Design and Fabricate a Lab Scale Cross Flow Water Turbine

by

Beh Sea Min

Dissertation submitted in partial fulfilment of
the requirements for the
Bachelor of Engineering (Hons)
(Mechanical Engineering)

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CERTIFICATION OF APPROVAL

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Approved by,

(AP. Dr. Chalilullah Rangkuti) 05/12/08

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

DECEMBER 2008

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

Beh

BEH SEA MIN

ABSTRACT

This final report had fully describes the approaches used to begin my research and fabrication of a cross flow water turbine. The report had described the whole process of my project as well. These cross flow water turbine must be capable of generating power as an alternative to fossil fuel and the 2 existing water turbines in the lab. Theoretical values do show reasonable values for the cross flow water turbine. The success of this project had lead to alternate power generator for the lab users. The cross flow water turbine had being well fabricated, but yet to be mounted and installed in the lab due to some issues, which had being further discussed in the report as well. The scope of this project had covered both the design and fabrication of the water turbine. The technique or method used to design and fabricate the cross flow water turbine had being discussed in detail. The project background shows the introduction, problem statement, objectives, the relevancy and feasibility of the project. The literature reviews are mainly based on related references and the researched are being studied. The methodology involves the process of designing, develop the water turbine. The discussion had shows the design which includes material and fabrication studies, schematic and detail drawing. The picture of the well made cross flow water turbine was being shown as well. No experimental data are collected due to the installation matter. Only theoretical results was be discussed and analyzed. The conclusion had shown the objectives fulfilled and on-going works. Recommendations are made for further works and future references.

Acknowledgement

It is a requirement for an UTP engineering student to complete an FYP Project for duration of 2 semesters in order to graduate. From the program, the student will develop skills in work ethics, communication, management, and also to expose the students to the world of work so that they can relate theoretical knowledge with application in industry.

Here I would like to express my outmost gratitude to these respective groups and individuals for making my internship a memorable one:

- My supervisor AP. Dr. Chalilullah Rangkuti, who give me a very full support throughout the Final Year Project period.
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- Last but not least, to all those who have assisted me throughout the whole Project duration, family and friends, thanks for making it all possible.

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Nomenclature (English)

A = area

B = width

C_D = Drag Coefficient

C_L = Lift Coefficient

C = absolute velocity for turbine

d = diameter

D = drag force

E = modulus of elasticity

f = friction factor

g = acceleration due to gravitational

H = water head

L = lift force

M = total mass of moment

I = second moment of momentum, mass moment of inertia

N = rotational speed

N_s= specific speed

P = pressure

Q = Volumetric flow rate

R = Universal gas constant

R = Outer radius

Re = Reynolds number

R = inner radius

T = Time

CHAPTER 1

INTRODUCTION

1.1 Background of Study

As we all know, world number 1 power generator is fossil fuel power generator. Fossil fuel power generator is simple and easy to be used. However, there are some drawbacks on the fossil fuel power generator. The fossil fuel generators are mainly the steam and gas turbine which require an amount of time before can be started. As for water turbine, the electricity can be generated almost instantly the water turbine is turned on. The emissions of the burning of the fossil fuel are harmful to the environment as well. The fossil fuel is not a renewable source as well. It would run out as time passed but this will never happened to the water turbine. If we are manage to have an alternative power generator, it would save us a lot. Cross flow water turbine is one of the alternate options. Cross flow water turbine might be low in power efficiency; however it can be operated almost the whole year except being shut down for maintenance.

The cross flow water turbines are impulse turbines and were developed in the 19th century. In the cross flow water turbines the water, in the form of a sheet or jet, is directed into the blades tangentially at about mid way on one side. The forms of the water are dependent on the environment. As for waterfall, the water is in the form of sheet but the water is in the form of jet if it is from the pump in lab. The flow of water "crosses" through the empty center of the turbine and exits just below the center on the opposite side. This is dependent on the side the water flow in. Thus the water strikes blades on both sides of the runner and there is where the “cross flow” come from. It is claimed that the entry side contributes about 75% of the power extracted from the sheet of water and that the exit side contributes the remainder. But one thing is sure; there is increase in efficiency as the water leaves the turbine center. The cross flow is an impulse turbine and requires a high head to be really efficient but it will "work" on heads as low as 3 inch. All the requirements do depend on the design of the turbine.

1.2 Problem Statement

With the current higher cost of fossil fuel, it is necessary to find alternative energy to be used for generating power. Water power possibly is one alternative energy source that can be explored Currently UTP has 2 (two) lab unit on water turbine, namely (Pelton) and (Francis) water turbine. It is required to make another type of water turbine for the lab work purposes.

The current mostly used power generator is fossil fuel power generator. Except the high cost of the fossil fuel, it is also;

- Causing water and air pollution which are harmful to the Mother Nature. The burning of any fossil fuel produces carbon dioxides as gas which contributes to the green house effect and thus warming the earth. There are some leftover of the burned fossil fuel in the phase of liquid and solid which will cause pollution to the earth as well. As for green house effect, the world average daily temperature will increase and results in an uncomfortable surrounding for a living place.
- Not a renewable energy resource and are running out. Once we have burn them all, there isn't any more. Then it will be too late to find the alternative source of energy. Our consumptions of fossil fuel are doubled every 20 years since the early discovery of fossil fuel due to the petrol consumption of vehicles. There is a particular problem for oil because we also use it to make plastics and many other products. However, most of the fossil fuel is used as petrol.
- Slow start up of the steam and gas turbine. The steam and gas turbine require some times for the steam or gas to be accumulated to move the turbine which is different from the water turbine which start almost instantly.

As we can see, there are plenty of drawbacks of the fossil fuel. It is recommended for us to research on alternative energy by now.

1.3 Objectives

The project will begin with the design of a small scale cross flow water turbine to be used in the existing lab facility set up. Then the cross flow water turbine is to be developed. The cross flow water turbine must be able to competite with the existing water turbines in the lab in terms of efficiency:

- Design a Cross Flow Water Turbine
- Fabricate The Designed Cross Flow Water Turbine

1.4 Scope of Studies

The main part of the project will be designing the cross flow water turbine and develop the cross flow water turbine in lab scale. Apart from this scope of study, external knowledge beyond my discipline must be employed as well. The design will cover the material in used and the manufacturing and machining process in used. The scope of study included the research on the cross flow water turbine, derive theory and design the cross flow water turbine. The drawing will be in detail drawing and assembly drawing. The methodology flow chart will be constructed in detail. The relevant knowledge applied in this scope of study will be mainly about turbo machinery and turbine.

CHAPTER 2

LITERATURE REVIEW

2.1 Definition

Cross Flow Water turbine is a kind of turbo machines. The basic concepts of them are almost alike. By definition, turbo machine is devices that extract energy or impacts energy to a continuously flowing stream of fluid which can be either liquid or gas depend on the kind of machine. The stream of fluid is extracted or impacted by the dynamic action of one or more rotating blade rows. The first word "turbo" simply means spin or whirl. It's a kind of movement of the machine which implying that turbo machines move in some kind of rotation, [1].

If the machine impacts energy to the fluid, it is a pump in general. If its extracts energy from fluid, it is generally a type of turbine.

2.2 Classification of Turbo Machines

Turbo machines are generally categorized as Shrouded or Unshrouded turbo machines. If the rotating member is enclosed in a casing or shrouded in such a way that the working fluid cannot be diverted to flow around the edge of impeller, it is known as a shrouded turbo machines. Examples of the shrouded turbo machines are turbines and pumps. If the fluid flows around the edges of the impeller which is not shrouded or in casing, then it is known as a unshrouded turbo machine. Example of the unshrouded turbo machines are wind mill, aero-generator and aircraft propellers. Cross flow water turbine by classification is a shrouded turbo machine, [1].

Turbo machines may be categorized into one of the two classes depending whether the work is done by the fluid on the rotating member (like hydraulic turbine or gas turbine) or work is done by the rotating member on the fluid (like pumps and compressors). For cross flow water turbine, the work is done by the fluid on the rotating member.

The turbo machine can also be categorized by the energy transferred from or to the rotating member, which are usually fixed on the shaft. In the work absorbing machines the fluid pressure of fluid head (for hydraulic turbine) or the enthalpy (for compressible flow machines) increased from inlet to outlet. But in work delivering machines the fluid pressure or enthalpy is decreased from the inlet to the outlet. Cross flow water turbine is a work absorbing machine.

The product change in head or enthalpy, and the mass flow rate of the fluid through the machine, represents by the energy absorbed by or extracted from the rotating member of the machine. In turbo machines, the energy transferred is accomplished by changing the angular momentum of the fluid and so the shapes of the rotating members differ from one type to another in terms of efficiency, [2].

Turbo machines can be categorized based on the direction of the fluid across the rotating member. If the fluid is axial, the turbo machine is called an axial flow machine. If the flow is radial, the turbo machine is called a radial flow or centrifugal machine. If the flow of the fluid is partially axial and partially radial, the machine is known as mixed flow machine. Unlike most water turbines or turbo machines, which have axial or radial flows, in a cross flow water turbine the water passes through the turbine transversely, or across the rotating member which is the turbine blades. As with a waterwheel, the water is admitted at the turbine's edge to run the turbine. After passing the runner, it leaves on the opposite side depending on the design and center of gravity of the design. Going through the runner twice provides additional efficiency, [3].

2.3 Classification of the Water Turbine

Water turbine converts the water potential energy into electrical energy. The main types of turbine used in this century are the impulse and the reaction turbines. The predominant of the impulse turbine is the Pelton water turbine. Reaction turbines are of two types which are radial or mixed flow and axial flow, [10].

The important classifications of hydraulic turbines are

Type of energy at the inlet

a. Impulse Turbine. Energy available at the turbine inlet is only kinetic energy and the pressure is atmospheric from the turbine inlet to the turbine outlet. Cross flow water turbine is an Impulse Turbine.
b. Reaction Turbine. Energy available at the turbine inlet is both kinetic energy and pressure energy.

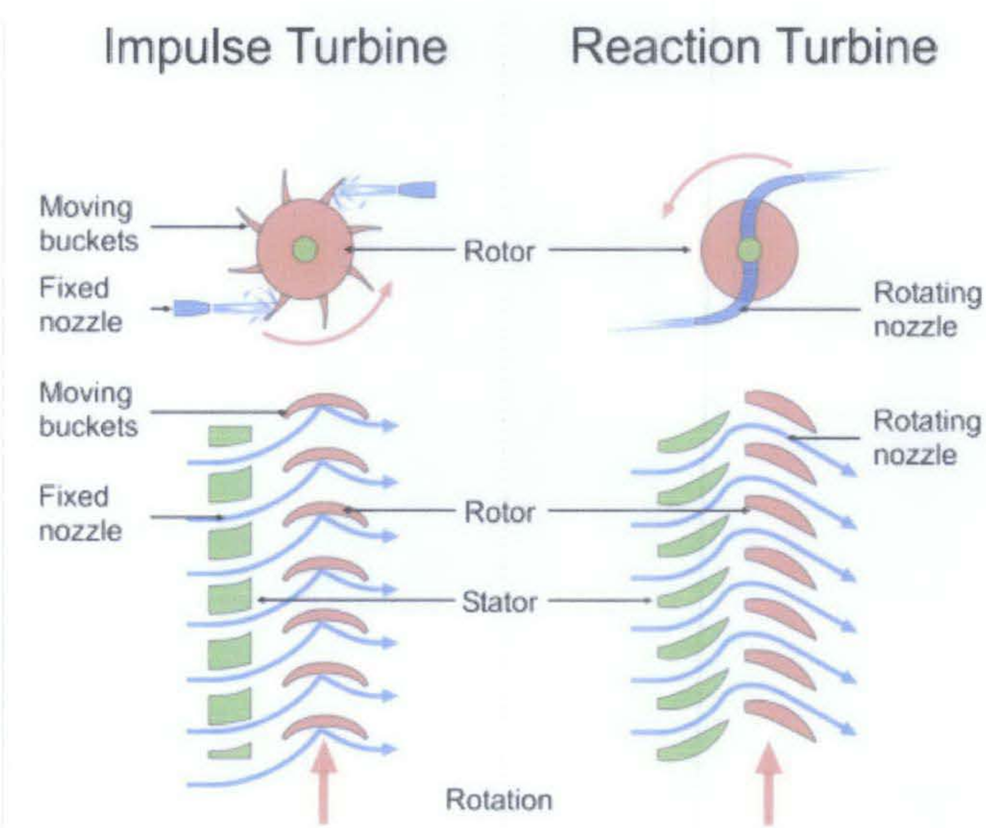

Figure 1: Differences between Impulse and Reaction Turbines

The characteristic of the impulse turbine is seen that the peak values of efficiency do not vary much for various gate openings. It is seen that the peak power occurs at the same speed irrespective of the nozzle settings. This is due to the nozzle velocity remaining constant in magnitude direction as the flow rate changes, giving an optimum value of U/C at a fixed speed.

2.4 Requirement of a Cross Flow Water Turbine's Design

The turbine consists of a cylindrical water wheel or runner with a horizontal shaft, composed of numerous blades (up to 37), arranged radially and tangentially. The blades' edges are sharpened reduce resistance to the flow of water. A blade is made in a part-circular cross-section (pipe cut over its whole length). The ends of the blades are welded to disks to form a cage like a hamster cage; instead of the bars, the turbine has trough-shaped steel blades, [2].

The water flows first from the outside of the turbine to its inside. The regulating unit, shaped like a vane or tongue, varies the cross-section of the flow. The water jet is directed towards the cylindrical runner by a fixed nozzle. The water enters the runner at an angle of about 45 degrees, transmitting some of the water's kinetic energy to the active cylindrical blades.

The regulating device controls the flow based on the power needed, and the available water. The ratio is that (0–100%) of the water is admitted to 0-100%×30/4 blades. Water admission is to the two nozzles is throttled by two shaped guide vanes. These divide and direct the flow so that the water enters the runner smoothly for any width of opening. The guide vanes should seal to the edges of the turbine casing so that when the water is low, they can shut off the water supply. The guide vanes therefore act as the valves between the penstock and turbine. Both guide vanes can be set by control levers, to which an automatic or manual control may be connected.

The turbine geometry (nozzle-runner-shaft) assures that the water jet is effective. The water acts on the runner twice, but most of the power is transferred on the first pass, when the water enters the runner. Only ⅓ of the power is transferred to the runner when the water is leaving the turbine.

The water flows through the blade channels in two directions: outside to inside, and inside to outside. Most turbines are run with two jets; arranged so two water jets in the runner will not affect each other. It is, however, essential that the turbine, head and turbine speed are harmonized.

The cross-flow turbine is of the impulse type, so the pressure remains constant at the runner.

2.5 Advantages of Cross Flow Water Turbines

The peak efficiency of a cross flow turbine is somewhat less than a Kaplan, Francis or Pelton turbine. However, the cross flow turbine has a flat efficiency curve under varying load. With a split runner and turbine chamber, the turbine maintains its efficiency while the flow and load vary from 1/6 to the maximum, [2].

Since it has a low price, and good regulation, cross flow turbines are mostly used in mini and micro hydropower units less than 2000 kW and with heads less than 200 m. Particularly with small run-of-the-river plants, the flat efficiency curve yields better annual performance than other turbine systems, as small rivers' water is usually lower in some months. The efficiency of a turbine determines whether electricity is produced during the periods when rivers have low heads. If the turbines used have high peak efficiencies, but behave poorly at partial load, less annual performance is obtained than with turbines that have a flat efficiency curve.

2.6 Basic Laws

According to the Newton's Second Law of Motion, the sum of all the forces acting on a controlled volume in a particular direction is equal to the rate of change of momentum of the fluid across the controlled volume in the same direction, [4].

Valan (2001) says that in turbo machines, the impellers are rotating and the power output is expressed as the product of torque and angular velocity due to the rotation and so angular momentum is the main parameter. Consider a fluid particle in detail moving across a controlled volume from one point (A) to another point (B). The fluid particles travel from point A to point B while simultaneously moving from the radius of point A (r_1) to the radius of point B (r_2). If Va_1 and Va_2 are components of absolute velocities in the tangential direction, then the sum of all the torques acting on the system is equal to the rate of change of angular momentum. Mathematically,

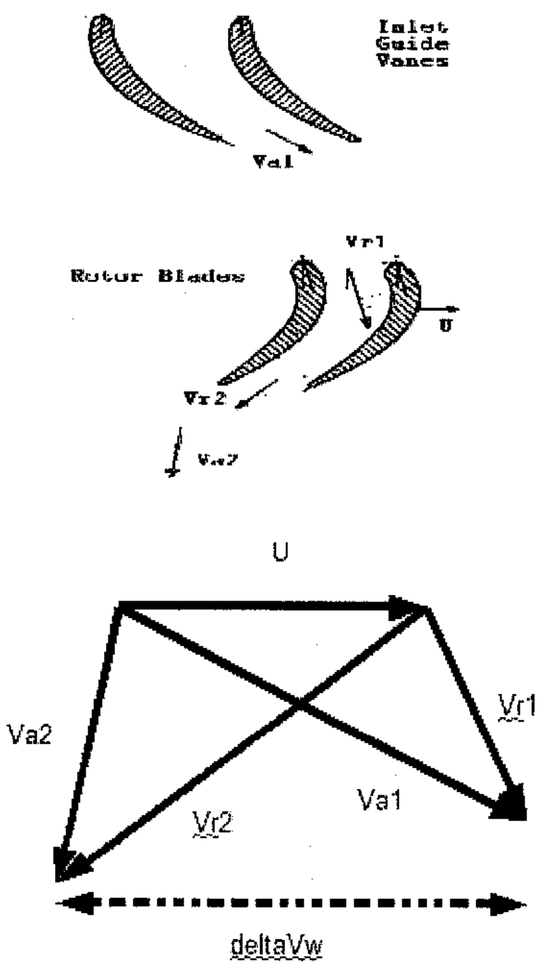

Figure 2: Velocities Triangle Diagram

$$\sum \tau = m\,(r_2 Va_2 - r_1 Va_1)$$

If the machine revolves with angular velocity ω, the power (in watt) is

$$\sum \tau \omega = m(\omega r_2 Va_2 - \omega r_1 Va_1)$$

Since $\omega r = U$

Where U is the impeller tangential velocity,

So, $W = m(U_2Va_2 - U_1Va_1)$

This equation is called the general form of Euler's Equation. The specific form of the Euler's Equation for turbine is

$$W = m(U_2Va_2 - Va_1C_1) > 0,$$

where τ = torque

$\sum$ = summation of all

m = mass

r = radius

V=velocity

ω = angular velocity

2.7 Draft Tube

The available head in the lab is only 15m which is more or less to be considered small in real world application. The loss in water head will be smaller as the water head is larger. The efficiency can be further increased by placing the turbine above the tail-race level and leading the water from the turbine outlet to the tail race by a tube such that the water reaches atmospheric pressure only at the tail race. The piping system will gradually increase the area which is used for discharging water from the turbine exit to the tail race is known as draft tube, [1].

The advantages of draft tube are

1. The pressure at the runner exit is below the atmospheric pressure due to the partial-vacuum situation and the turbine operates efficiently as if it is placed at the tail-race.
2. The kinetic energy ($C^2 / 2g$) of water at the turbine runner outlet is converted into useful pressure energy which will further increase the different in pressure.
3. The turbine may be inspected easily and properly as it is placed above the tail-race.

There are four types of draft tubes, used in practice, depending upon the flow conditions and the height on the turbine above the tail-race which are

1. Conical draft tube. A straight conical type stretching from the turbine to the tail-race. The simplest draft tube.
2. Bell mouthed or moody spreading tube. A straight draft tube except that it is bee-shaped. This type of draft tube has an advantage that it can allow flow with whirl component to occur with very small losses at the turbine exit. In any turbine, the exit absolute velocity usually has a whirl component especially at part load operation; the bell-shaped draft tube may be preferred where the operation is at part load operation for long periods of time.
3. Simple elbow or bent tube. The bent draft tube is used when the turbine must be located very close to or below the tail race level for some reason. However, the efficiency of the simple elbow or bent tube is not as high as other draft tube.
4. Elbow draft tube with circular inlet and rectangular outlet. Exactly the same with simple elbow or bent tube except for the inlet and outlet shapes.

The height of the draft tube is governed by two factors which are

1. Cavitation, which requires that the pressure at the turbine exit or draft tube entry should not be less than one-third of the atmospheric pressure.
2. Separation, which occurred if the draft tube has too large an angle of flare. In practice, the angle should not be more than 10 degree to prevent separation.

Draft tube efficiencies range generally from 0.7 to 0.9 for the first and second type but only 0.6 to 0.85 for the third and forth type.

Let's assume the turbine exit is at a height H_s above the tail race level. Let the subscript 1 denote the conditions at the rotor exit and subscript2 denote the conditions at the outlet.

Valan (2001) says that applying Bernoulli's equation to inlet and outlet of the draft tube taking the draft tube, exit as the datum line we get,

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + (H_s + X) = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + 0 + h_d$$

Where x is the distance of bottom of draft tube from the tail-race and h_d is head loss in the draft tube and

$$\frac{P_2}{\rho g} = \text{Atmospheric Pressure } + X$$

$$\frac{P_2}{\rho g} = \frac{P_a}{\rho g} + X$$

Substituting the value of the left hand side of the equation in the previous equation yield

$$\frac{P_1}{\rho g} = \frac{P_a}{\rho g} - H_s - \left(\frac{V_1^2}{2g} - \frac{V_2^2}{2g} - h_d\right)$$

The above equation shows that the inlet pressure at the draft tube inlet is less than the atmospheric pressure.

It is defined that the ratio of the actual conversion of kinetic energy into pressure energy in the draft tube to the kinetic energy available at the draft tube inlet, so

$$\eta dt = \frac{(V_1^2 - V_2^2) - 2gh_d}{V_1^2}$$

where P = pressure

ρ = density

g = acceleration due to gravity

V= velocity

H = water head

η = efficiency

2.8 Cavitations in Turbines

Agarwal (1997) says that the turbine cavitation occurs on the suction surfaces of the blades, and at the runner outlet where the static pressure is a minimum and the absolute velocity is high. It should be avoided although it has little effect on the performance of the turbine since it occurs after the runner.

Applying the energy equation between the runner outlet and tail-race yields

$$V_2^2 - V_3^2/2g = \frac{(P_3 - P_2)}{\rho g} - Z_2 + Z_3 + h_d$$

Putting $Z_3 = 0$, and as the outlet velocity V_2 increases, P2 decreases and has its lowest value when the vapor P_r is reached. At this pressure, cavitation occurs (begins) and hence putting P_3 equal to P_{atm} and P_2 equal to P_{vap} yield

$$\frac{V_2^2 - V_3^2}{2g} - h_d = \frac{P_{atm} - P_{vap}}{\rho g} - Z_2$$

Dividing this equation by the net head across the turbine gives the "Thoma cavitation parameter" for the turbine,

$$\sigma = \left[\frac{P_{atm} - P_{vap}}{\rho g}\right] / H$$

which will yield

$$\sigma = \frac{NPSH}{H}$$

where V = velocity

g = acceleration due to gravity

P = pressure

P = density

h = water head

NPSH = Net Positive Suction Head

The critical value of NPSH at which cavitation occurs is determined from the testing on a model or full size machine in which P_2 decreases until the minimum value at which cavitation begins or the efficiency suddenly decreases is found. Knowing Z_2 and H will make computing the sigma value easier, which is the value belowσ, as shown by the equation.

2.9 Derivation of the Turbine Specific Speed

Valna (2001) says that the power developed by any turbine in terms of overall efficiency is given by

$$P = \eta\, g\, d\, H\, Q$$

or P is proportional to Q H because η, g and d are constants.

The absolute velocity, U, tangential velocity, C, and head, H, on turbine are related as the absolute velocity is proportional to the tangential velocity and proportional to the square root of water head.- (1)

The tangential velocity, U is given by

$$U = \frac{\pi DN}{60}$$

or U is proportional to D N.-(2)

From the previous two statements, (1) and (2), we have D is proportional to square root of H divided by N. - (3)

The discharge through the turbine is given by

$$Q = \text{Area x Velocity}$$
$$= A \times C$$

The area is proportional to the width, B and the diameter, D. Since the width, B is proportional to the diameter, D; area is proportional to square of diameter.

From previous equations and statements

$$A \, \alpha \, \frac{H}{N^2}$$

$$C \, \alpha \, \sqrt{H}$$

so $$Q \, \alpha \, \frac{H^{\frac{3}{2}}}{N^2}$$

Substituting for Q in the power equation, we get

$$P \, \alpha \left(\frac{H^{\frac{3}{2}}}{N^2}\right).H$$

or $$P \, \alpha \left(\frac{H^{\frac{5}{2}}}{N^2}\right)$$

or $$P \, \alpha \left(\frac{H^{\frac{5}{2}}}{N^2}\right)(\text{constant})$$

The constant is called the constant of proportionality. According to the definition of specific speed, $N = N_s$ when $P = 1\text{kW}$ and $H = 1\text{m}$ makes the constant $= N_s^2$.

so $$P \, \alpha \left(\frac{H^{\frac{5}{2}}}{N^2}\right).N_s^2$$

or $$N_s^2 = \frac{N^2 .P}{H^{\frac{5}{2}}}$$

The turbine specific speed, N_s is

$$N_S = \frac{N\sqrt{P}}{H^{\frac{5}{4}}}$$

The specific speed of a turbine is defined as the speed at which the turbine develops unit power when working under units head where N is the speed in rpm, H is the head in meter and P is the power in kilowatts.

2.10 Summarization of Literature Reviews

The basic definition of the Cross Flow Water Turbine had given us the basic understanding of a Cross Flow Water Turbine. After all the definition, we know that Cross Flow Water Turbine is:

- A shrouded turbo machine that extract energy from flowing fluid. The work is done by the rotating members which are the blades of the runner.
- An impulse hydraulic turbine.

We also know that the work done by the blades from the basic laws of the turbine and the Euler's Turbine Equation. The information of the draft tubes are useful if a draft tube is going to be designed for the turbine in the future.

CHAPTER 3

METHODOLOGY

3.1 Project Activities

The project was divided into 2 stages and was work sequentially. The 1st half of the project was focused on the design and the 2nd half was on fabrication.

3.1.1 Design of the cross flow water turbines

A lab scale of the cross flow water turbine had been designed as the begun of the project. The component of the cross flow water turbine will be picked accordingly from anything available in the market. This design had been modified from time to time to make sure the cross flow water turbine meet its requirement. A few technologies had been adapted to simulate the cross flow water turbine.

3.1.2 Develop of the cross flow water turbines

The design had been developed in a lab scale. The testing can't be fulfilled due to the time constraint and the time consuming procedure to mount and install the turbine.

3.2 TOOLS and components

To design the cross flow water turbines, designing engineering software will be needed. Examples of the software are AutoCAD and CATIA. The simulation can be done by MATLAB.

3.3 Project Flow Chart

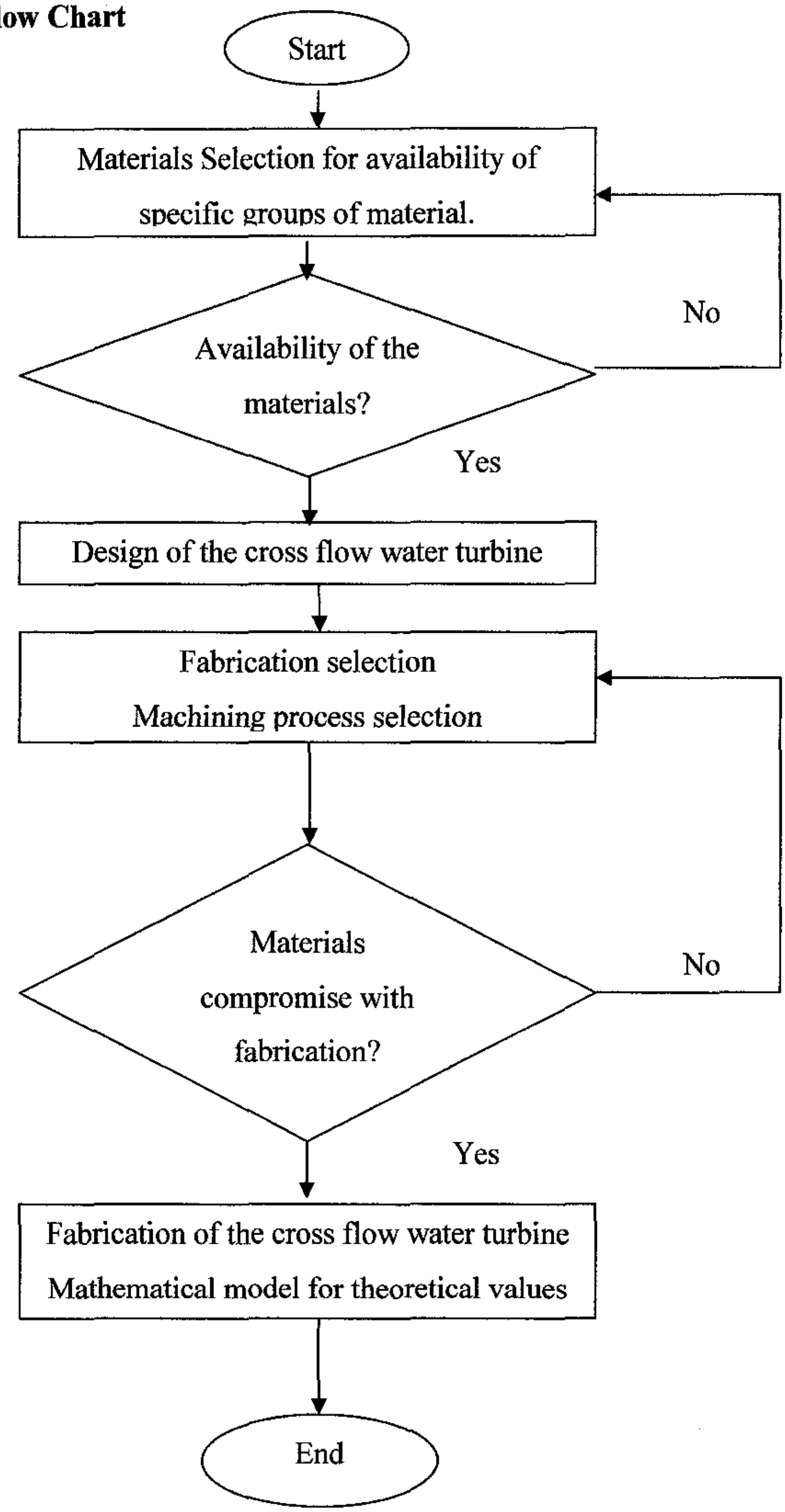

Figure 3: Flow Chart of the process of the design and Fabrication of Cross Flow Water Turbine

- The project started by initial findings on information about the Cross Flow Water Turbine. This step is indeed necessary to make sure the turbine produced is indeed a Cross Flow Water Turbine.

- The Research Study is the study on the basic laws for turbine designing.

- Materials Selection for the turbine was done to select the materials available in the market and cope with the design capability in order to make the design possible for fabrication and power performance.

- From the material, the design was made parts by parts. Then the fabrication methods were chosen to cope with the materials.

CHAPTER 4 (PART A)

DESIGN

4.1 Design Specification

The design was begun with the parts by parts of the water turbine separately being modeled. Listed below are the parts designed.

- Shaft
- Blade/Runner
- Bearing
- Housing
- Tank (available)
- Pump (available)

Both the tank and pump are available in the lab. So the design of the rest of the parts must cope with the existing design of the pump and water tank. The water tank, the existing turbine, the piping, the pump and the valve controller for tank are shown.

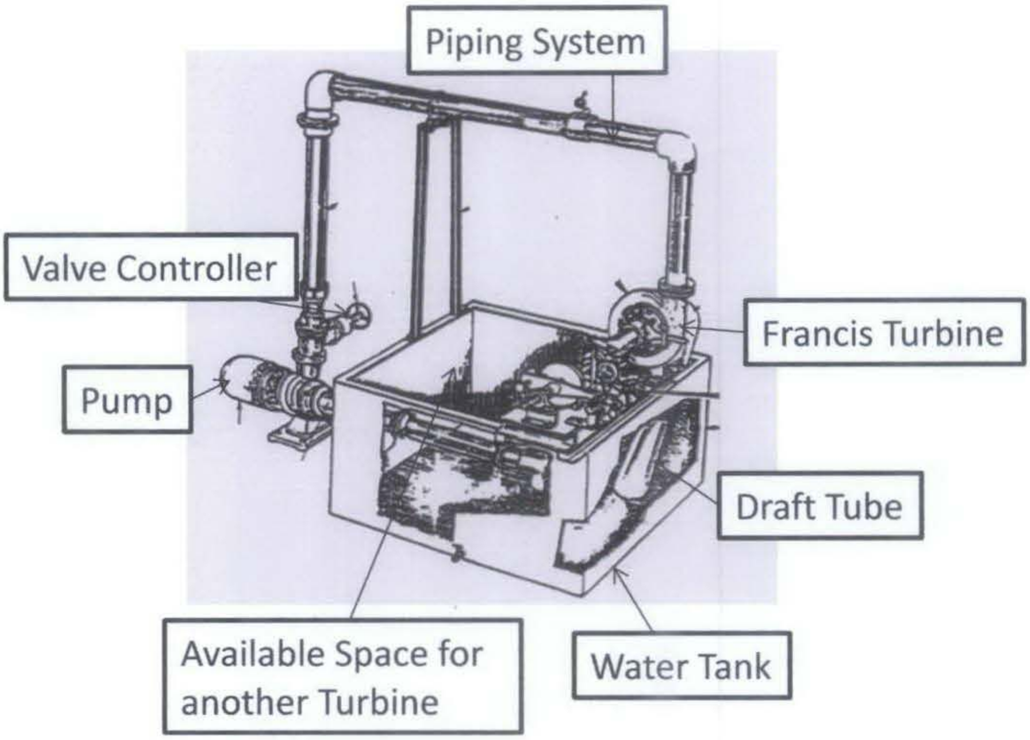

Figure 4: Existing Equipment with Existing Turbine

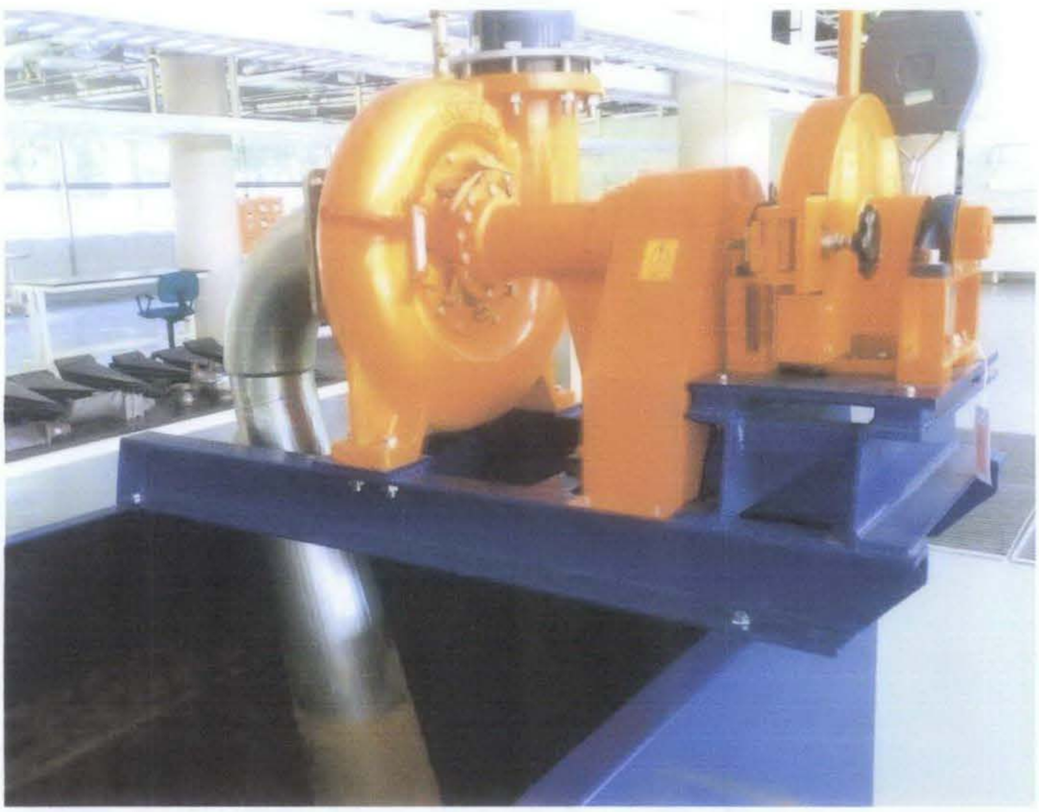

Figure 5: Existing Turbine

4.2 Designed Cross Flow Water Turbine

The preliminary design of the system is represented by the drawing below for the system. The whole system starts with the water from the water tank to be transferred to the cross flow water turbine via the pump. Then, we will determine the power gained in the cross flow water turbine to determine the cross flow water turbine efficiency. The whole system is represented by the following mathematical modeling. Mathematical modeling is a set of mathematical representation to study the behavior of the system under different conditions. The water tank is approximately 1.2m x 1.5m x 0.8m in size. The pump can deliver an output of 15m water head and $0.05m^3/s$ water flow rate.

4.3 Water Flow from the Water Tank

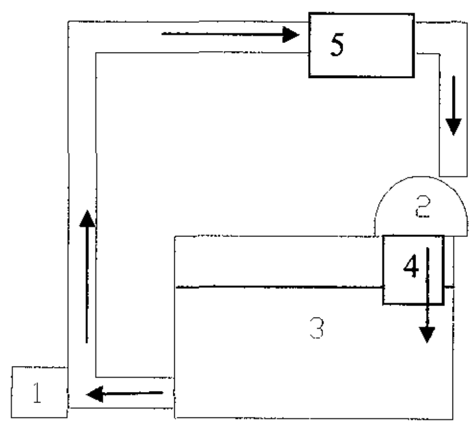

Figure 6: Equipment Set Up

1	Pump
2	Turbine
3	Water Tank
4	Draft tube
5	Water Flow Rate Indicator

The pump will pumps the water from the water tank to the turbine through the piping route. The water will go through the turbine and back to the water tank.

4.4 Assumptions made

- The whole turbine must be able to fit in the existing tank. The water tank is approximately 1.2m x 1.5m x 0.8m in size.
- The performance is based on the pumps performance. The pump can deliver an output of 15m water head and $0.05m^3/s$ water flow rate.
- The performance is in steady state
- Thermal properties of materials are independent of temperature
- The water is flowing over the piping at a constant rate
- Uniform flow of water in the pipe
- Friction is negligible
- Water density at room temperature is $997kg/m^3$
- The value of g is $9.807m/s^2$
- All calculation are rounded to the nearest 4 significant figures

4.5 Shaft Design

Figure 7: Shaft Design

The pump available in the lab is able to produce a maximum water flow rate of $200m^3/hr$ or approximately $0.05m^3/s$ and a maximum head of 15m. The maximum theoretical power developed by the turbine is, [7]

$$P = g\,d\,H\,Q$$

P	=	Power
G	=	Gravitational Force
H	=	Water Head
Q	=	Water Flow Rate
D	=	Density

$$P = 9.81m^2/s \times 997kg/m^3 \times (15m) \times (0.05m^3/s)$$

$$P = \underline{7335 Watt}$$

The Equation is the general equation for calculating turbine's power.

If we assume a 100% efficiency ideal turbine, the turbine power is equal to the shaft's mechanical power. The rotational speed of the turbine is assumed to be 1000rpm – 2000rpm. Taking the lower end of the rotational speed, we have the rotational speed equal to 1000rpm. The equation for shaft's mechanical power is

$$P = 2\,\pi\,N\,T\,/\,60$$

P	=	Power
N	=	Rotational Speed
T	=	Torque

Rearrange the equation to solve the torque, we get

$$T = 60\,P / 2\,\pi\,N$$

$$T = 60\,(7335\text{Watt}) / 2\,\pi\,(1000\text{rpm})$$

$$T = 70.04\text{Nm}$$

From the torque, we can calculate the required minimum diameter of the shaft. The equation that relate the torque and shaft diameter is

$$T = \pi \text{ x Shear Stress x } D^3 / 16$$

T	=	Torque
D	=	minimum diameter required

Rearrange the equation to solve for the diameter we get

$$D = (16\,T / \pi \text{ x Shear Stress})^{1/3}$$

The shear stress of mild steel (low carbon alloy steel with 500MPa Grade) is 500MPa which is 500MN/m^2. So

$$D = (16 \text{ x } 70.04\text{Nm} / \pi \text{ x } 500\text{x}10^6\text{N/m}^2)^{1/3} \text{ x safety factor}$$

$$D = 8.935 \text{ x } 10^{-3}\text{m x } 2$$

$$\underline{D = 17.87 \text{ mm}}$$

This is the minimum thickness required for the shaft. We multiply in the safety factor of 2 [11] and we will get roughly 20mm. So the design was based on this specification. The minimum diameter of the shaft designed is 24mm and thus is appropriate.

The shaft will be extended for the attachment of the measurement device in the future. The simple breaking device is recommended for power measurement.

Table 1: Shaft Dimensions

Dimension	Value
Length	655mm
Minimum Diameter	24mm
Intermediate Diameter	30mm
Maximum Diameter	40mm

4.6 Runner Design

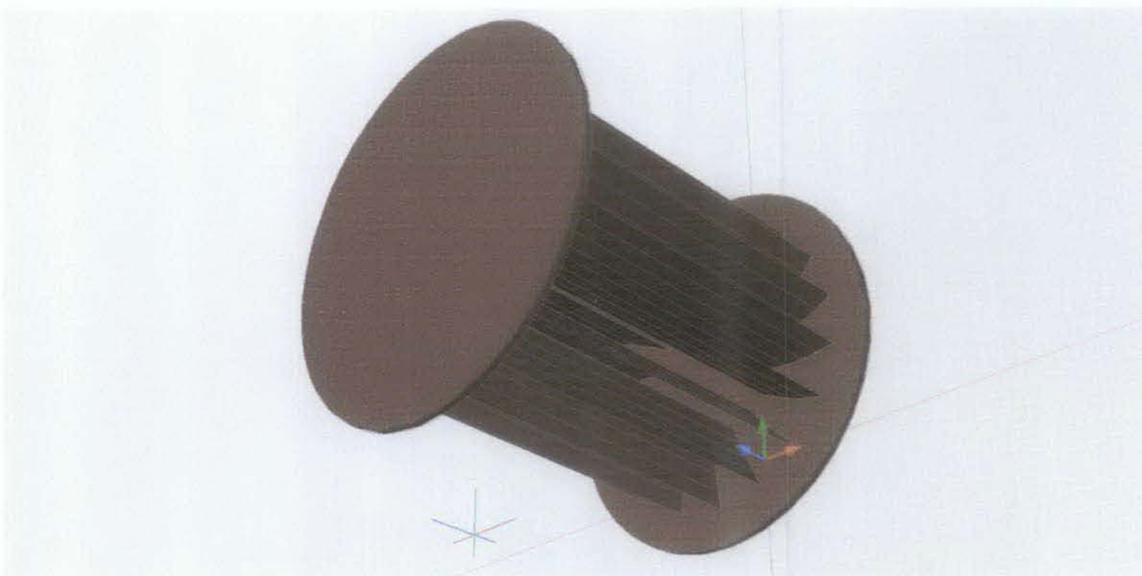

Figure 8: Runner Overview

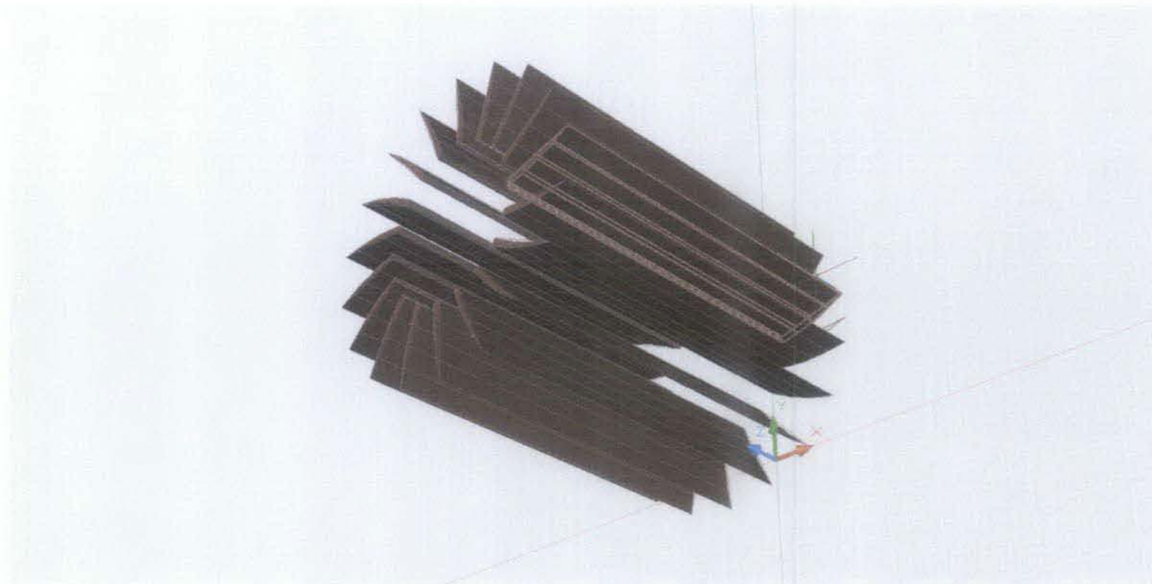

Figure 9: Blade Overview

To begin the design, we have accumulated a few data from the pump and a few efficiency data.

From Pump:

Water flow rate, $Q = 0.05m^3/s$

Water Head, $H = 15m$

Pump Supply = three phase 60Hz

Efficiency Assumptions

Volumetric Efficiency, $N_v = 0.96$

Overall Efficiency, $N_o = 0.3$

Range of impeller speed = 1000 – 2000rpm

Other initial Assumptions

We assumed that impeller has 18 blades. This can be changed later if desired.

An impeller ID will be assumed that lies between d_o and $d_{HUB;\ Once}$ these have been calculated, the value of d_1 may be determined.

A diameter ratio of 0.75 (d_1/d_2) is assumed initially.

Power into the pump shaft, [11]:

$$P_{shaft} = g\ d\ H\ Q / N_o$$
$$= 9.81 m^2/s \times 997 kg/m^3 \times (15m) \times (0.05 m^3/ s)/0.3$$
$$= 24.45 kW$$

Specific Speed is calculated by:

$$N_s = N\ (Q)^{0.5} / (H)^{0.75}$$

Where;

N = rotations in a minute.

Inlet and Outlet Velocities of the impeller are calculated from:

$C_{m1} = K_{cm1} \times (2gH)^{0.5}$ and

$C_{m2} = K_{cm2} \times (2gH)^{0.5}$

where $K_{cm1} = 0.001923 \times N_s + 0.0615$ and
$K_{cm2} = 0.001805 \times N_s + 0.0948$

Table 2: A Range of Data for C_{m1} and C_{m1} from Equation

N (rpm)	N_s	k_{cm1}	K_{cm2}	C_{m1} (m/s)	C_{m2} (m/s)
1000	40	0.1384	0.1670	3.065	3.699
1100	44	0.1461	0.1743	3.235	3.859
1200	48	0.1538	0.1815	3.406	4.018
1300	52	0.1615	0.1887	3.576	4.178
1400	56	0.1692	0.1959	3.746	4.338
1500	60	0.1769	0.2031	3.917	4.498
1600	64	0.1846	0.2103	4.087	4.657
1700	68	0.1923	0.2176	4.257	4.817
1800	72	0.2000	0.2248	4.428	4.977
1900	76	0.2076	0.2320	4.598	5.137
2000	80	0.2153	0.2392	4.768	5.297

Torsional strength of mild steel, $\tau = 500MPa$

Values of d_{shaft} were calculated for the range of N considered and are given in table 3. The hub diameter on the inlet side is usually taken to be:

$$D_{hub} = 1.4 \times d_{shaft}$$

A common volumetric efficiency for centrifugal pumps is 96%. Therefore, the design Q becomes:

$$Q_{design} = Q/0.96$$

The inlet cross-sectional area is:

$$A_0 = Q_{design}/C_{m1}$$

It is usually to increase this by5%. The total inlet cross-sectional area is:

$$A_{0,design} = A_0 + A_{Hub}$$

Values of $A_{0,design}$ are given in table 3. The diameter of the inlet eye impeller is:

$$D_0 = (4\ A_{0,design}/\ \pi)^{0.5}$$

If there is forward extension of the impeller, this will be reduces. However, the extension is beyond this project and will never be discussed here. In this case we will just assume no extension. Blade velocity at inlet is:

$$U_1 = (\pi d_1 N)/60$$

Water enter the impeller freely, that is , $\alpha_0 = 90^0$, so:

$$\tan \beta_1 = c_{m1} / u_1$$

The flow angle of incident at inlet is δ_1. Thus, the flow angle becomes:

$$B'_1 = \beta_1 + \delta_1$$

The values of $A_{0,design}$, d_0, d_{hub}, c_{m2} and β_1 are given in table 3. The value of the outlet angle β_2 is

$$U_2 = c_{m2}/2 \tan \beta_2 + \{(c_{m2}/2 \tan \beta_2)^2 + (gH/\eta_h)(1 + Cp)\}^{0.5}$$

$(1 + Cp)$ is evaluated from the Pfleiderer correction for a finite number of blades

$$(1 + Cp) = 2 (\psi/z)[1/(1-(d_1/d_2))^2]$$

where $\psi = k(1 + \sin \beta_2)(d_1/d_2)$

The value of k = 1 or 1.2, depending on whether the pump has guide vane. For pump without guide vanes k = 1.2

Table 3: A Range of Data U_2 from Equation

N (rpm)	D_{shaft} (cm/s)	D_{hub} (cm/s)	Q_{design} (m^3/s)	$A_{0,design}$ (m^2)	D_0 (cm)	U_1 (m/s)	B'_1	U_2 (cm/s)
1000	4.48	6.27	0.05	0.079	31.8	7.85	24.3	15.71
1100	4.34	6.08	0.05	0.075	30.9	8.64	23.5	17.28
1200	4.21	5.89	0.05	0.071	30.2	9.42	22.9	18.85
1300	4.1	5.74	0.05	0.068	29.4	10.21	22.3	20.42
1400	4	5.60	0.05	0.065	28.7	11.00	21.8	22.00
1500	3.91	5.47	0.05	0.062	28.1	11.78	21.4	23.56
1600	3.83	5.36	0.05	0.060	27.5	12.57	21	25.13
1700	3.75	5.25	0.05	0.057	27.0	13.35	20.7	26.7
1800	3.68	5.15	0.05	0.055	26.4	14.14	20.4	28.27
1900	3.61	5.05	0.05	0.053	25.9	14.92	20.1	29.85
2000	3.55	4.97	0.05	0.051	25.5	15.71	19.9	31.42

The hydraulic efficiency is defined as:

$$\eta_h = (\eta_0)/[\eta_v x\ \eta_m]$$

where: η_0 = overall efficiency

η_v = volumetric efficiency

η_m = mechanical efficiency

Thus, the theoretical head is:

$$H_{th} = H / \eta_h$$

$u_2 = 2\ u_1$ thus, all terms is known except β_2 and been evaluated for a range of N

Table 4: A range of β_2 determined by [11]

β_2	tan β_2	Sin β_2	1 + Cp	U_2 (m/s) from 1000rpm	U_2 (m/s) from 1100rpm	U_2 (m/s) from 1200rpm
22	0.404	0.375	0.550	17.98	18.24	18.51
23	0.425	0.391	0.556	17.75	18.00	18.25
24	0.445	0.407	0.563	17.55	17.79	18.03
25	0.466	0.423	0.569	17.38	17.60	17.82
26	0.488	0.438	0.575	17.22	17.43	17.64
27	0.510	0.454	0.582	17.08	17.28	17.48
28	0.532	0.470	0.588	16.96	17.15	17.34
29	0.554	0.485	0.594	16.84	17.02	17.21
30	0.577	0.500	0.600	16.74	16.91	17.09

We need to match the value of u_2 I each table. There is a match in table 3, N =1100rpm, u_2 =17.28m/s and $\beta_{1,design\ 1} = 23.5^0$. Another match is from table 4, N =1100rpm, u_2 =17.28m/s and $\beta_{1,design\ 2}$ =27.

The inlet angel is further adjusted by means:

$$\tan \beta_{1,design\ adjusted} = \tan (\beta_{1,design\ 1})[\cos(\beta_{1,design\ 2})]/2$$
$$\tan \beta_{1,design\ adjusted} = \tan (23.5)[\cos(27)]/2$$
$$\tan \beta_{1,design\ adjusted} = 0.1937$$
$$\beta_{1,design\ adjusted} = \underline{10.96^0}$$

The adjusted inlet angle is 10.96 degree and we take the value of 10 degree to ease the calculation. Now we check the blade number, z by the Pfleiderer Equation:

$$Z = 6.5[(d_2 + d_1) / (d_2 - d_1)] \sin (\beta_1 + \beta_2)/2$$
$$= 6.5[(1+7.5) / (7-0.75)] \sin (23.5 + 27)/2$$
$$= \underline{17.55}$$

The value of z is close enough to the assumed value of 18, so there is not necessary to change.

The runner is consisting of 18 blades arranged within 2 plates. The material used for the runner will be mild steel (low carbon alloy steel with 500MPa Grade) with the lowest shear modulus around 500MPa. The maximum moment and torque applied on the shaft will be: [9]

$$(M^2 + T^2)^{1/2} = (200m^3/hr)\ (1hr/3600s)\ (1s)\ (997kg/m^3)\ (9.807m/s^2)\ (0.3m)$$

M = Moment

T = Torque

C = Required Thickness

$$(M^2 + T^2)^{1/2} = 163.0Nm$$

The required thickness is based on the maximum shear stress theory, [9]

$$C = [2\ (M^2 + T^2)^{1/2} / \pi \times t_{allow}]^{1/3} \times \text{Safety Factor}$$
$$C = [2\ (163.0/18) / \pi\ (500 \times 10^6)]^{1/3} \times 2$$
$$C = 0.004518m$$

The thickness required for the mild steel blades to withstand the load is around 0.0002259m. So we have thickness of each blade = 0.005m. Safety Factor is close to 2.

Table 5: Runner Dimensions

Dimension	Value
Length of blades	0.09m
Width of blades	0.08m
Bending degree of blades	10 degree from center of runner
Maximum thickness of blades	0.006m
Minimum Thickness of blades	0.005m
Minimum range of blade from blades to center of runner	0.07m
Diameter of disks	0.2m
Thickness of disks	0.01m

Since this is a cross flow water turbine, the water flow should be inward to hit the blade twice in order to maximize the usage of the potential energy.

4.7 Bearings Design

A bearing arrangement does not only consist of rolling bearings but includes the components associated with the bearings such as the shaft and housing. The lubricant is also a very important component of the bearing arrangement because it has to prevent wear and protect against corrosion especially for a hydraulic turbine so that the bearing can deploy its full performance. Beside these, the seal is also a very important component, the performance of which is of vital importance to the cleanliness of the lubricant. Cleanliness has a profound effect on bearing service life. To design a rolling bearing arrangement it is necessary to select a suitable bearing type and to determine a suitable bearing size, but this is not all. Several other aspects have to be considered, such as:

- a suitable form and design of other components of the arrangement
- appropriate fits and bearing internal clearance or preload
- holding devices

- adequate seals
- the type and quantity of lubricant
- installation and removal methods, etc.

Each individual decision affects the performance, reliability and economy of the bearing arrangement.

The Calculations were shown in appendix. The bearings will be protected by the bearing's protectors. The bearings chosen are the ball bearings.

The recommended bearings by SKF [12] are 2 Deep groove ball bearings, single row, unsealed with designated serial number 61906.

Table 6: Bearings' Recommendations

	Option 1	Option 2
Inner Diameter	30mm	30mm
Outer Diameter	42mm	47mm
Basic Dynamic Load Ratings	4.49kN	7.28kN
Basic Static Load Ratings	2.9kN	4.55kN
Fatigue Load Limit	0.146kN	0.212kN
Reference Speed Ratings	32000rpm	30000rpm
Limiting Speed Ratings	20000rpm	19000rpm
Mass	0.027kg	0.051kg
Designation Serial Number	61806	61906

Table 7: Bearings' Dimensions

Dimension	Value
Overall length	0.08m
Overall width	0.05m
Circular radius	0.03m
Thickness	0.05m

There will be 2 bearings cover the 2 sides of the runner. The 2 bearings will be separated by the runner with the bearing's protectors to avoid the splashing water. The bearings will need lubrication.

The overall force for the bearings to support the turbine is 545N. Each bearing will have to support approximately 272N. We will need a surface area of $4000mm^2$ to $5000mm^2$. From the available bearings, we will the one with 50mm x 100mm.
Whenever a shaft rotates, it needs a bearing arrangement for smooth and effective operation. Wherever there is a bearing, you will always find a seal helping it to reach its maximum service life and reliability. The most common types of seals used in bearing arrangements are radial shaft seals.

Selecting the correct lubrication for a particular bearing is a crucial step if a bearing is to live up to design expectations in its application.

CHAPTER 4 (PART B)

FABRICATION DISCUSSION

4.10 Runner's Fabrication

The runner is consisting of 18 blades and 2 plates. The blades and the plates can be shaped through forging or cut through milling and weld together. In one technique, the forgings are provided with a central, axial cavity or hole at a planned site of a circumferential weld to reduce the depth of the required weld to the minimum necessary for adequate strength. That is, the weld cross section is typically an annulus when viewed axially down a rotor.

4.11 Turbine's Fabrication

The shaft was first welded with the runner. Then the shaft was attached to the bearings. The casing was soon being put on. The whole turbine was then being mounted on a stand. The fabrication of the turbine was being done outside UTP due to the unavailability of some of the machining equipment in UTP. The fabrication was mainly done outside of UTP.

4.12 The Base Construction for the Turbine

We will use some metal such as steel to make a base construction for the turbine. The specs for the base construction will be calculated as the turbine is fabricated to get the real value of the turbine.

The base construction will be consists of 2 beams of steel on top of 2 beams in perpendicular. The thickness of the beams should be nothing less than 30mm. The width of the beams should be nothing less than 60 mm. The approximate value of the volume of the turbine is 5×10^3 cm^3. The weight of the turbine is approximately 13kg. From that value, we can find the suitable specifications for the stand. The approximate drawing are shown.

4.13 Prototype

Due to the time constraint and certain issue, the water turbine is yet to be mounted and installed in the lab for testing. So, there is currently no experimental data for this project. However, the remaining objective had been fulfilled. The material in used for the cross flow water turbine was the mild steel. The size of the cross flow water turbine can be place on the existing water tank at the lab.

Figure 11 shown the main parts of the turbine still in part as the fabrication is still in process. Figure 12a, figure 12b and figure 12c show the turbine in full assemblies. Figure 12b and figure 12c show the turbine mounted on the stand.

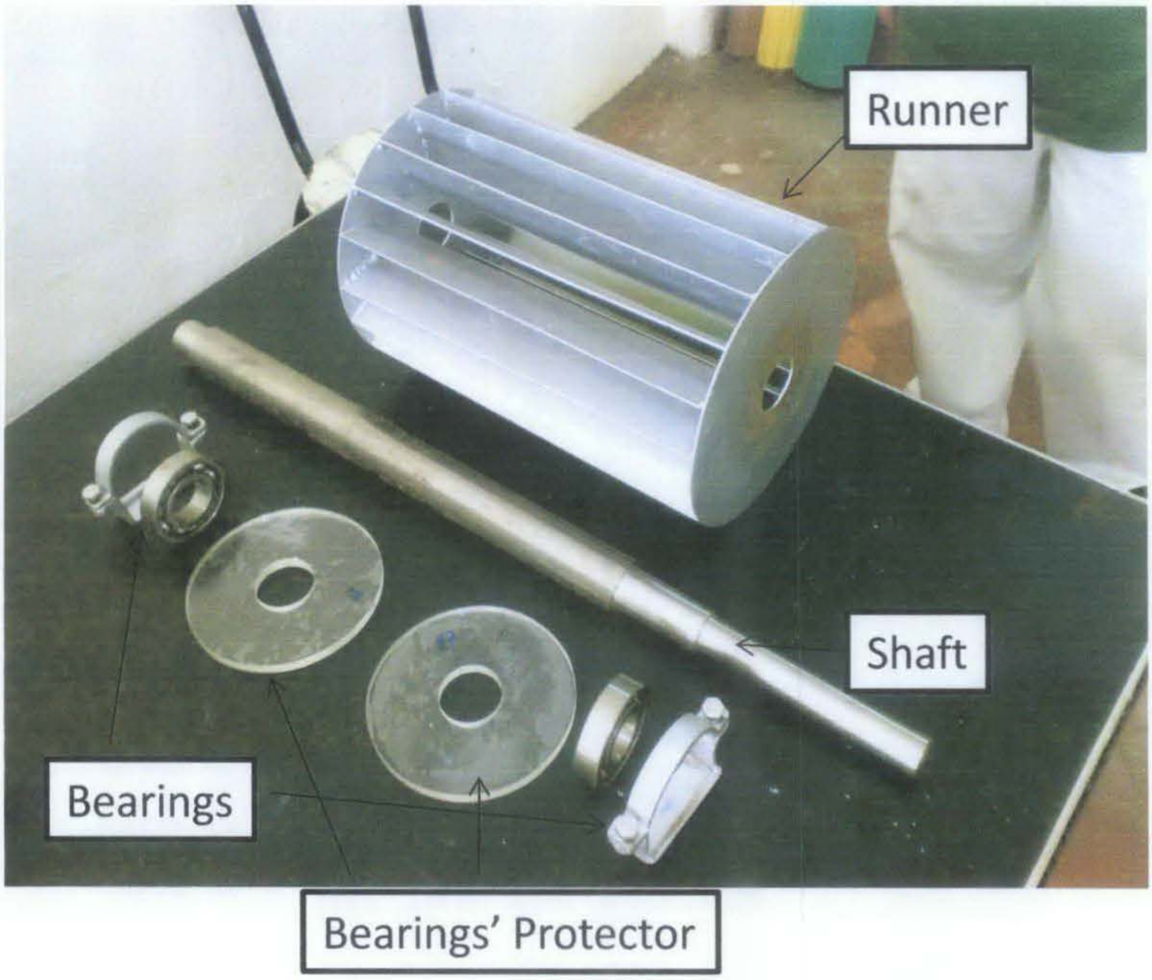

Figure 11: Turbine's Part

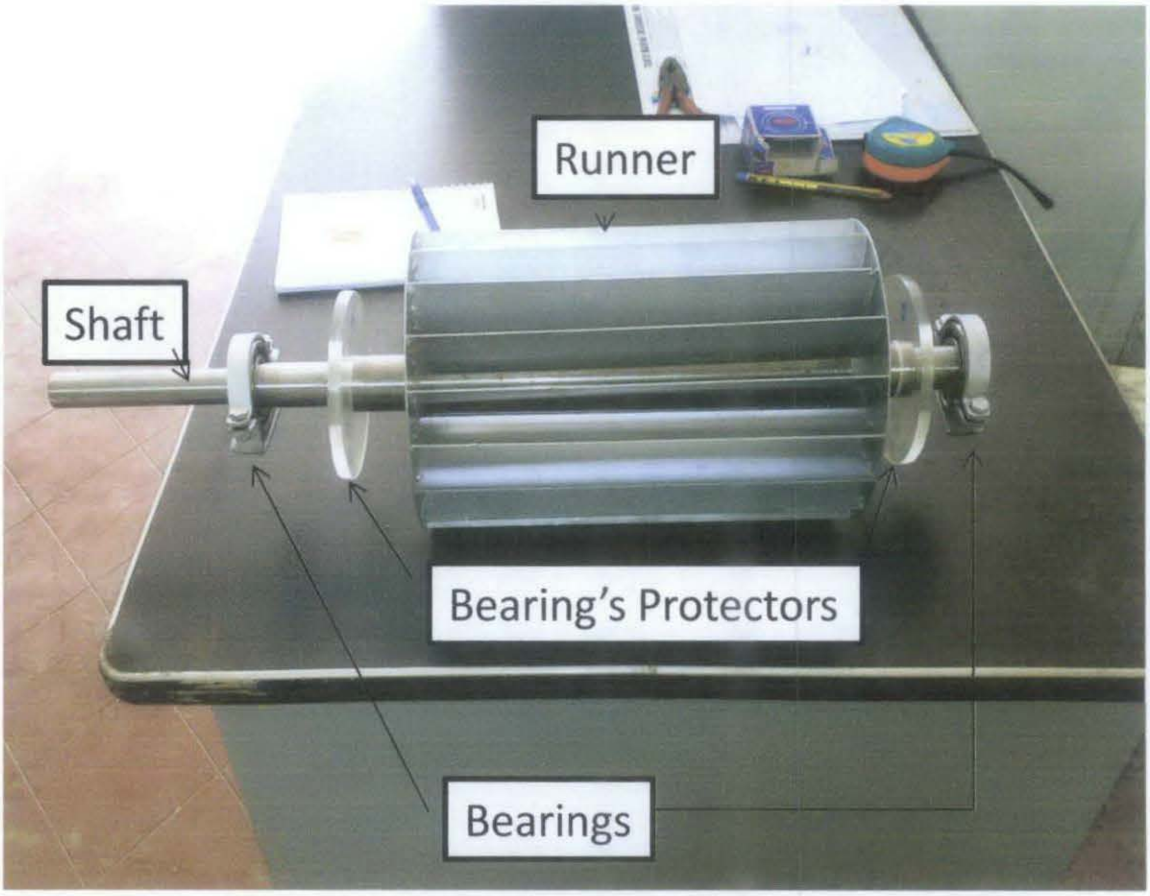

Figure 12a: Turbine's Assemblies

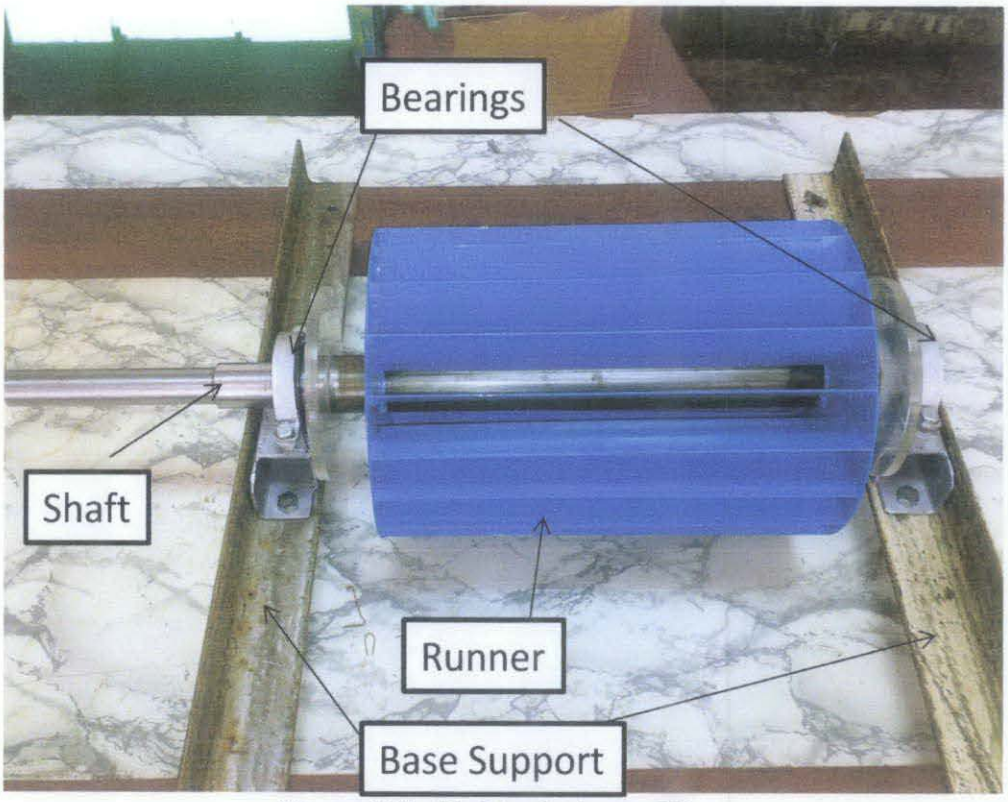

Figure 12b: Turbine's Assemblies

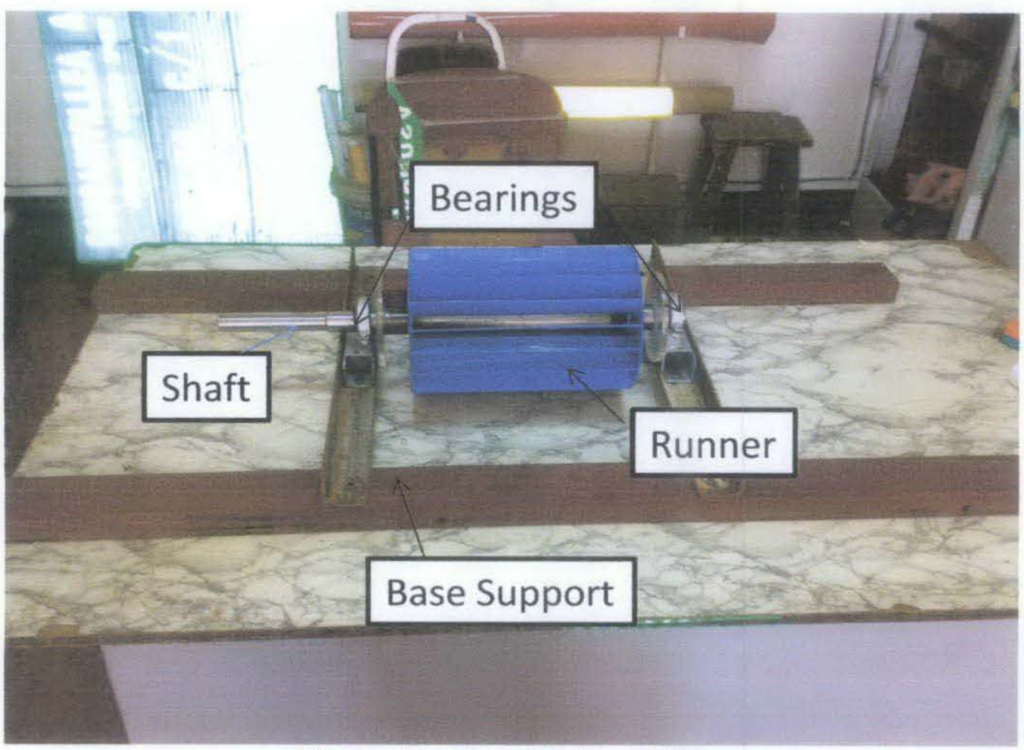

Figure 12c: Turbine Assemblies

CHAPTER 4 (PART C)

PERFORMANCE BASED ON MATHEMATICAL CALCULATION

4.14 Theoretical Performance by Equation

Due to the unavailable of the testing, a theoretical performance was carried out to approximate the actual performance of the Cross Flow Water Turbine based on the mathematical calculation. Velocity of stream of water at radius r from the center is, [1] [11]

$$C = Q / (2\pi b)$$

Where

Q = flow rate

B = distance separating disk.

In this project's design, the distance separating disks is the length of blade.

Specific speed for turbine is

$$N_s = NP^{0.5} / \rho^{0.5} (gH)^{1.25}$$

Or

$$N_s = N_p^{0.5} / (H)^{1.25}$$

The general equation for power is

$$P = ngHdQ \quad , \text{assume } n=0.3$$

H is the water head which must be modified due to head loss from nozzle, fluid friction and kinetic energy loss.

Nozzle head loss

$$= (1 / C_v - 1)\ (V_{j2} / 2g)$$

Fluid friction

$$= kw_1^2 / 2g$$

Lost kinetic Energy

$$=V^2 / 2g$$

The value for C_v is around 0.97 – 0.98. In this case, we will take it as 0.97.

The value V_j is calculated from the equation of jet velocity

$$V_j = C_v\ (2gH)^{0.5}$$

Solving for the modified H and theoretical H, we get

$$\begin{aligned} H_{mod} &= C_v^2\ (2gH_{theo}) + H_{theo} \\ &= 0.97^2\ (2 \times 9.81 \times H_{theo}) + H_{theo} \\ &= 19.5\ H_{theo} \end{aligned}$$

4.15 Code for C Programming

Since the testing can't be conducted, a C program had been done to ease the future testing. The C program will calculate the blade information as some of the results obtained from the testing were inserted. Please refer to Appendix A for the C program

4.16 Theoretical Results

As the equation shown, the output power is directly proportional to the water head or water flow rate; a graph had been drawn to show the impact of water head and water flow rate on the output power from zero to maximum output. Since the pump can't control the water head and water flow rate separately, both the water head and water flow rate will share a same percentage of their maximum capability at a time.

The maximum water head of the pump is 15m and the maximum flow rate of the pump is $0.05m^3/s$. The graph had been drawn with the point where both the variables are at their 20%, 40%, 60%, 80% and 100% of their capability.

Table 5: Theoretical Output Power with varied input

Percentage of capability (%)	0	20	40	60	80	100
Theoretical Output Power (kW)	0	0.292	1.168	2.628	4.672	7.3
Water Head (m)	0	3	6	9	12	15
Water Flow Rate (m^3/s)	0	0.01	0.02	0.03	0.04	0.05

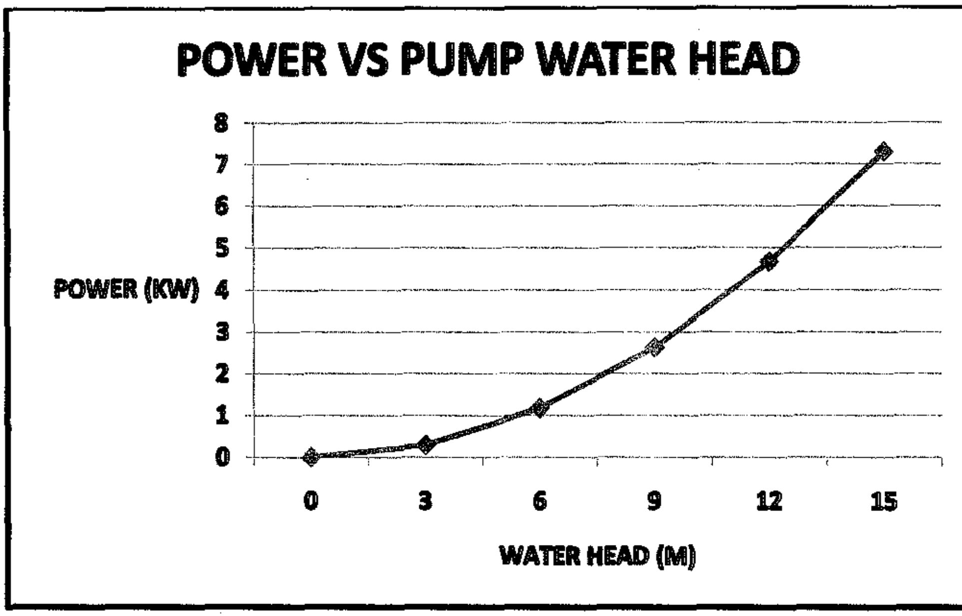

Figure 13: Power vs Water Head

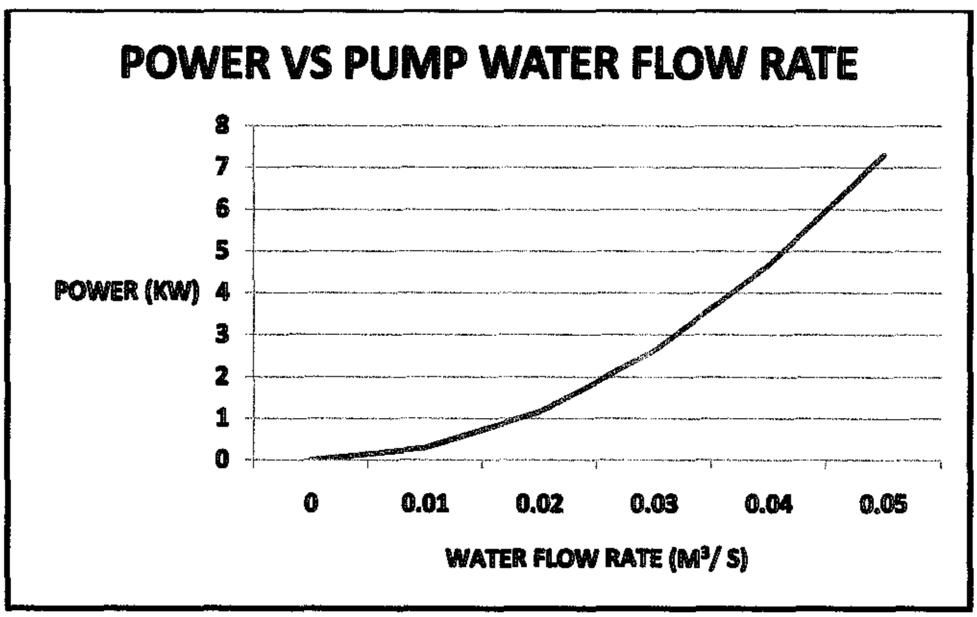

Figure 14: Power vs Water Flow Rate

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

The cross flow water turbine is yet to be installed. It is recommended that the project is continued so the cross flow water turbine can be installed and tested in the lab. A testing system should be designed so the cross flow water turbine could be tested.

The objectives fulfilled including the design of the cross flow water turbine, the theoretical performance of the cross flow water turbine and the fabrication of the design itself.

The Cross Flow Water Turbine designed and fabricated was based on the available water tank and pump in the lab. The size of the turbine was mostly based on the size of the water tank and the performance of the turbine was based on the performance of the pump.

The design of the Cross Flow Water Turbine was mainly focused on the main parts of the turbine namely shaft, runner and bearings. The shaft, runner and bearings of the turbine are important to measure the turbine's performance and feasibility.

The prototype was well fabricated and furnished. Due to the unavailability to test the turbine, the turbine was forecasted the theoretical result of the turbine. The turbine's overall performance should be proportional to the pump's water head and water flow rate.

5.2 Recommendations

After this project was done, the cross flow water turbine is produced. It is recommended that the cross flow water turbine is installed and tested in the future. The testing measurement should be design and fabricated in order to measure the performance of the cross flow water turbine.

A simple braking system is recommended as the measurement system. The braking force can be calculated from the braking system. Then the torque of the shaft can be calculated by multiplying both the braking force and the radius of the pulley. The torque will be used to calculate the mechanical power of the shaft for the measurement purposes.

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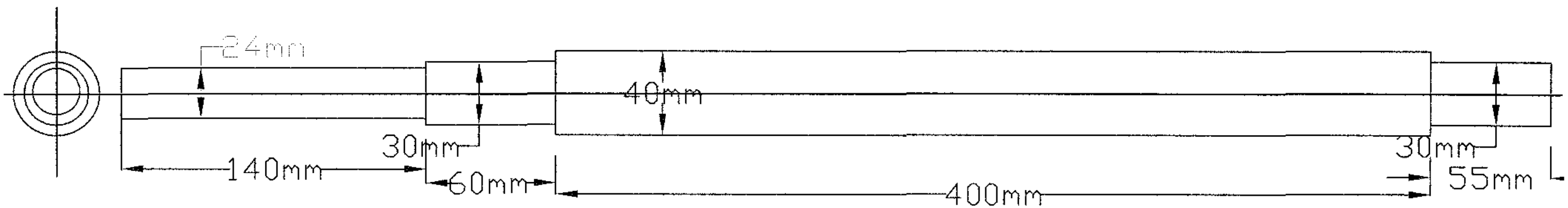
APPENDIX

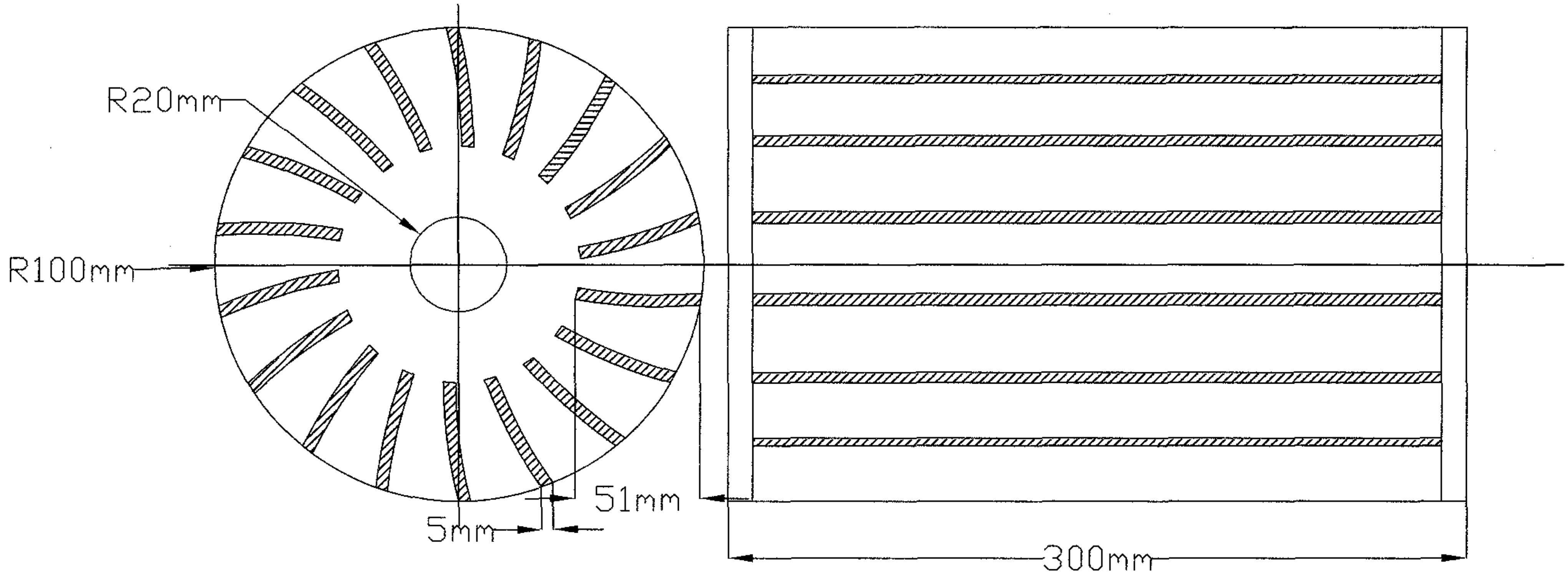
APPENDIX A

```
# include <stdio.h>
# include <math.h>
# include <conio.h>
Main ( )
{
Double Ca, Cx1m, Cx2m, Cx1, Cx2, I, j, k, l, A1, A2, B1, B2;
Float alm, blm, a1, a2, b1, b2, b2m, Dm, Dt, D, N, Um, Ut, Ur, U, R;
Int x;
Clrscr ( );
Printf ("enter blade tip diameter (m) = ");
Scanf ("%f", &Dt);
Printf ("enter blade root diameter (m) = ");
Scanf ("%f", &Dr);
Printf ("enter mean rotor blade inlet angle (deg) = ");
Scanf ("%f", &blm);
Printf ("enter mean root blade outlet angle (deg) = ");
Scanf ("%f", &b2m);
Printf ("enter mean root blade outlet angle (deg) = ");
Scanf ("%f", &alm);
Printf ("enter the turbine speed (rpm) = ");
Scanf ("%f", &N);
Clrscr ( );
Dm = (Dt + Dr) / 2;
Um = (3.14 * Dm * N) / 60;
I = alm * 3.14 / 180;
j = blm * 3.14 / 180;
k = tan (i) – tan (j);
l = b2m * 3.14 / 180;
Ca = Um / k;
Cx1m = Ca * tan (i);
Cx2m = Ca * tan (1) – Um;
For (x = 1; x<=2; x++)
{
If (x = = 1)
{ printf ("blade root angles (deg.) : \n\n);
D = Dr;
}
Else
{ printf ("blade tip angles (deg.): \n\n);
```

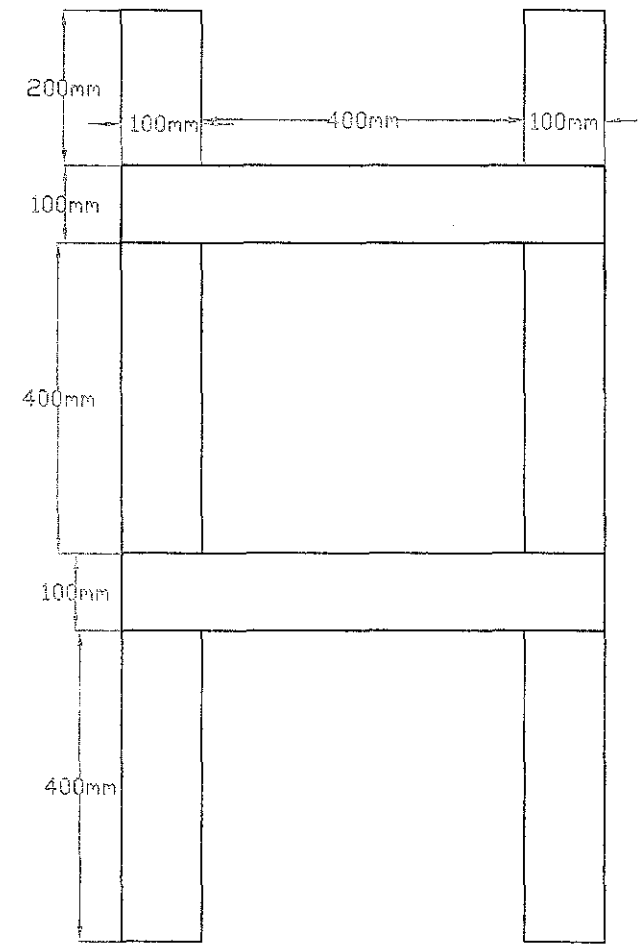
```
D = Dt;
}
Cx1 = Cx1m*Dm/D;
A1 = atan (Cx1/Ca);
a1 = A1*180/3.14;
Printf ("Stator blade exit angle = %f \n", a1);
U = 3.14 * D * N / 60;
B1 = atan (tan (A1) – U / Ca);
B1 = B1 * 180 / 3.14;
Printf ("Rotor blade inlet angle = %f \n", b1);
Cx2 = Cx2m * Dm / D;
A2 = atan (Cx2 / Ca);
a2 = A2 * 180 / 3.14
Printf ("Stator blade inlet angle = %f \n", a2);
U = 3.14 * D * N / 60;
B2 = atan (tan (A2) – U / Ca);
b2 = B2 * 180 / 3.14;
printf ("rotor blade exit angle = %f \n", b2);
R = Ca* (tan (B2) – tan (B1) / (2*U);
Printf ("degree of reaction = %f \n\n", R);
}
}
```



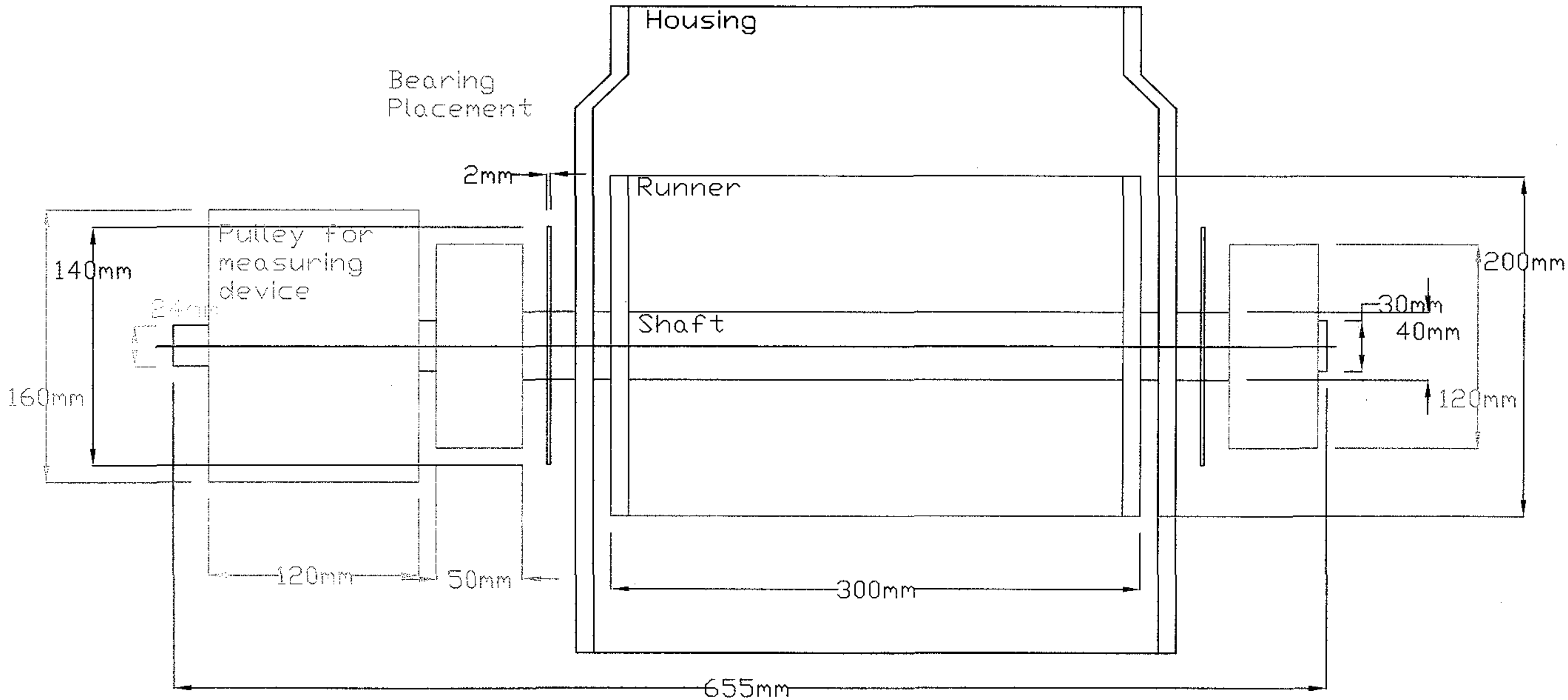
Mechanical Engineering Department
Material: Mild Steel
Drawing: Shaft Design
Ratio: 1:1
Measurement in mm



Mechanical Engineering Department
Material: Mild Steel
Drawing: Runner Design
Ratio: 1:1
Measurement in mm



Mechanical Engineering Department
Material: Mild Steel
Drawing: Base Support Design
Ratio: 1:1
Measurement in mm



Mechanical Engineering Department
Material: Mild Steel
Drawing: Turbine Design
Ratio: 1:1
Measurement in mm