosity of fluid, surface roughness and fluid flow situation i.e. either laminar or turbulent. At laminar flow (low speed) frictional drag is very significant as Reynolds number is very low. Pressure drag also known as form drag is due to pressure difference between front and back side of the object. It also depends on the shape of the object that is for smooth streamlined object its value is low while for the objects causing flow disturbance its value is very high. Pressure and friction drag is shown as. [10]

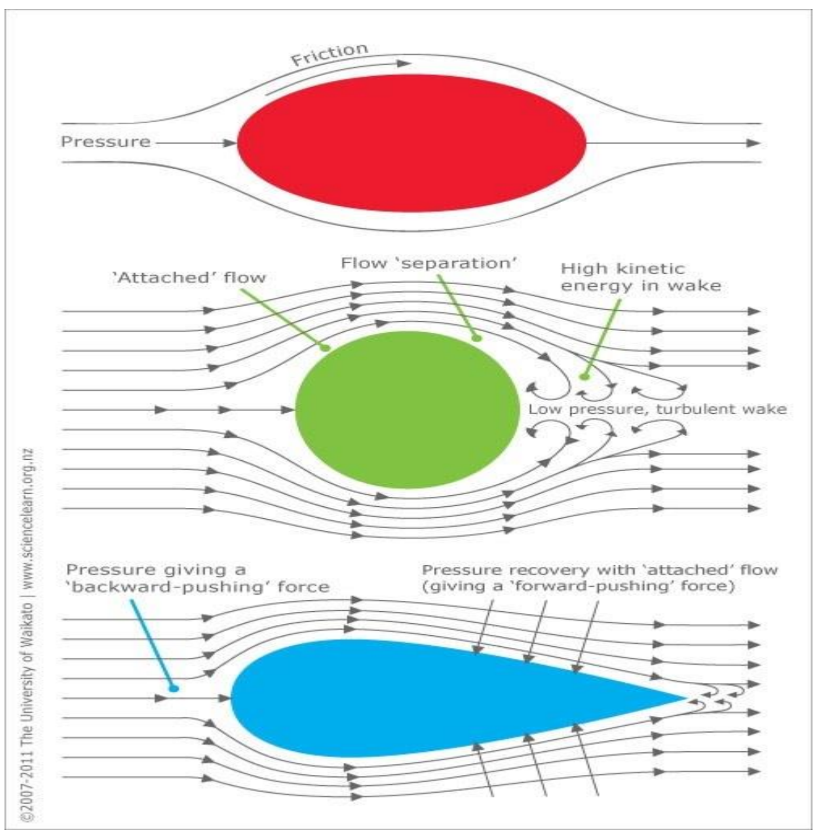

Figure 2: Horizontal and vertical axis turbine

Pressure drag occurs when air layers separate from each other and began to swirl causing turbulence. [10] When swirling increases it is impossible for the fluid to follow the surface thus it separates creating a very low pressure region at the back of body. Thus we can say that with the increase in pressure gradient (from leading to trailing edge) Form drag continues to increase as the AOA increases.

Drag force (F_D) is given as:

$$F_D = {}^1/_2\ \rho v^2 c_d A$$

Where ρ is density of fluid, c_d is drag coefficient, A is frontal area of object and v is velocity of object moving through the fluid. C_d is very important as it is used to represent drag force as a function of angle of attack (AOA) and Reynolds number (Re).

2.2.2 Lift force

It is the force that is required to raise a body in the fluid by exceeding its magnitude than gravitational force. In turbines it is caused by the flow of fluid over air-foil surface causing pressure distribution. The top and bottom of the airfoil is designed in a way that low pressure profile is generated on the top surface while high pressure contour at the bottom surface. This variation in pressure forces the airfoil to move upward.

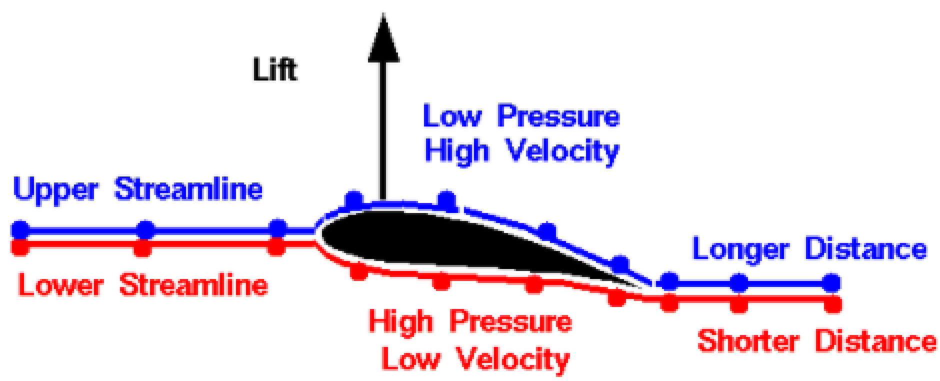

Figure 3: Lift force and pressure distribution

According to many sources the amount of air before and after the airfoil should be same keeping in mind the conservation of mass principle but the difference in top and bottom path lengths causes the air particles to attain different speeds. The particles on the upper side have to cover longer distance so they achieve high speed causing pressure to drop on the upper surface. This phenomenon is described as ‘Longer path’ or ‘Equal Transit’ Theory. But it is incorrect theory according to NASA [11]. But actually according to continuity law, we know that surface area at top of foil is larger so it displaces more air and this displacement of fluid increases the velocity of air. As velocity increases pressure decreases according to Bernoulli’s equation. Thus pressure difference occurs and air foil moves upward [12]. Lift force (F_L) is given as:

$$F_L = {}^{1}/_{2}\ \rho v^2 c_L A$$

C_L is lift coefficient and it is very important as it has different values for different shapes of foils depending on AOA.

2.3 Savonius Hydrokinetic Turbine

As described earlier Savonius Hydrokinetic turbine is a cross flow vertical axis hydrokinetic turbine. Hydrokinetic turbines are low head or zero head turbines that do not require high water pressure for their working rather they use slow moving water to extract energy.

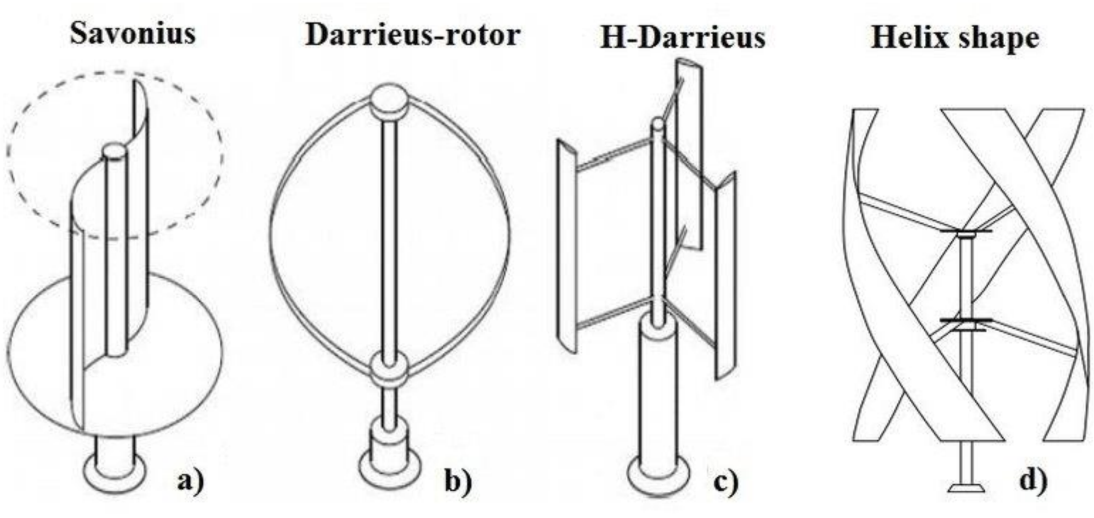

Figure 4: Various types of Vertical Axis Turbines

As shown in above figure Savonius rotor is an S-shaped rotor that may or may not some have overlapping between its blades.

Conventional savonius rotor has the semi-circular blade profile with small overlap. It is a drag type turbine as water exerts drag force on both (advancing and returning) blades to move them. Advancing blade is the one that has concave face in the direction of incoming flow and it experiences more F_D as compared to returning blade. [13] The difference in both forces keeps the rotor rotating.

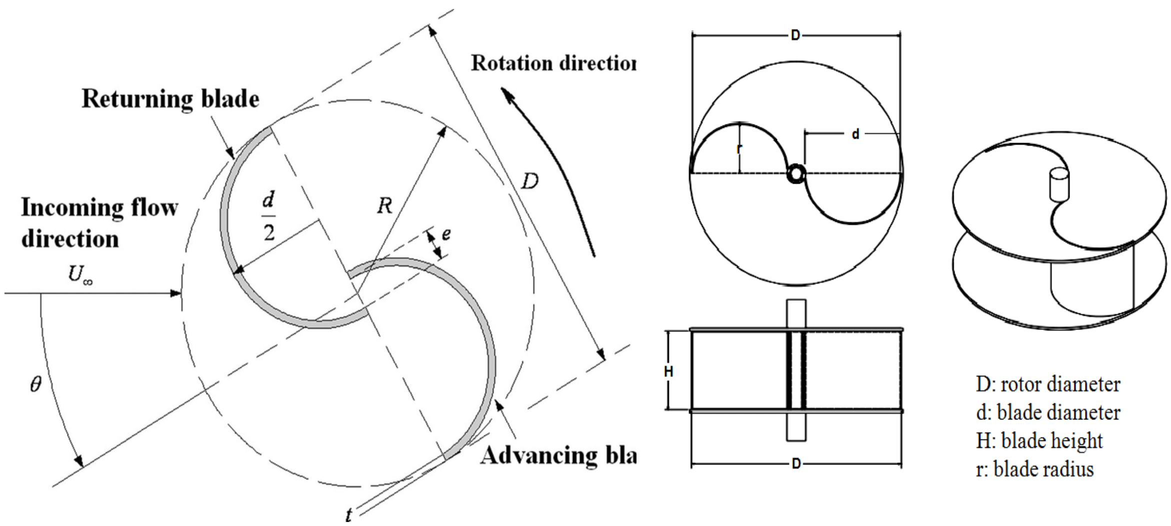

Figure 5: Conventional Savonius Rotor's schematic diagram

The 'e' in this figure represents blade overlap while 'd' represents blade diameter and their ratio represents overlap ratio 'β' i.e. $\beta = e/d$. Both savonius wind turbine and hydrokinetic turbine have countless similarities between them but hydrokinetic turbine is more energy efficient as water density is nearly 835 times more than air which helps in easy energy extraction.

Savonius Hydrokinetic turbine can work at very low water speeds as low as 0.5 m/s and low tip speed ratios. From literature it is found that geometric, operational and flow parameters affect the efficiency of savonius turbine significantly. The maximum performance is however indicated by Betz limit (cp = 0.593).

2.4 Design Parameters of Savonius Hydrokinetic Turbine

Geometric configuration of Savonius rotor is fundamental in its design study while there are many parameters that affect its performance. All of these parameters are listed below:

2.4.1 Aspect Ratio

It is a dimensionless design criterion that can be defined as ratio of rotor height to its diameter.

$$\text{Aspect ratio (AR)} = H/D$$

Alexander and Holownia [14] performed various wind tunnel tests on a number of Savonius rotor geometries with different wind speeds and concluded turbine performance is directly proportional to aspect ratio i.e. performance is better when AR is high. The effect on performance of a wind turbine for different wind speeds is shown. [15]

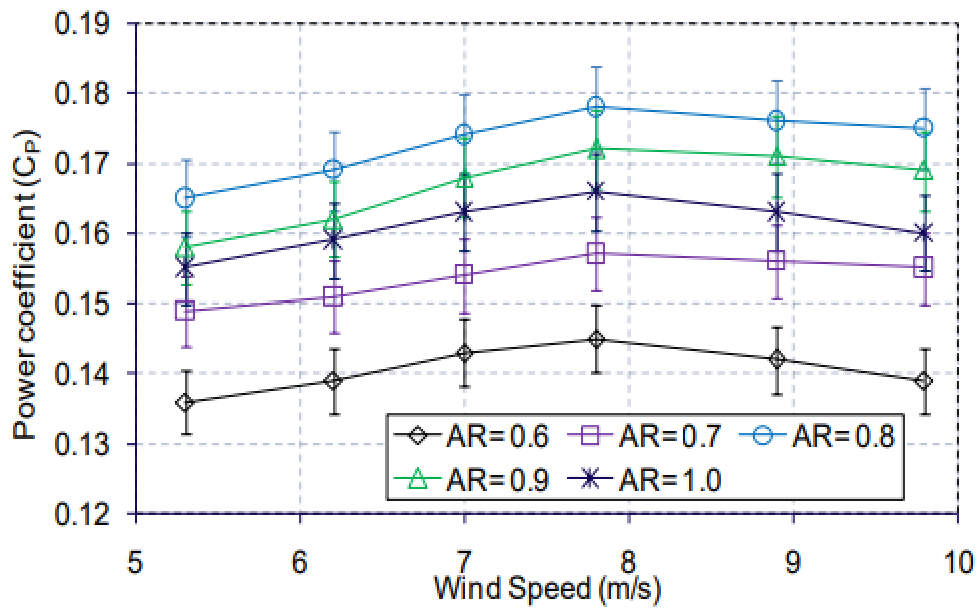

Figure 6: Performance Variation with different rotor aspect ratios

It is clear that performance of turbine is best at 0.8 aspect ratio. Although this graph is for wind turbine but we can understand the effect of AR from it very easily due to operational similarities between water turbine and wind turbine and the attained results are best for both .Studies also divulges that rotor with aspect ratio in range of 1.0 to 2.0 shows better performance.

2.4.2 Overlap Ratio

It is the ratio of blades overlapped length to the diameter of a blade. From literature effect of OR on the Savonius rotor's performance is shown in following figure:

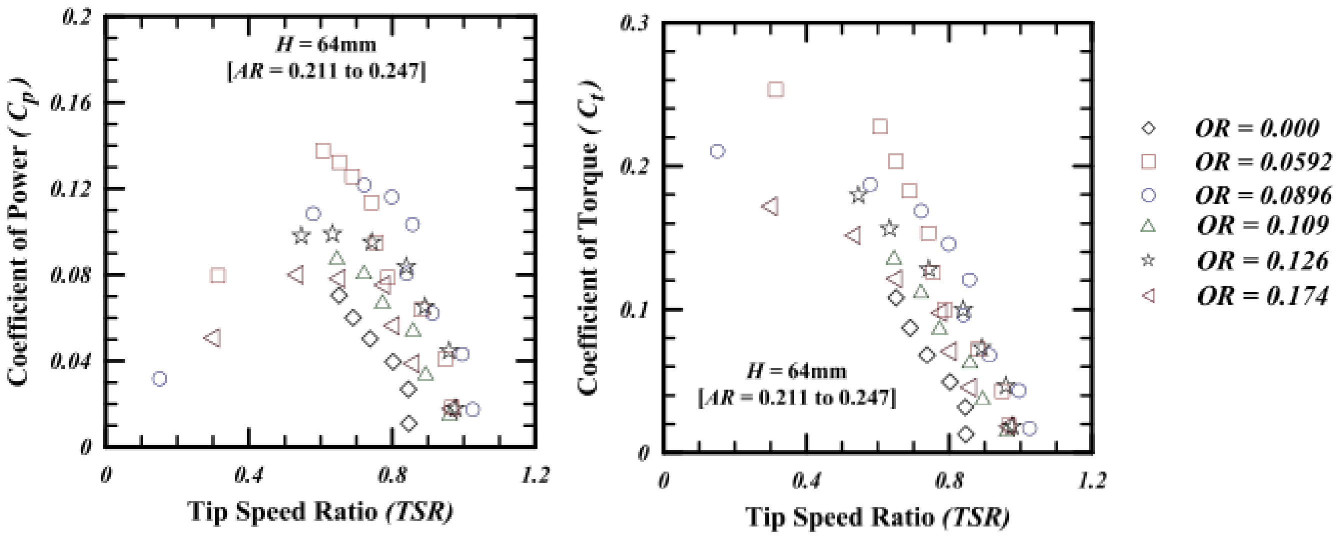

Figure 7: Performance Variation with different rotor overlap ratios

C_p, C_t, TSR all are determined using corrected velocity by **Maskell's correction method** [16]. From Literature we found out that these experiments are performed unless the turbine stops rotating due to load. Usually, $C_{p\,(max)}$ is observed within a TSR of 0.5 to 0.8. And it can be seen at zero overlap ratio condition that both Cp and Ct increases with increase in TSR but after some time both start decreasing.

2.4.3 Tip Speed Ratio

Tip speed ratio is one of the major concerns while designing turbine blades. From study we found out that by increasing the tip speed ratio velocity of fluid decreases in the rotating region. And increasing the tip speed ratio increases power significantly [17]. We need an optimal TSR for our design as if the turbine rotor rotates very slowly it cannot capture majority of the water due to which very low amount of water will pass through the turbine rotor. However, if the rotor rotates too fast, then it will always travel through used turbulent water. TSR affects both coefficient of performance C_p and torque coefficient C_t along with overlap ratio and end plate ratio. The effect on velocity is shown as:

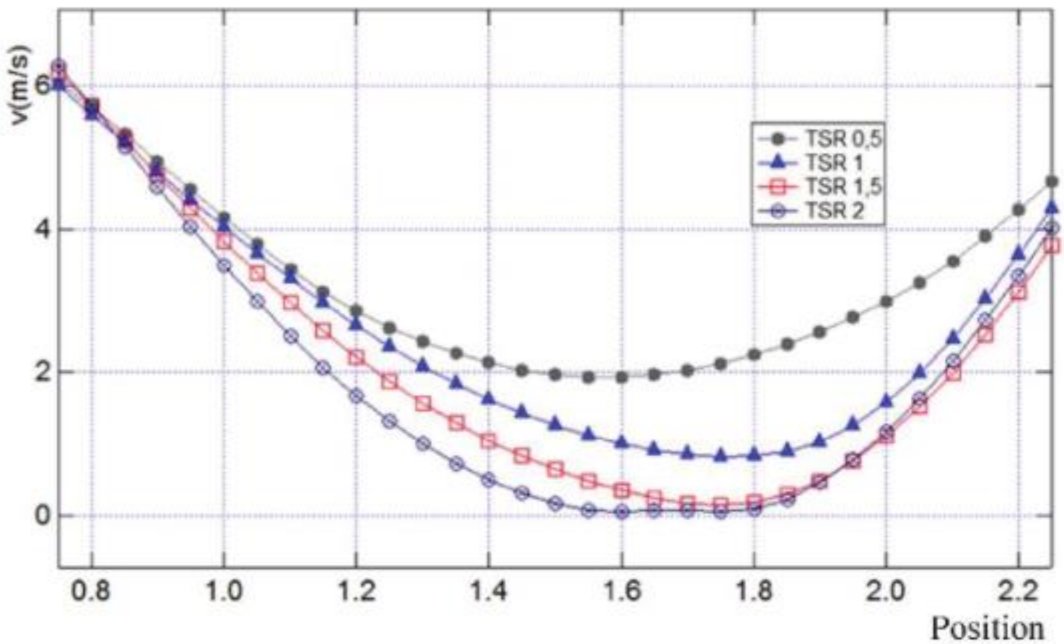

Figure 8: Effect of TSR on velocity of turbine

2.4.4 Blade Shape

Savonius rotor blade profile is an important criterion which affects the rotor performance. A lot of researcher work has been done on 2D and 3D profiles with a certain twist angle to maximize *Cp* of the rotor. Roy and Saha [18] tested experimentally four blade profiles for a two blade savonius rotor and established their own new modified model. The maximum *Cp* of the newly developed model was 0.31.

The tangential velocity is affected by the shape the blade. We can say that the more precise the turbine blade, the more resulting tangential force [19]. From Literature different blade shapes were examined and we found out that maximum tangential velocity occurs in the form of blade bowl [20]. So, we have chosen our blade profile accordingly. The graph shown below shows the effect of blade shape on tangential velocity for three different numbers of blades.

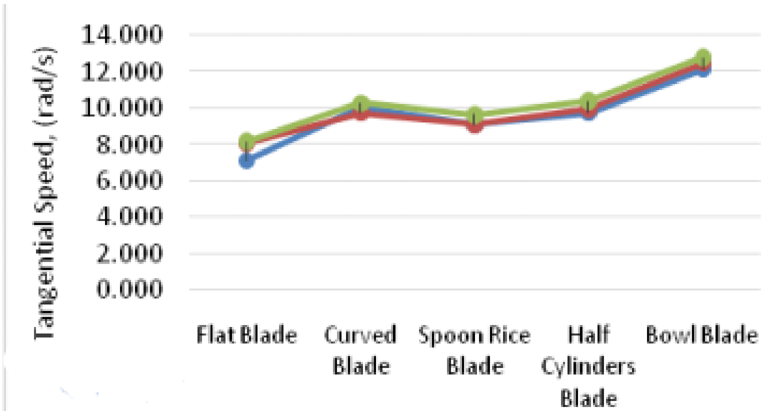

Figure 9: Tangential speed variation due to blade shapes

2.4.5 Number of Blades

No of blades plays a significant role in turbine performance. As we increase number of blades, variation can be reduced in the dynamic and static moment, angular position of vanes of a Savonius rotor. From figure we can see that number of blades and turbine power has inverse relation with each other. For the 14 blades turbine we have maximum turbine output in this case [19].

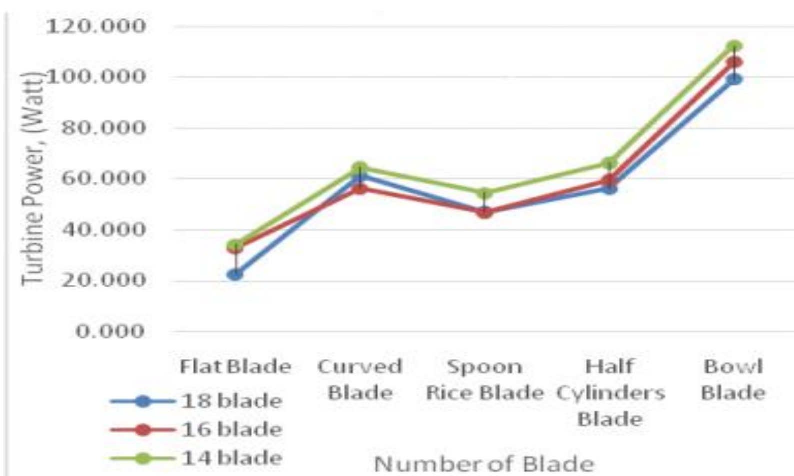

Figure 10: Effect of number of blades on turbine power

Turbine performance decreases with the increase in no of blades so it can be easily inferred from the experimental data and results that a two-bladed rotor system gives ideal performance. Mahmoud et al. [21] performed experiments to show the performance of single and double stages Savonius wind turbines. To find an ideal number of blades the OR was considered zero. It was concluded that the two blades rotor offers better performance than three and four blades rotors for all aspect ratios as well as for single or double stages too.

2.4.6 Number of Stages

Combining of rotor blades over one another is known as multi-staging. From literature we can find that two stages have better efficiency than singe stage. The stacking of rotors is usually done with a phase shift angle of 90 degrees. A standard Savonius rotor does not have self -starting capacity because of its very low starting torque. Numbers of stages are usually increased in order to improve starting torque characteristics

From literature we listed some experimental results comparing c_p and number of stages.

Table 3: No. of stages vs Coefficient of performance

Sr. #	Turbine tested	Stages	Optimum no. of blades	Optimum c_p value
1	2 blade savonius rotor	1-3	1	0.157
2	2 blade semi-circular savonius	1,2	2	0.29
3	Semi-circular twisted savonius	1-3	2	0.31

From table we can conclude that two-stage turbine with twisted blades is ideal for minimizing variation in torque and maintaining high c_p value. For three stages torque variation was minimum, but due to greater inertia power coefficient decreases. Therefore, two stage rotors are optimum to use.

Kamoji et al. [22] studied the performance of single stage, two stage and three stage Savonius rotor. From figure we can see that two stage rotor has optimum performance at every rotor angle.

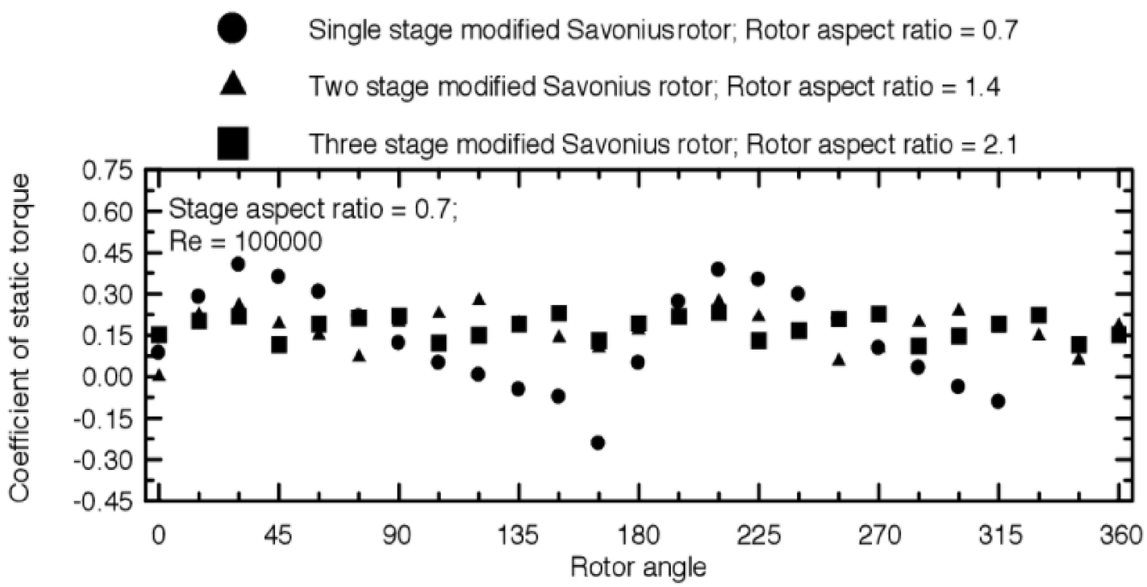

Figure 11: Effect of number of stages on coefficient of static performance

Reynolds number was considered around 80,000 and 10,000 for this experiment and stage aspect ratio was around 0.7, when we increase no of stages aspect ratio of rotor decreases.

2.4.7 Rotor Angle

Rotor angle affects the static torque and power generation significantly. Torque increases with increasing fluid speed and optimum value is achieved when rotor angle is perpendicular to the flow. [23]

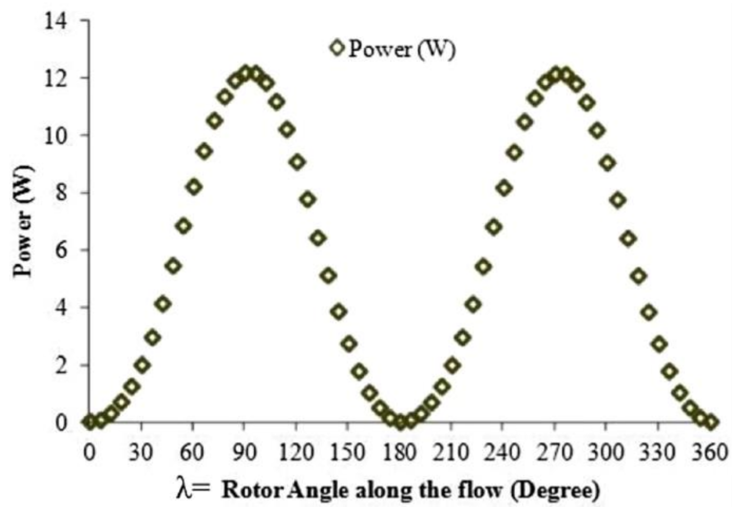

Figure 12: Power generation at different rotor angles

2.4.8 Twist Angle

A twisted blade will always have a better performance than a non-twisted simple blade. Syamsul [24] performed different experiments with twist angle α=0° to 25° and the test results shows higher performance than non- twisted blades of Savonius rotors. These performance parameters include higher efficiency, excellent spin and better starting characteristics. Next figure illustrates the relation between twist angle and coefficient of performance.

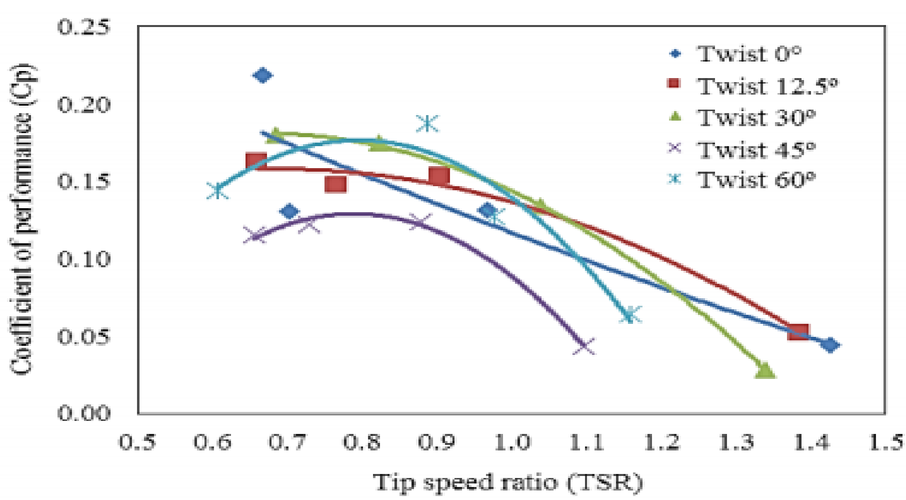

Figure 13: Variation in C_p value due to twist angle

2.4.9 End Plate

Coefficient of power is calculated by taking in account the corrected velocity due to blockage effect. There is a huge c_p difference in experimental data for both open and closed ended vanes, as fluid particles flows over from bottom and top end of the open ended vane causing no contribution towards rise in stagnation pressure at the upper side of advancing vane [16]. Thus the torque in open ended vanes is significantly due to stagnation pressure. In simple words end plates actually stops the seepage of fluid from the concave side of the blades to the external flow thus maintaining the pressure difference between the concave and the convex sides of the blades at normal levels [25].

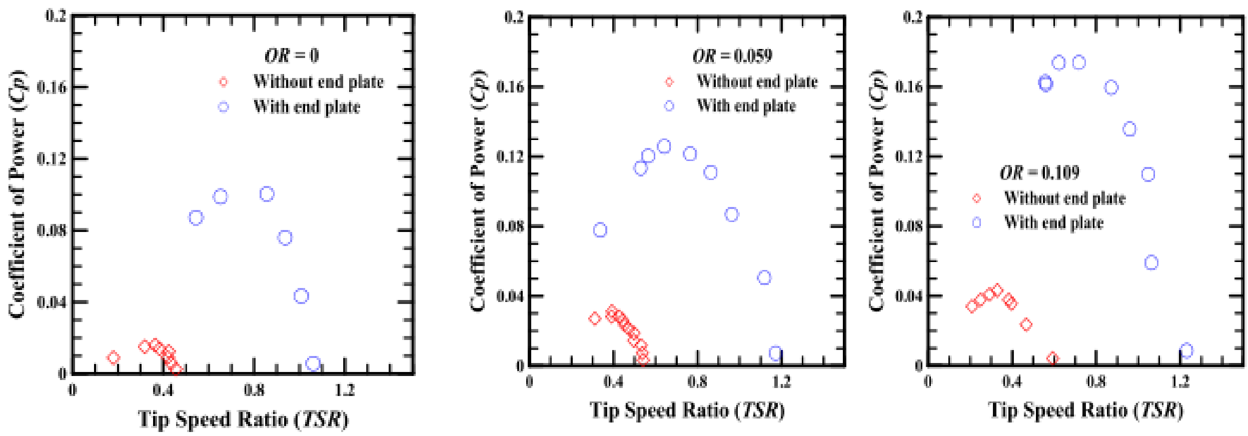

Figure 14: Performance of Savonius turbine for different overlap ratios and end plates

We can clearly see from above figure that for given overlap ratios, closed ended vanes have far better performance than open ended vanes. In conclusion, by changing the distance between the closed ended vanes the rate of rise in maximum c_p with end plate is quite high.

End plate area ratio (ER) is the ratio of the area of the end plate and cross-sectional area of turbine perpendicular to the rotating axis (¼ πd2) i.e. **ER = AE/AC**. Four different end plates were taken into consideration for the study as shown: [26]

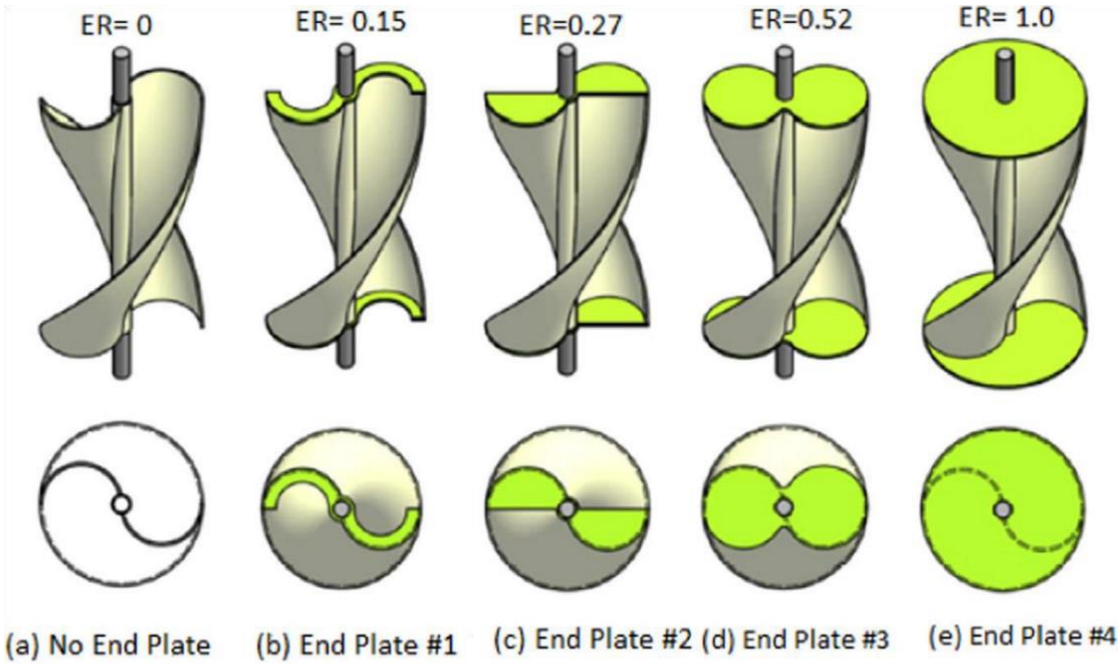

Figure 15: Different types of end plates

From study we found out that end plate no. 4 is best choice for our project as it maximizes fluid blockage and increases performance of turbine.

2.4.10 Deflector plate

Deflector plates are usually installed to direct the water flow to the turbine or rotor blades. This increases turbine performance as most water was directed already. [27] Several geometric parameters required for installing deflector plates are shown.

Configuration No.	$\frac{X_1}{R}$	X_1 (mm)	$\frac{X_2}{R}$	X_2 (mm)	$\frac{Y_1}{R}$	Y_1 (mm)	β
1	1.102	135	1.102	135	0.450	55	90°
2	***1.240***	***152***	***1.102***	***135***	0.450	***55***	***101°***
3	1.877	230	1.102	135	0.450	55	137°
4	1.102	135	1.877	230	0.450	55	44°
5	1.877	230	1.102	135	0.882	108	159°
6	1.877	230	1.102	135	0	0	123°
7	1.102	135	1.877	230	0	0	58°
8	1.102	135	1.102	135	0	0	90°

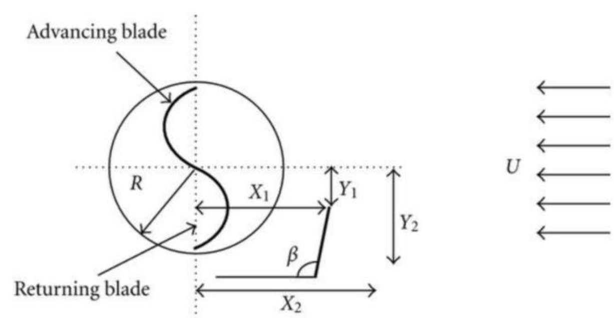

Figure 16: Geometric parameters for Deflector plate

2.4.11 Reynolds Number

In fluid dynamics Reynolds number is the most important criterion to define a fluid flow. Kamoji et al. [28] studied the Reynolds number effect for a Savonius rotor without shaft with an aspect ratio of 0.7,overlap ratio of zero, blade arc angle of 124°, and blade shape factor of 0.2. They concluded that with the increase in the Reynolds number, coefficient of performance also increases.

2.4.14 Installing Parameters

After design parameters there are many physical and installation parameters too that should be considered for better performance of turbine. These include direction of rotation of rotor, height at which the rotor will be installed, stream depth. Distance between the rotor and river bed with respect to diameter of rotor is also an important parameter to ensure maximum performance of turbine. This parameter is known as clearance ratio. Lesser the clearance ratio greater the coefficient of performance.

CHAPTER 3: METHODOLOGY

3.1 Design

3.1.1 Dimensions

Theoretical dimensions are calculated from the betz limit which is 0.593 and power usage of a normal house which we take as 18 kWh. Dimensions are scaled down for the prototype and following are the prototype dimensions.

3.1.1.1 Rotor Dimensions

Table 4: Design Parameters and their values

Length (One-Stage), L or H	0.92 m
Twist Angle	60 degrees
Pitch Angle	124 degrees
End Plate Dia	2.024 m
Rotor Dia, D	1.84 m
Shaft Dia, e'	0.1 m
Blade dia, d	1.002 m
Overlap (including shaft dia), e	0.211 m
Sheet Thickness	15mm

3.1.2 3-D Model of Rotor

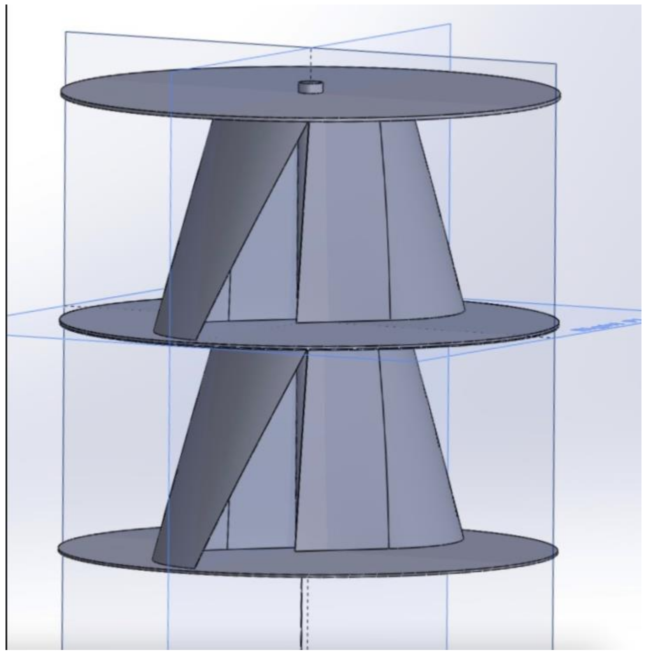

Figure 17: Front View of 3-D Motor of Rotor

3.1.2.1 Frame Assembly

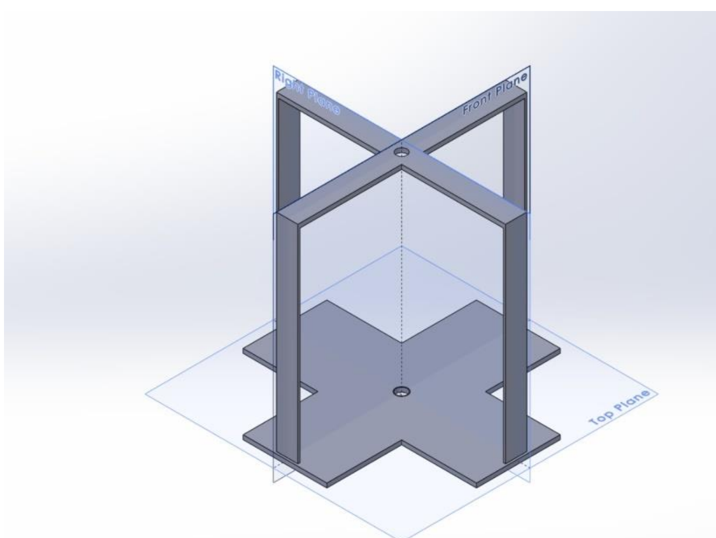

Figure 18: Side View of Frame

3.1.2.2 Assembly

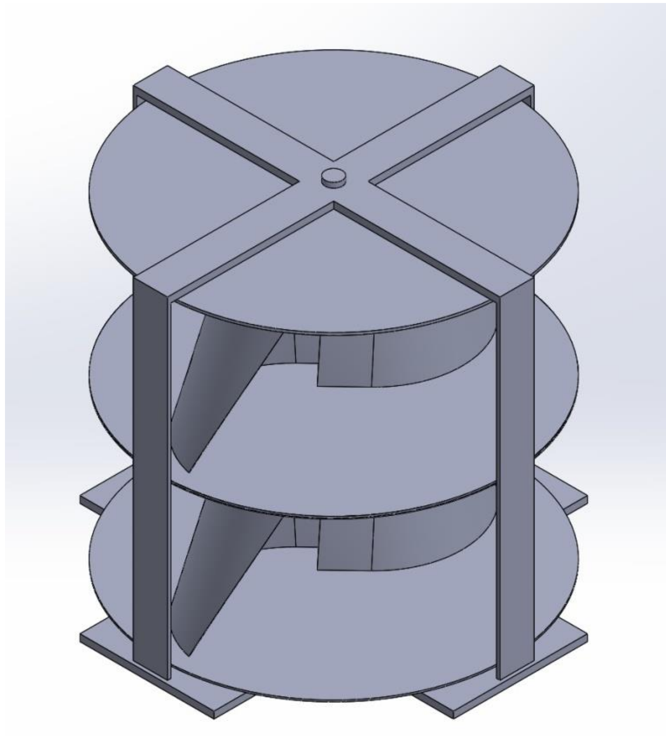

Figure 19: Side View of 3-D Assembly

3.2 MATHEMATICAL MODEL

The working principle of Savonius rotor is based on the difference of drag force between the advancing and the returning blades. This difference in drag forces creates torque in the rotor which results in the performance of the turbine.

3.2.1 Coefficient of Power

Coefficient of power of a Savonius turbine is given by:

$$C_P = \frac{P}{P_A}$$

It can also be written as:

$$C_P = \lambda \times C_t$$

3.2.2 Power Available

Power available in the water is given by:

$$P_A = \frac{1}{2}\rho A_S V^3$$

3.2.3 Power Generated

Power generated by the Savonius rotor is determined by:

$$P = T \times \omega$$

3.2.4 Swept Area

As the rotor turns, its blades generate an imaginary surface whose projection on a vertical plane to wind direction is called the swept area. The amount of energy produced by a wind turbine primarily depends on the rotor area, also referred to as cross-sectional area, swept area, or intercept area.

The swept area for Savonius wind turbine can be calculated from the dimensions of the rotor:

$$Swept\ Area = A_s = H \times D$$

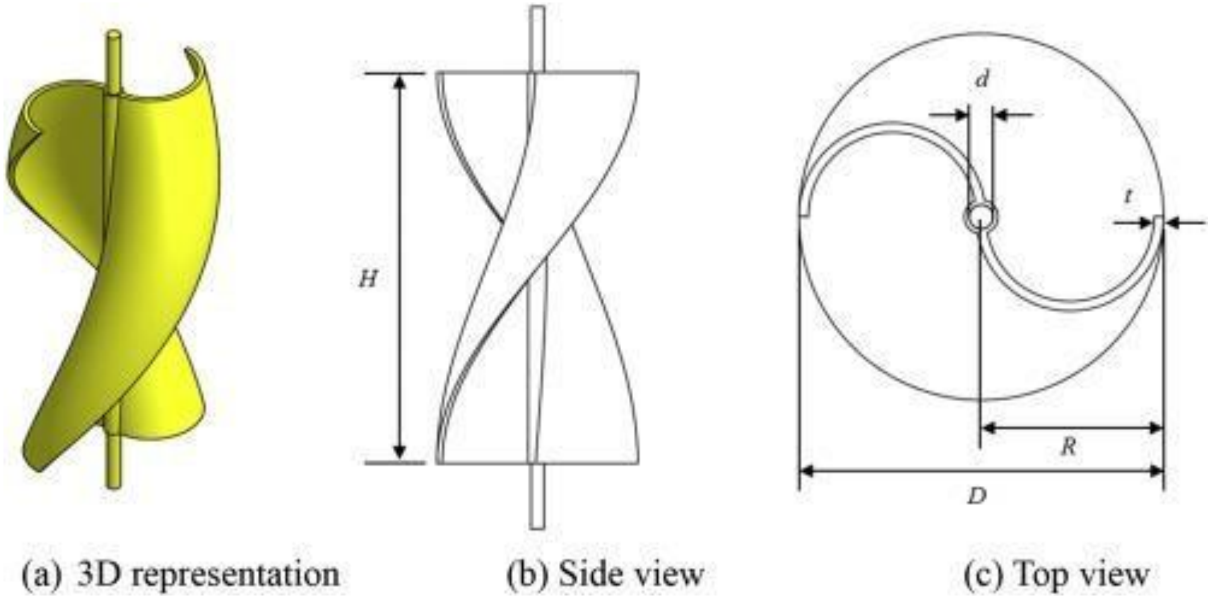

Figure 20: Rotor Blade

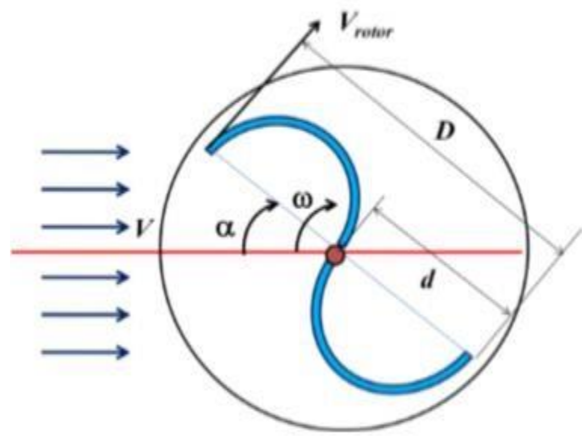

Figure 21: Schematic Diagram of a Savonius Rotor

3.2.5 Tip Speed Ratio (TSR)

The tip speed ratio is the ratio of the product of blade radius and angular speed of the rotor to the wind velocity. The tip peripheral velocity of the rotor (V_{rotor}) is defined as:

$$V_{rotor} = \omega \times \frac{D}{2}$$

Now the Tip Speed Ratio (TSR) of a turbine is expressed as:

$$The\ Tip\ Speed\ Ratio = TSR = \lambda = \frac{V_{rotor}}{V} = \frac{\omega \times D/2}{V}$$

3.2.6 Torque Coefficient (C_t)

It is defined as the ratio between the actual torque developed by the rotor (T) and the theoretical torque available in the wind (T_w), thus the torque coefficient (C_t) is given by:

$$C_t = \frac{T}{T_A}$$

3.2.7 Torque Available

Torque available is the maximum torque that can be produced in ideal case. It is defined as:

$$T_A = \frac{1}{4}\rho A_s D V^2$$

3.2.8 Required Torque

Torque produced by rotor is defined as:

$$T = \frac{F_A D}{2}\int_0^{\pi/2} \cos\theta \sin(\theta + \emptyset)d\theta$$

3.2.9 Force Available

$$F_A = \frac{1}{2}\rho A_S V^2 C_d$$

3.4 Portability

The main aspect of our design was that it was portable and could easily be assembled by anyone without any special training.

That was achieved by designing the whole blade assembly in such a way that each part had to be separately such that it could be portable while having the same amount of strength of the assembly. The rotors were designed separately and the end plates were modeled to have grove in them so as to have the blade slide I easily and kept in place.

To address the issue of blades moving or disassembling when the water flows through it, we used nut and bolts to hold the blades in a place. The bolt runs throw the overlaps of both blades as well as that of shaft, making sure that we have sufficient amount of strength as well as ease of removing the parts when needed.

3.4 Flow SIMULATION

SolidWorks Flow Simulation is used to perform CFD analysis of the rotor. The actual dimensions are used to study effect of flow on rotors.

Since we have only 1 parameter here, that is speed, we ran the simulation at different speeds varying between .5 to 1.25 m/s. Our main study speed is 1 m/s. All the others test speeds were to select the material on the basis of stresses that were generated. The difference in the drag coefficients of returning and advancing blades causes a generation of force on the advancing blade. The product of this force and rotor radius generates a torque. Therefore, values of Ct are written in a file and are later used to calculate Cp using the relation mentioned above.

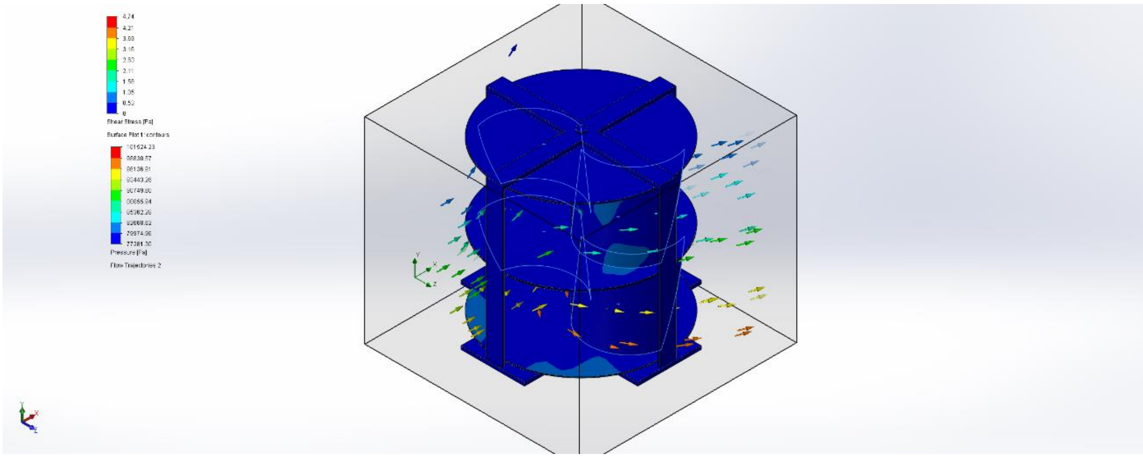

Figure 22: Shear Stress at 0.5 m/s

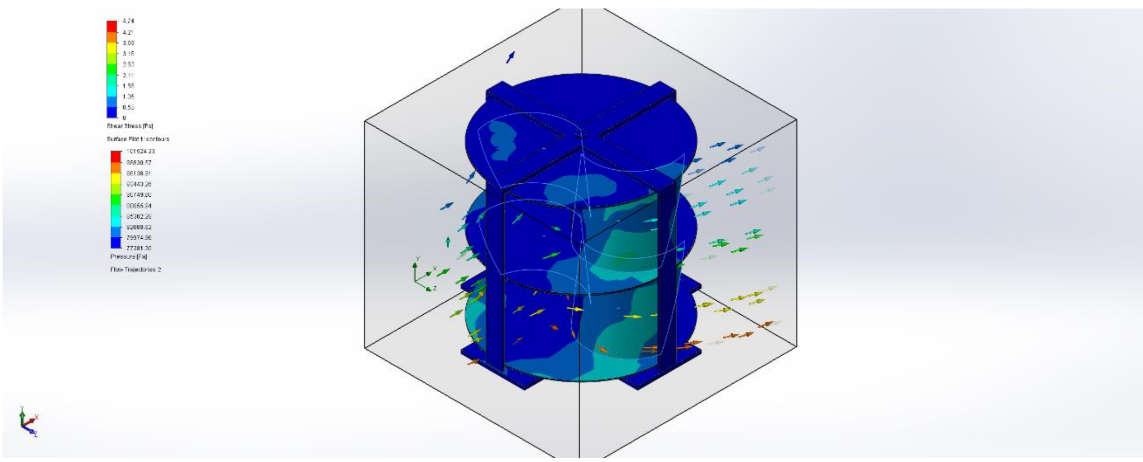

Figure 23: Shear Stress at 0.75 m/s

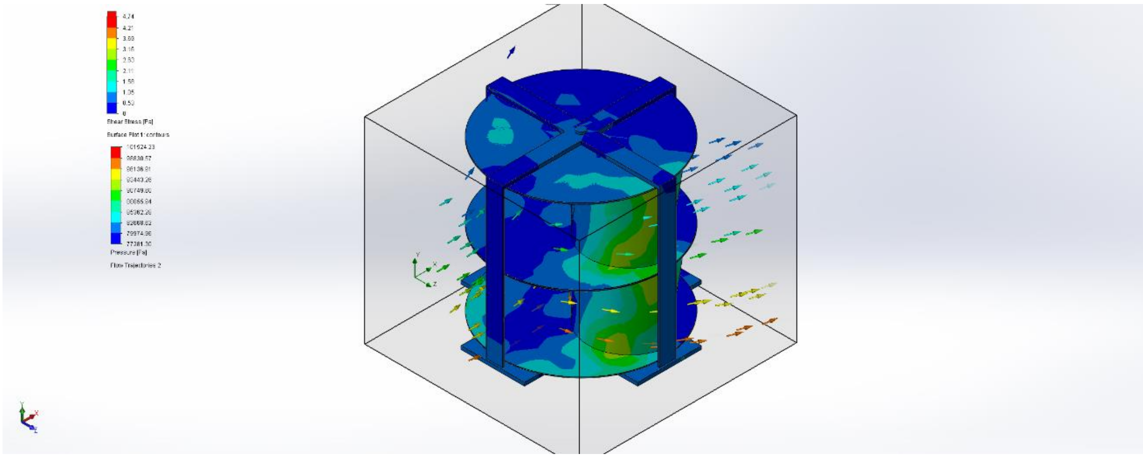

Figure 24: Shear Stress at 1.00 m/s

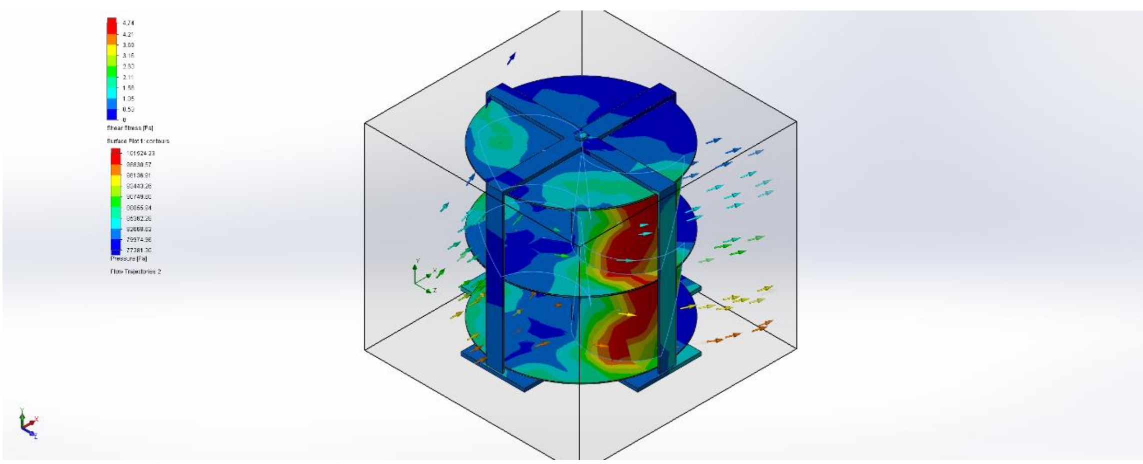

Figure 25: Shear Stress at 1.25 m/s

CHAPTER 4: RESULTS AND DISCUSSIONS

This chapter will cover the results that were obtained from our calculations and analysis and the discussion that entails. Furthermore, it will contain a comparison as well as the changes incorporated in the design, from which we took inspiration from.

4.1 Results

First of all we implemented the optimized blade design by using the S1048 section profile rotor. To visualize the behavior and performance of the rotors, the pressure is an important parameter. CFD analysis was performed on both the previous design and our new optimized design. The optimized savonius turbine shows a larger pressure difference between the concave and convex side of the blades, hence it provides a larger torque. As the rotation angle increases the pressure on advancing blade increases while the pressure on returning blade decreases. At a certain angle, the total pressure difference is higher hence at that location rotor attains maximum torque. This further increases the torque coefficient of the turbine as well. Figure 2 below shows the pressure contours. This can be seen in the stress analysis below:

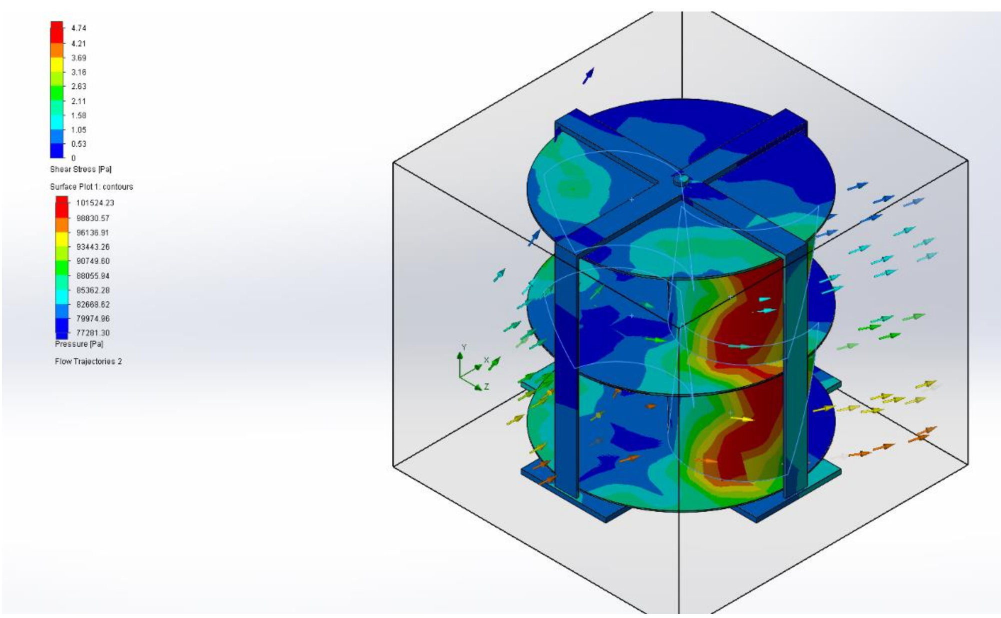

Figure 26: Isometric View of Pressure Contour

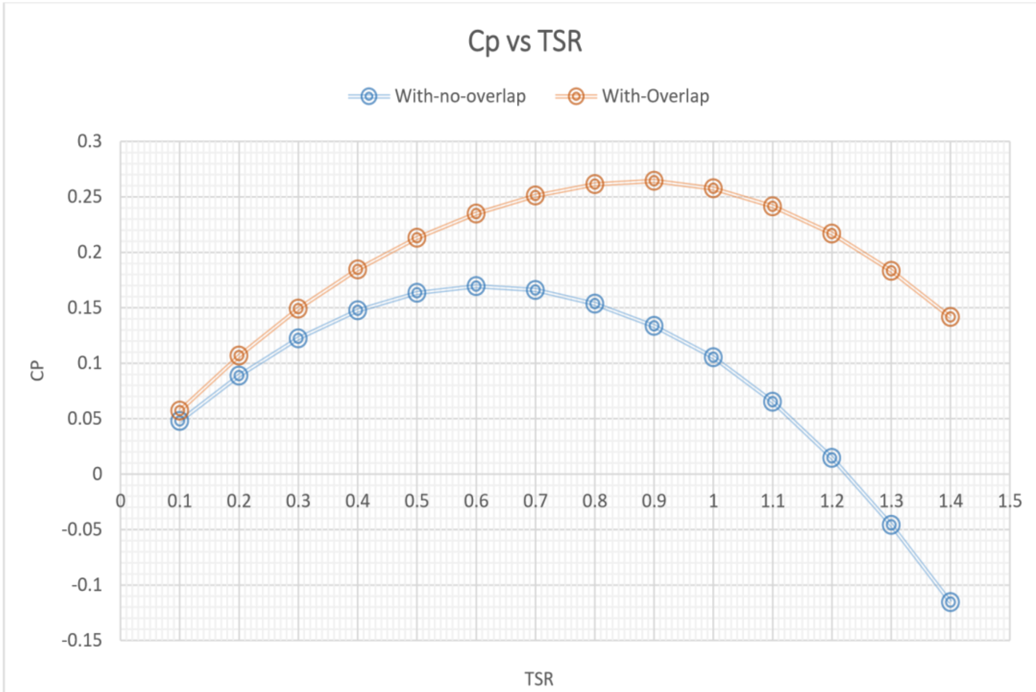

Figure 27: Cp vs TSR comparison performed on the previous model

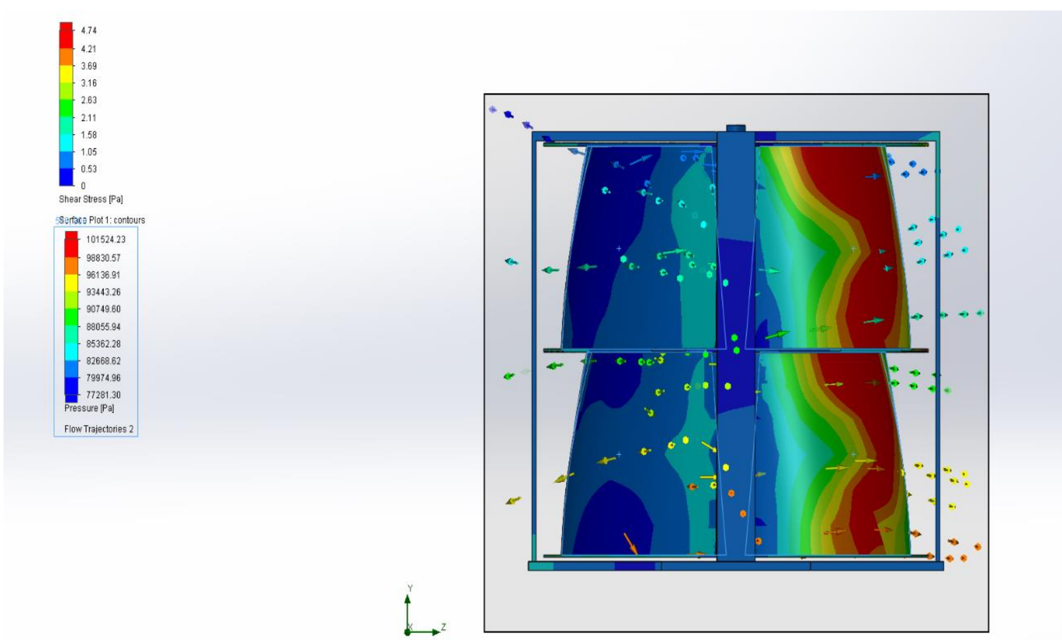

Figure 28: Shear Stress Distribution

We also increased the dimensions of the turbine to incorporate the greater power generation. Changes were made to the aspect ratio as well. The power coefficient changes depending on a few factors, primarily TSR and overlap ratio. Figure 28 shows the relationship, taken from Sarmad Saleem et. Al [29].

4.2 Comparison between our model and previous model.

The previous model was a horizontal axis hydrokinetic turbine whereas we have been working on a vertical axis turbine of a similar type. Our rotor also differs significantly from our predecessors. Using a S1048 section profile rotor in our design helped us in improving the torque and power coefficients of our turbine. In addition to this, changes in the aspect ratio and height were part of the upscaling. All of this helped us in reaching our goal of upscaling the power of the turbine.

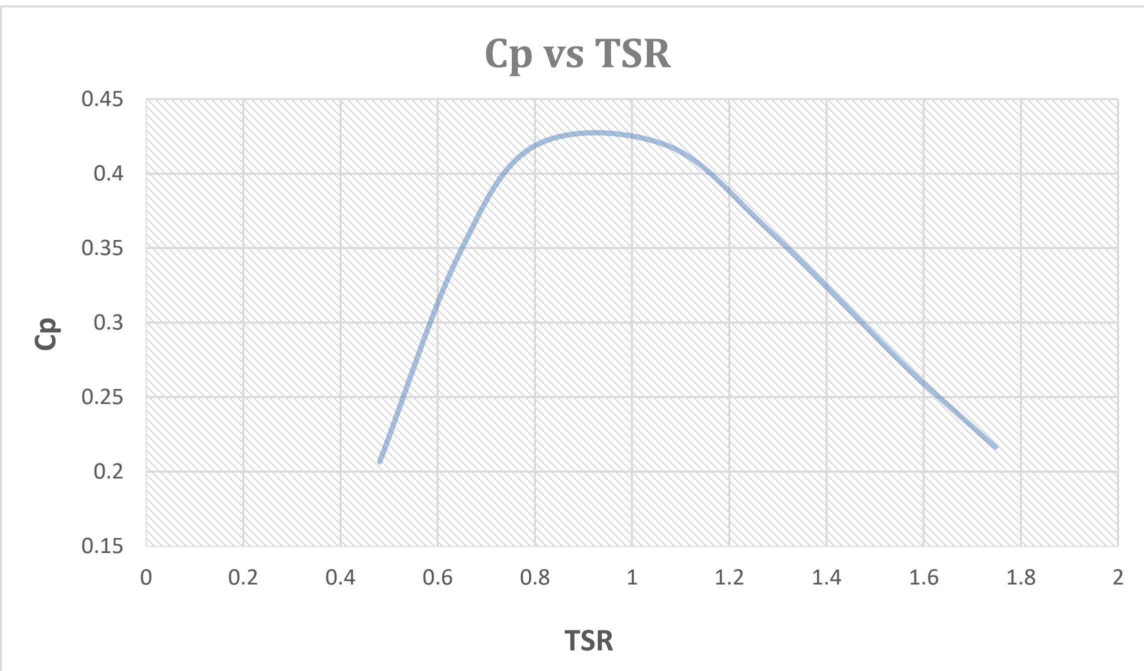

Figure 29: Cp vs TSR comparison performed on our model

Our design was successful in giving a reasonable Cp along with increased power output.

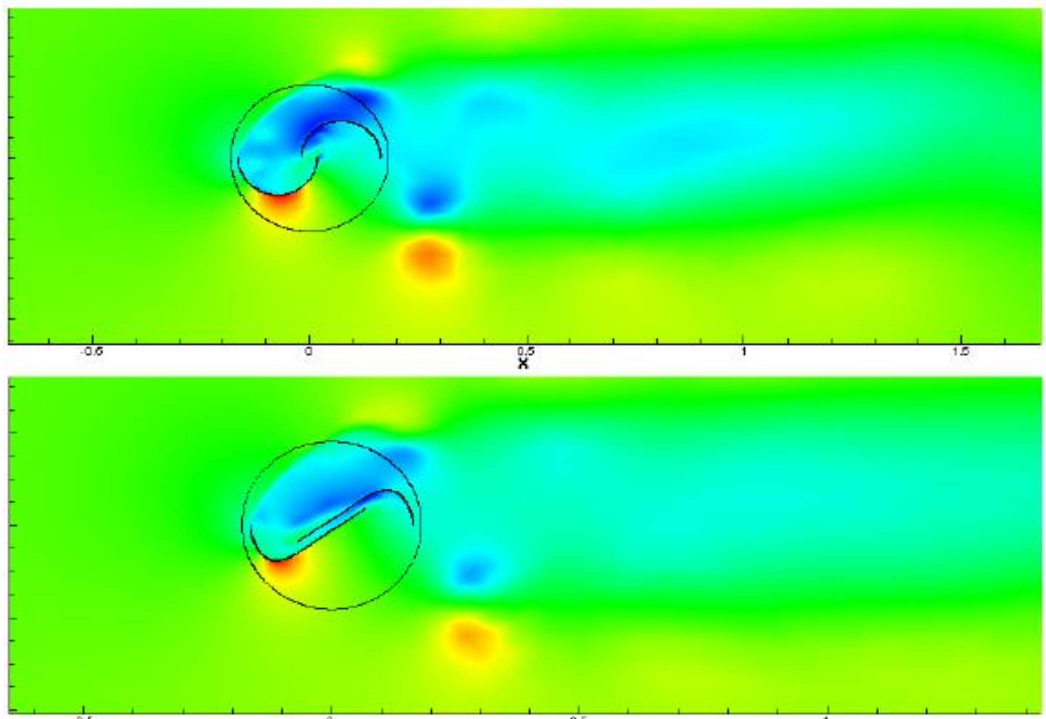

Figure 30: Velocity contours of conventional rotor and S1048 section rotor at rotor angle 360 degree

Using a S1048 rotor in our design as compared to the conventional savonius rotor, figure 3 above shows how the velocity contours around the blades vary for both rotors. At the rotor angle 360 degree, the velocity at the left-hand side is same as the inlet velocity specified in boundary condition. The maximum velocity zone has been found near the tip of the advancing blade. The Magnitude of tip velocity is higher for S1048 section rotor. Low velocity zone or wake zone exists behind the rotor in downstream. The length of the wake region is larger in the case of S1048 section rotor. The general difference between the velocity field of S1048 section rotor and conventional rotor is described as:

1. The S1048 section rotor has a higher magnitude of tip vortices. Due to this reason pressure reduction is high on the convex side of the blade.
2. The S1048 section rotor has a wider low speed wake zone which leads to extra pressure drop in the downstream region due to adverse pressure gradient.
3. The stagnation point in S1048 section rotor has shifted towards the tip of the blade which leads to higher pressure in this region.

Following are the pictorial comparisons of our current design with the previous model:

The above two comparisons show how our design varies in terms of blade design and model

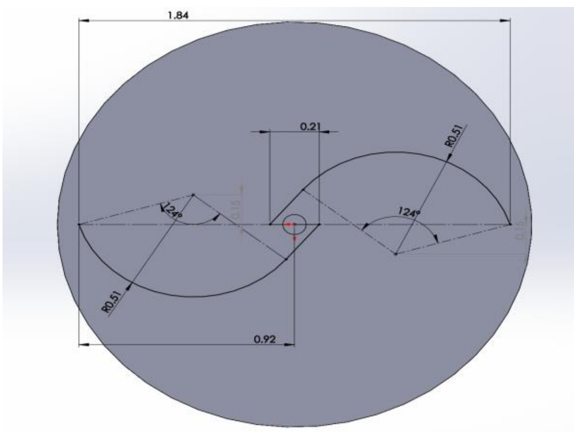

Figure 31: Blade specifications of current model

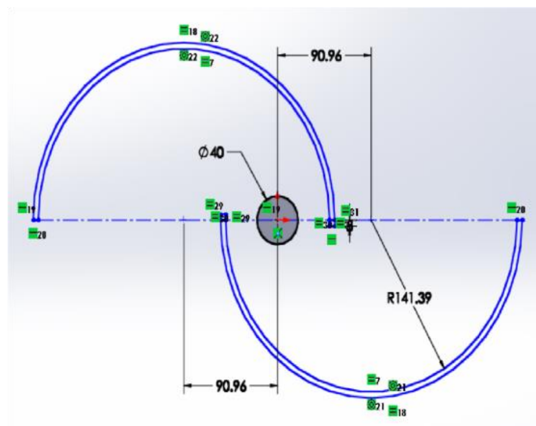

Figure 33: Blade specifications of old model

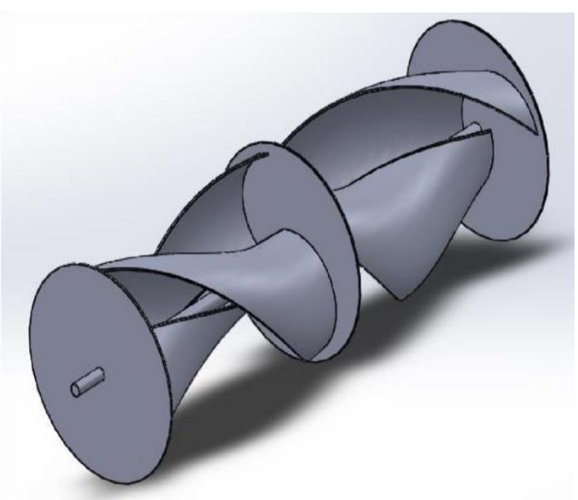

Figure 32: Conventional 3-D Model

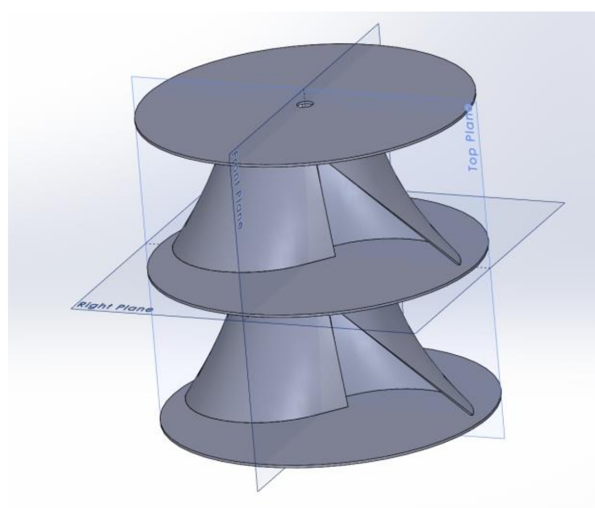

Figure 34: Optimized 3D model

shape and size. The semicircular shape of the conventional model was dropped and the above shape of the S1048 rotor was selected.

4.3: Electrical System:

The electrical system was designed to be submerged with the turbine so the generator can be fixed to the river bed. The central shaft of the turbine was extended from the top end plate and was connected to a horizontal shaft with the help of bevel gears with a gear ratio of 2:1. The generator has three phases and runs at a speed of 40 rpm. Furthermore, a simple control system for the generator was also setup [30]. It consisted of an emergency brake in case of an emergency along with a parking brake when the generator was temporarily out of order. Other than that, a DC inverter was also part of the control system in order to help with starting up the machine. Finally, the generator was connected to the required loads. Waterproof cables were used for the submerged wires.

In order to provide the standard 60 Hz power supply, the generator will have 180 poles while running at a speed of 40 rpm. An electronic governor can be employed to ensure the control of rpm.

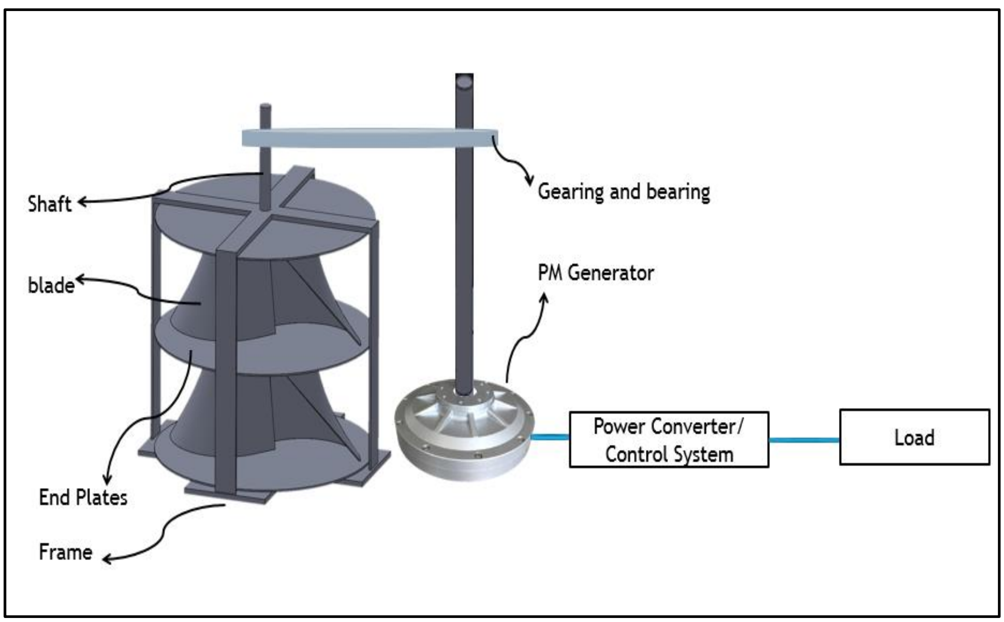

Figure 35: Electrical System

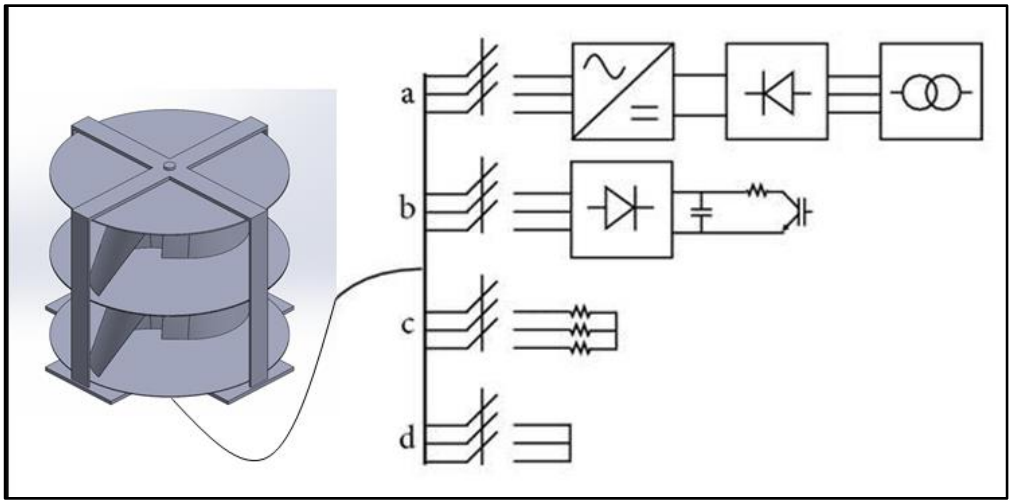

Figure 36: Control System

4.4 Analysis:

Ansys FLUENT is used to perform a 2-dimensional CFD analysis of the rotor. The dimensions are deduced using the maximum power required for a house and the Betz Limit. The dimensions of the rotor are then scaled down for prototyping. A 2D design of three different rotors is developed in Ansys Design Modeler. A fine mesh, as shown below, is generated to deduce accurate results. Water inlet velocity is set at 1 m/s. The flow is transient, and the k-epsilon model is used. This model keeps the constant mesh element size of about 550,000, and constant mesh generation is used to simplify the analysis. Ideally, this is between 500,000 to 600,000. The mesh type chosen for an exact flow simulation is tetrahedral for the blade geometry. However, the analysis time takes around one and a half hours on ANSYS FLUENT. The difference in the drag coefficients of returning and advancing blades causes a generation of force on the advancing blade. The product of this force and rotor radius generates torque.

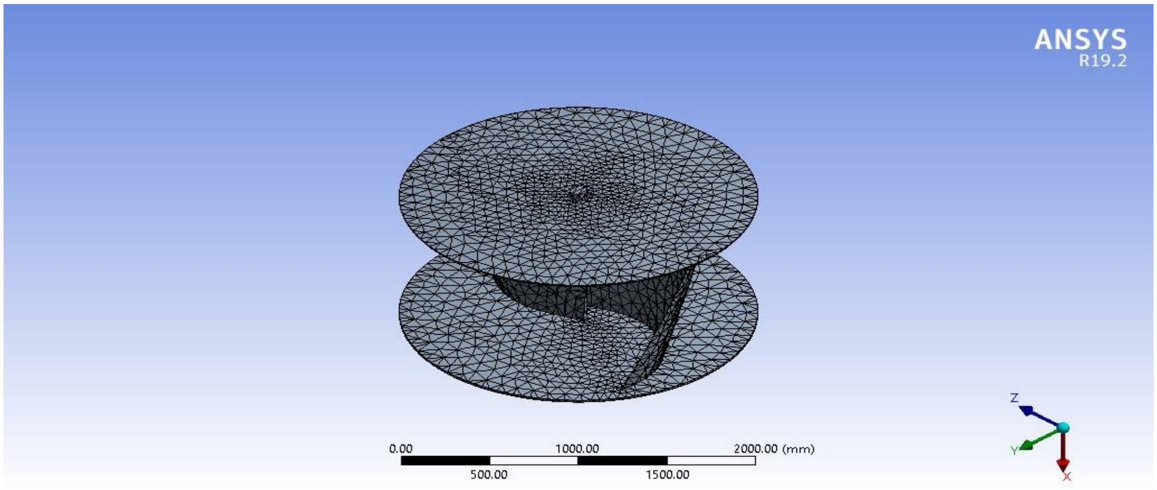

Figure 37: Mesh Generated on ANSYS

TABLE 11
Model (B4) > Static Structural (B5) > Loads

Object Name	*Fixed Support*	*Pressure*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	1 Face	10 Faces
Definition		
Type	Fixed Support	Pressure
Suppressed	No	
Define By		Normal To
Applied By		Surface Effect
Magnitude		1.2 MPa (ramped)

Figure 38: Boundary Conditions

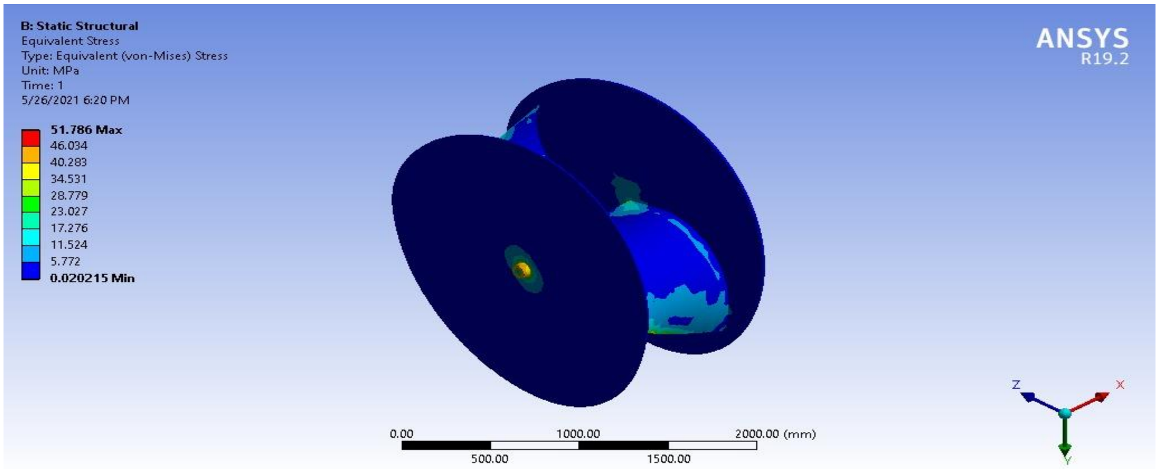

Figure 39: Von Mises Stress

The stress analysis performed gave the von Mises stresses acting on the rotor. The maximum stress acted on the shaft ends and had a value of about 42 MPa.

4.5 Portability

One of our primary deliverables was ensuring that the turbine was easier to move from one location to another. The model was designed such that it would be easy to install and remove if required and would not require the help of specialists. The blades, endplates, shaft, nuts and bolts are all packaged separately and can be joined on site. There are more than 2 ways of assembling the parts according to the power requirement. The blades are attached to the shaft with the help of nut and bolts and the outer frame is fixed to the bed of the river. The end plates have grooves built in to them where the blades can be positioned before fixing to the shaft.

If both stages of the turbine are assembled, the power output is 1000W. Similarly, if only one stage is assembled, the power output lowers to 500W. Therefore, the size and assembly can be adjusted according to the power requirements.

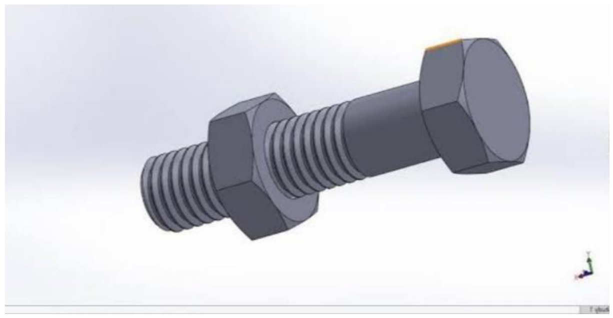

Figure 40: Nut and Bolt

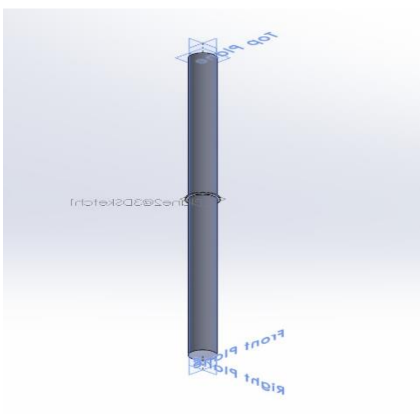

Figure 41: Shaft

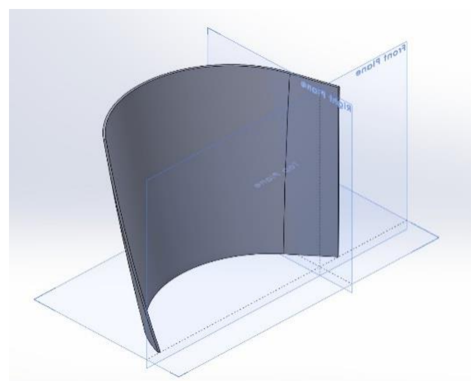

Figure 42: Blade

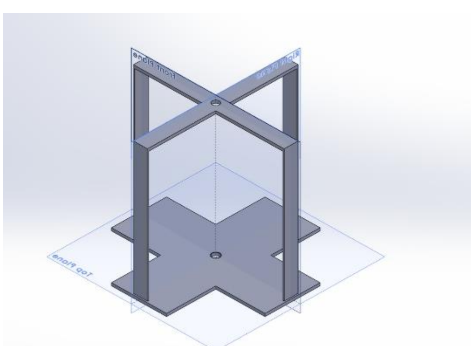

Figure 43: Frame

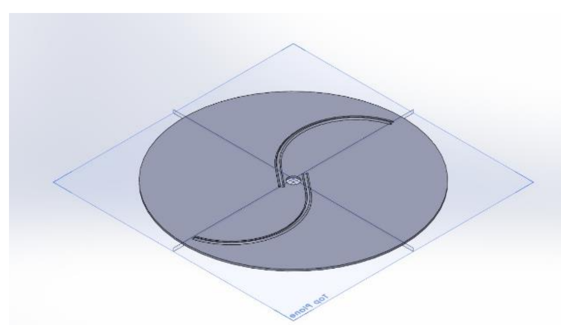

Figure 44: End Plate

CHAPTER 5: CONCLUSION AND RECOMMENDATION

5.1 Conclusion

To improve the performance of low-head hydrokinetic water turbines, numerous studies are carried out. As a result, the best possible parameters are chosen for the optimum performance of the devised model. The tests performed on SolidWorks and ANSYS fluent are very similar to the trend that already exists in the literature.

In light of the results as mentioned above, the following can be concluded;

5.1.1 Material

Material selection was done on the basis of a few factors. Firstly, the material must be capable of bearing the stresses exerted on the turbine. Furthermore, it must be readily available or easy to manufacture and hence not costly. Finally, the density of the material must be low in order to have a lightweight turbine assembly and the material must be waterproof and corrosion resistant.

The previous conventional model had chosen aluminum 6061 as their material of choice. Although this material is a suitable option, its properties exceed our requirements. It is also costly and has a high density. For our design, we have decided to choose a polymer matrix composite material [31]. Since the blades and endplates are not to be welded, this eliminated the constraint of choosing a weldable material. Moreover, this material is much lighter and can also bear the stresses exerted on our turbine. The table given below shows some properties of Aluminum 6061 which the previous model used and polymer matrix composite which we'll be using. The density of aluminum 6061 is more than twice than that of the polymer matrix composite. Although tensile and shear strengths of the composite are lower than that of aluminum 6061, it suits our application.

Table 5: Comparison between Aluminum 6061 and PMC

PROPERTIES	ALUMINIUM 6061	POLYMER MATRIX COMPOSITES
Tensile Strength	270 MPa	145 MPa
Young's Modulus	69 GPa	45 GPa
Density	2.7 g/cm^3	1.15-1.2 g/cm^3
Shear Strength	207 MPa	70 MPa

5.1.2 Performance

After implementation of the optimized blade design by using the S1048 section profile rotor in a simple savonius turbine following points are concluded for the efficiency and performance of the turbine:

- The optimized savonius turbine shows a more significant pressure difference between the concave and convex sides of the blades; hence it provides enormous torque.
- We also increased the dimensions of the turbine to incorporate the more significant power generation. Changes were made to the aspect ratio as well. The power coefficient changes depending on a few factors, primarily TSR and overlap ratio, and using overlap ratio increase the performance coefficient by 10%.
- The power coefficient of the S1048 section rotor is increased due to high-speed tip vortices.
- 2-Stages yield more power than the single and triple stages.
- End plates increase output power.

5.2 Future work

A full-scale model of the proposed rotor (S1048 section rotor) could be manufactured and tested experimentally in a water tunnel for future work. Furthermore, the upscaled model of the turbine that we designed could be used in real-time solutions. The countless advantages could include the electricity supply to the small villages alongside the river streams, use under bridges for free street lights, a restaurant near a river for its electricity, etc. Thus, there are countless possibilities for the future that lies ahead.

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