DESIGN, CONSTRUCTION AND TESTING OF AN OUTWARD RADIAL-FLOW REACTION WATER TURBINE

BY

AYINDE ABDULLAHI NURUDEEN

DEPARTMENT OF MECHANICAL ENGINEERING
AHMADU BELLO UNIVERSITY, ZARIA

DECEMBER, 2007

DESIGN, CONSTRUCTION AND TESTING OF AN OUTWARD RADIAL-FLOW REACTION WATER TURBINE

BY

AYINDE ABDULLAHI NURUDEEN B. Eng (ABU 1991),
PGDM (ABU 2002), M.Sc/Eng/35197/02-03

A DISSERTATION SUBMITTED TO THE POSTGRADUATE SCHOOL
AHMADU BELLO UNIVERSITY, ZARIA
NIGERIA

IN PARTIAL FULFILMENT FOR THE AWARD OF MASTER OF SCIENCE IN MECHANICAL ENGINEERING (ENERGY CONVERSION SYSTEMS)

DEPARTMENT OF MECHANICAL ENGINEERING
AHMADU BELLO UNIVERSITY, ZARIA - NIGERIA

DECEMBER, 2007

DECLARATION

I declare that the work in the dissertation entitled `Design, construction and testing of an outward radial – flow reaction turbine' has been performed by me in the Department of Mechanical Engineering under the supervision of Dr. E.J. Bala.

The information derived from the literature has been duly acknowledged in the text and a list of references provided. This dissertation has not been presented for another degree or diploma at any University previously.

____________________	_______________	____________
Name of Student	**Signature**	**Date**

CERTIFICATION

This dissertation entitled `Design, construction and testing of an outward radial flow reaction turbine' by AYINDE Abdullahi Nurudeen meets the regulations governing the award of Master of Science – Mechanical Engineering (Energy Conversion Systems) of Ahmadu Bello University, Zaria – Nigeria, and is approved for its contribution to knowledge and literary presentation

________________________ ____________
Chairman, Supervisory Committee Date

________________________ ____________
Member, Supervisory Committee Date

________________________ ____________
Member, Supervisory Committee Date

________________________ ____________
Prof. P. Madakson Date
Head of Department

________________________ ____________
Prof. S.A. Nkom Date
Dean, Postgraduate School

DEDICATION

This work is dedicated to Almighty God.

ACKNOWLEDGEMENT

First, I thank Almighty God for giving me the opportunity (i.e. strength, good health and life) to pursue this degree. Then I thank my supervisor, Dr. E.J. Bala for his invaluable assistance throughout the duration of this work. I also thank the Postgraduate Coordinator, Professor C.O. Folayan for his untiring support and encouragement in the course of this work. I sincerely appreciate the untiring support, advice and physical participation of Engr. F.O Anafi. He was always there for me. Engr. M.N. Gungura, my classmate and a staff of NCAT, Zaria, Mr. M.A Yusuf, also a staff of NCAT, Zaria and other NCAT staff played a part in the success of this work. God bless you all.

I appreciate the support of the Acting Executive Director in person of Engr. S.O. Jolaiya and Mr. A. Sani for allowing me use the facilities at the Centre for Automotive Design and Development, Zaria. I also appreciate the criticism and advice of Engrs. L.A. Isah, F.M. Achiv and N.O. Omisanya. Engr. Sam John who was the acting head of department as the time I applied for this course deserve commendation for ensuring that the Postgraduate courses were continued in the 2002/2003 session after more than one decade suspension. I thank him for his encouragement. My sincere thanks go to the head of Mechanical Engineering Department, Prof. P. Madakson and Engr. G.Y Pam for their encouragement and support.

I also say thank you to the Chief Technologist, Mr. Yakubu Lawal of Mechanical Engineering Department who apart from assisting in other ways, directly machined one of the components of this work.I want to specifically thank Mr. Mike A. Okah and Chima John for their advice in the electrical aspect of the work. I thank

Technicians like Elisha, Bawa, Bashir, Shehu, all of CADD, some IDC, Zaria staff and NCAT staff, some Civil Engineering Department staff and Mr. Samaila and Mr. Shehu in the Mechanical Engineering Workshop for their contributions to the fabrication of this water turbine. Mr. Emmanuel Ochigbo was always receptive whenever I wanted to cast or repeat any cast part; I say a big thank you to him. I thank Engr. Aminu in-charge of A.B.U, Water Works and other staff of his department for helping me with the leakage problems and also testing the turbine.

Let me take this time out to say a big thank you to my wife Mrs. Bintu Taiwo Ayinde, my late son Abubakar, our new baby Adnan and my younger brother Abdulrafiu for their understanding. I love you all.

I thank Mr. Mike V. Vongdul and Mrs. Eucharia Egwu for carefully typing this dissertation. I also thank Engr. A.O. Oyetade and Mr. A.Giwa for helping out with tables, graphs and drawings. Finally I thank all those others who contributed in one way or the other to the success of this work.

ABSTRACT

An outward radial flow reaction water turbine was designed based on Euler one-dimensional theory, constructed and tested to drive a 6W bicycle electric dynamo using water from supply mains. The turbine watered and lit the place it was tested eliminating the use of energy from other costlier sources which are not as environmental friendly. Available literature showed that water and steam had been used in providing the needed pressure for turbines based on the one-dimensional theory with successes recorded in the past. Dedicated efforts were made in Europe and America at improving different designs until Pelton impulse turbine, the Francis turbine and the Kaplan turbine now universally accepted were satisfactorily designed. In the production of this turbine (water sprinkler), the interest was more on power generation than watering, therefore four arms were used to discharge more water so as to deliver surplus of useful power. Consequently, the design angular speed of the turbine, the rotor arm diameter, the work done on each arm by the fluid, the tangential force and tolerances were determined. Mild steel, cast aluminium alloy and stainless steel were used to fabricate this turbine because they were readily available, cheap and machinable. A range of power output of 0.42W to 2.66W was obtained for a range of mains pressure of $1.5 \times 10^5\, {}^{N}\!/_{m^2}$ to $2.8 \times 10^5\, {}^{N}\!/_{m^2}$. The results obtained from readings taken were tabulated and graphs plotted accordingly. The optimum performance occurred at $2 \times 10^5\, {}^{N}\!/_{m^2}$, a speed of 600rpm and a mass flow rate of 0.685 kg/s. The maximum power obtained at maximum mains pressure of $2.8 \times 10^5\, {}^{N}\!/_{m^2}$ was 2.66W. The cost of the water turbine stood at thirty five thousand, three hundred and eighty naira only (₦35, 380.00) and this would reduce greatly when

the water turbine is mass produced. Power was generated to lit the place that the water turbine was used from the residual power head.

TABLE OF CONTENTS

CHAPTER 4: CONSTRUCTION OF THE OUTWARD RADIAL FLOW REACTION WATER TURBINE

CHAPTER 5: TESTS AND RESULTS

LIST OF TABLES

TABLE

LIST OF FIGURES

FIGURE

LIST OF DRAWINGS

DRAWING

LIST OF PLATES

LIST OF SYMBOLS

A_{in}	-	Area of the mains inlet pipe (m^2)
A_1	-	Area of the rotor arm (m^2)
A_2	-	Area of the nozzle (m^2)
D_m	-	Diameter of the pinion on the shaft of electric motor (m)
D_d	-	Diameter of the pinion of the dynamo (m)
D_T	-	Diameter of pinion of the out shaft of the turbine (m)
d_1	-	Diameter of the rotor arm (m)
d_2	-	Diameter of nozzle at exit (m)
d_{in}	-	Diameter of mains inlet pipe (m)
E	-	Head (m)
$E_{available}$	-	Available head (m)
$E_{utilized}$	-	Utilized head (m)
E_t	-	Work done per second (W)
F	-	Tangential force (N)
g	-	Acceleration due to gravity (m/s^2)
H_{th}	-	Ideal theoretical head (m)
h_1, h_2	-	Heads at inlet and outlet (m)
$\dot{m}$	-	Mass flow rate (kg/s)
N_m	-	Maximum angular speed of the output shaft of the electric motor (rpm)
N_d	-	Maximum angular speed of the dynamo (rpm)
N_r	-	Angular speed of the turbine (rpm)

P_{oth} - Theoretical output power (W)

P_{OA} - Actual output power (W)

P_{in} - Input power (W)

P_1 - Pressure at inlet (N/m^2)

P_2 - Pressure at outlet (N/m^2)

ΔP - Mains Pressure (N/m^2)

Q - Volume flow rate (m^3/s)

R - Degree of reaction or reaction

r_1 - Radius of a cylindrical surface inlet (m)

r_2 - Radius of a cylindrical surface at outlet (m)

r - Radius of one rotor arm i.e. distance between the axis of nozzle and axis of rotation of the rotor (m)

T - Torque (Nm)

U_1 - Tangential velocity of the rotor in general (m/s)

U_2 - Tangential blade velocity (m/s)

V_s - Space velocity (m/s^2)

V_1 - Absolute velocity at any rotor inlet in general (m/s).

V_2 - Absolute velocity at any rotor outlet in general (m/s)

V_{in} - The velocity of fluid in the main inlet pipe (m/s)

V_{r_1} - Relative velocity of fluid with respect to the rotor blades at r_1 (m/s)

V_{r_2} - Relative Velocity (m/s)

V_{f_1} - The radial component of the absolute inlet velocity (m/s)

V_{f_2} - The radial component of the absolute outlet velocity (m/s)

V_{w_1} - The tangential component of the inlet absolute velocity perpendicular to the radial one (m/s)

V_{w_2} - The tangential component of the outlet absolute velocity perpendicular to the radial one (m/s)

Y - Rate of energy transfer/unit mass of fluid flowing (J/kg)

α_1 - Angle the absolute inlet velocity makes with the tangent at a particular radius.

α_2 - Angle the absolute outlet velocity makes with the tangent at the outlet.

β_1 - The blade angle at inlet

β_2 - The blade angle at outlet

ε - Utilization factor

η_h - Hydraulic efficiency

η - Overall efficiency

η_m - Mechanical efficiency

η_{vane} - Vane efficiency

ρ - Density (kg/m^3)

ω - Angular speed of the rotor (rad/s)

APPENDIX

CHAPTER 1 INTRODUCTION

1.1 Statement of the Problem

Water in nature is a useful source of energy. Its energy comes directly in mechanical form, without the losses involved in heat engines and fuel cells, and no fuels are necessary. Solar heat evaporates water, mostly from the oceans, where it is mixed into the lower atmosphere by turbulence, and moved by the winds. Through meteorological processes, it falls on the earth as precipitation, on the oceans, but also on high ground, where it makes its way downhill to the sea, without evaporative and other losses. A cubic meter of water can give 9800J of mechanical energy for every meter it descends, and a flow of a cubic meter per second in a fall of 1m can provide 9800W, or 13hp. The efficiency of hydraulic machines can be made close to 100%, so that this energy is available, and it can be converted to electrical energy with an efficiency of over 95%. Hydropower and increasing population cannot co-exist; the limits of hydropower are fixed and obvious. The percentage of hydropower is dropping by the day. It is not that hydropower is decreasing in absolute terms; it had remained roughly constant while the total market had expanded greatly. The oil reserve is depleting and the cost is constantly on the increase.

Sprinklers are mainly used for sprinkling water in many areas, namely; Truck crops, nurseries, orchards, gardens, lawns, application of fertilizer, soil amendments, log curing, water distribution for compaction of earth fills, setting of dust, farm fire protection, frost protection, cooling crops and animals, dewatering of mines and excavation.

Considering these areas where sprinklers are used, lighting these places will greatly enhance the working time, and ease working in the places at night. This is currently possible by using power from National grid or other power sources which are costly and could be eliminated if we can use the residual power head to rotate a dynamo and generate electricity.

1.2 Significance of the Study or Justification for the Study

Hydro-energy technology is environment friendly, renewed naturally by rainwater and melting of snow on high mountains during summers and simple. Waterpower is available whenever a sufficient volume of steady water flow exists. Hydro schemes are multipurpose. The water is used before and after power generation and there is no wastage of water. Only the head is lost during power generation. The operating cost of water turbine is very low, it has long service life and the renewable energy resource occurs free of cost.

Energy is essential for sustaining civilization, the economic prosperity of a nation or region or state or individual consumer is directly influenced by the quantity of energy generated and consumed. Thus, there is a need to generate as much energy as possible from all sources to meet human requirements. One way of achieving this is through power generation from reaction hydro turbines (sprinkler heads).

A system is more efficient and effective, if it is made to operate as independent as possible from other system which is the case of this reaction turbine (sprinkler).

1.3 Theoretical Framework

The real flow through a rotor is three-dimensional, that is to say the velocity of the fluid is a function of three positional coordinates, say, in the cylindrical system, r, θ and z. Thus, there is a variation of velocity not only along the radius but also across the passage in any plane parallel to the rotor rotation, which constitutes an abrupt change – a discontinuity. Also, there is variation of velocity in the meridional plane, i.e. along the axis of the rotor. The velocity distribution is, therefore, very complex, and dependent upon the width of the rotor and its variation with radius.

The one – dimensional theory simplifies the problem very considerably by making the following assumptions.

(i) That the flow is axisymmetric, which means that there is a perfect symmetry with regard to the axis of rotor rotation. Thus;

$$\frac{\delta V}{\delta \theta} = 0$$

(ii) Over that part of the rotor where transfer of energy takes place,

There is no variation of velocity in the meridional plane, i.e. across the width of the rotor. Thus,

$$\frac{\delta V}{\delta z} = 0$$

The result of these assumptions is that whereas, in reality,

$$V = V(r,\theta,z)$$

For the one-dimensional flow

$$V_\infty = V(r)$$

The suffix ∞ stipulates axisymmetry

(iii) That there is imaginary body forces acting on the fluid and producing torque

1.4 Objectives of the Study

Given that there is increasing need to find other sources of energy and the fact that a sprinkler is the simplest form of a reaction water turbine, it is necessary to devise a means by which electricity could be generated from the sprinkler heads. In view of the importance of lighting the places that sprinklers are used when it is dark, use of other power sources rather than the sprinkler head will be eliminated. The objective of this research is therefore to review the design of a simple outward flow reaction water turbine and develop it to serve as a water sprinkler as well as a source of mechanical power for driving an electric dynamo for the generation of electricity to lighten up the place it is used when it is dark whenever there is water in the supply mains. Test is also to be carried out to ascertain the efficiencies of the device under various water pressures.

1.5 Statement of Research Questions

The fact that sprinklers could be built to serve as water turbine as well, and given that hydro power had remained roughly constant while the total market had expanded greatly, water could be further exploited by building water turbines to serve as sprinklers as well as a source of mechanical power for driving electric dynamos for the generation of electricity to lighten up the places. They would be used when it is dark whenever there is water in the supply mains.

The focus of this work is therefore how to exploit energy from water turbines (sprinklers) to generate electricity. The effect of load (electric dynamo) on the efficiency of sprinkler is also to be determined.

CHAPTER 2 LITERATURE REVIEW

2.1 Introduction

The word turbine was coined in 1828 by Claude Burdin (1790-1873) to describe the subject of an engineering competition for a water power source. [Calvert J.B (2003).] It comes from Latin word turbo, meaning a "whirling" or a "vortex", and by extension a child's top or a spindle. Defining a turbine as a rotating machine for deriving power from water is not quite exact. The precise definition is a machine in which the water moves relatively to the surfaces of the machine, as distinguished from machines in which such motion is secondary, as with a cylinder and piston. The common water wheel is a rotating machine, but not a turbine. Many types of prime movers are discussed in this report, but mainly turbines, the fundamental theory of which is explained.

The disadvantage of energy from water is that it is strictly limited, and widely distributed in small amounts that are difficult to exploit. Only where a lot of water is gathered in a large river, or where descent is rapid, it is possible to take economic advantage. Most of these possibilities are quite small, as are the hydropower sites along the fall line on the Atlantic coast of the United States, or on the slopes of the pennies in England. These were developed in the early days of the Industrial Revolution, but are now abandoned because their scale is not the scale of modern industry. Each site provided a strictly limited horsepower, and in the autumns the water often failed. For expansion and reliability, all were rapidly replaced by steam engines fueled by coal, which were expandable and reliable. Today, hydropower usually means a large project on a major river, with extensive environmental damage. The fall in head is provided by a dam, which creates a lake that will be of limited life, since geological processes hate lakes and destroy them as rapidly as possible.

Niagara Falls, east of central North America in western New York and southeastern Ontario, forming part of the U.S-Canadian boundary is an excellent example of a hydropower site. It is unique; there is only one, and hardly anything else similar. The Niagara River carries the entire discharge of the Great Lakes, about $5520m^3/s$, and the concentrated elevation difference is about 50m. The visible falls carry nothing like this much water today; most is used for power. Hydropower could destroy the falls as a sublime view; we are lucky it has not. The power available from this discharge and drop is $3.6x10^6$hp. The figures given in the encyclopedia for the power available from the Canadian and U.S. power projects on each side add up to considerably more than this. Perhaps they use more drop, or perhaps they are just optimistic. The first large-scale hydropower development here was in 1896. This was also the site of Nikola Tesla's two-phase plant that pioneered polyphase power in the U.S.

For comparison, the more than 190 million registered motor vehicles in the U.S. probably have an aggregate power capability of nearly $2 \times 10^{10} hp$, equivalent to 5000 Niagara. Hydropower and increasing population cannot co-exist; the limits of hydropower are fixed and obvious. It is really too bad that small-scale hydropower projects are no longer economically viable. In 1920, about 40% of electric power in the U.S. came from hydropower; in 1989 that percentage had dropped to 9.5%. It was not that hydropower had decreased in absolute terms, but had remained roughly constant while the total market had expanded greatly. [Calvert J.B (2003).]

2.2 Types of Turbo machines

There are many ways of classifying turbo machines having common features, although all methods tend to overlap. There is no clear-cut division into groups of unique design and performance, which is one reason for developing simple theory on a general basis. However the behavior of real fluids under particular conditions leads to special design analyses for a number of well-recognized types, and it is convenient to deal with each of these separately.

The first major division, which can be made, is into machines transferring rotor energy to fluid energy, namely, pumps and compressors, and into machines transferring fluid energy to a rotor or mechanical energy, namely turbines. Then there can be a division into machines with radial-flow and those with axial flow. Some machines have both, that is, mixed flow, and these are included in the radial-flow group, axial flow implying no significant element of radial flow at all. The radial-flow group can be subdivided into inward flow and outward flow, as in practice this largely separates machines into familiar types. Thirdly, there can be division into impulse and reaction machines as these have been seen to have distinguishable characteristics. However, since reaction can have such a range of value, it is rather a broad classification, chiefly useful for delineating the pure impulse machine. Finally, there is a difference between machines handling compressible and incompressible fluids, but although in some cases, notably turbines, there is a marked difference in mechanical design, in others such as centrifugal pumps and compressors there is marked similarity. There is a considerable advantage in treating pumps and compressors together, as their basic behavior is very similar and a rigid separation into two technologies is unnecessarily complex.

Having discussed the fundamental behavior of turbo machines as a whole, a further analysis will now be made of several of the above groups and table 2.1 shows method of classification, where a particular type of machine has received a recognized title, usually that of the inventor or original proponent, this is given as the type name; otherwise the title is descriptive of a somewhat broad class. Only common types of machine now being manufactured are considered.

Table 2.1 Classification of Turbines

	Radial flow		Axial flow
	Inward	Outward	
Energy from rotor to fluid		Centrifugal pump, fan, blower, compressor	Axial pump, fan, blower, compressor, propeller
Energy from fluid to rotor	Francis turbine, Gas turbine	Ljungstrom turbine	Steam and gas turbines. Pelton wheel. Kaplan turbine.
	Banki turbine		

2.3 Evolution of Turbines

2.3.1 A Practical Hero Aeolipile

Perhaps in the first century of the current era, Hero of Alexandria described in his pneumatic a rotary machine driven by steam. A hollow sphere was connected to a cauldron by a tube that also served as the axle. Jets of steam from two nozzles attached to the sphere at opposite ends of its diameter, perpendicular to the axle, caused it to rotate at a high speed. Historians of ancient technology regard the Hero turbine as a toy, since it probably couldn't have produced enough power except for the most trivial applications. It might have been powerful enough to turn a roasting spit. [Lyman, F.A. 2004]

Hero's turbine remained a curiosity for more than a thousand years. In the early 19^{th} century, several inventors, including the steam engine builders James Watt and Richard Trevithick in England and Oliver Evans in America, experimented with steam reaction turbines of the Hero type, but without much success. It was in the 1830s that an obscure and now largely forgotten mechanic, William Avery, designed and built a Hero turbine that could manage significant, useful work. It powered several gristmills and sawmills in New York State, and even drove a locomotive. [Lyman, F.A. 2004]

Avery might have read about Hero's machine, but more likely, he had seen or heard of a water turbine invented in England about 1740 by Robert Barker. This turbine, called Barker's mill, was used in Europe and America. It worked by action of water flowing down through an axial pipe, proceeding radially outward through two arms, and finally exiting tangentially from holes on opposite sides of the arms. [Lyman, F.A. 2004]

In 1747, Andreas Segner of the University of Gottingen described a water turbine similar to Barker's mill, but with six jets instead of two. Segner's turbine drew the attention of the Swiss mathematician Leonhard Euler, who analyzed the flow of an ideal fluid in the rotor passages and determined the torque and power that such a machine could produce. [Lyman, F.A. 2004]

Euler’s theory (1754-56) had little influence on the technology of water turbines in the next hundred years. How ever, Euler’s son, Johann Albrecht, built an improved version of Segner’s turbine for experimental purposes, and several other versions of the Barker – Segner – Euler turbine were used to provide power for factories and mills. [Lyman, F.A. 2004]

James Watt apparently tried to get useful work by running some version of Barker’s mill on steam, but without much success. Oliver Evans, who had built one of the first successful high-pressure steam engines, believed Watt failed because he used low-pressure steam and cooled the rotor, causing steam to condense in the arms. Evans wrote that he had tested a 3-foot-long rotary tube at a steam pressure of 56 pounds per square inch, and it turned at speeds of about 700 to 1,000 revolutions per minute. According to Evans, “it exerted more than the power of two men, and would answer to turn lathes, grindstones, etc., when fuel is cheap”. As a young man, he “was continually contriving water mills”, and his skills as a mechanic and millwright were in demand throughout central New York. [Lyman, F.A. 2004]

In 1824, he invented a machine for making wire harness for looms, and his nephew later wrote that thereafter “hardly a year passed without a patent being granted to him”.

Ambrose Foster and William Avery were granted a patent on September 28, 1831, for “their improvement in the steam Engine, commonly called the reacting engine”. According to the patent, “what we claim as our invention is, simply, the giving the oblate, or flat, form to the revolving arms, so that in proportion to their capacity, they shall experience much less resistance from the air than that to which they have been heretofore subjected, thereby obtaining a greatly increased power”.

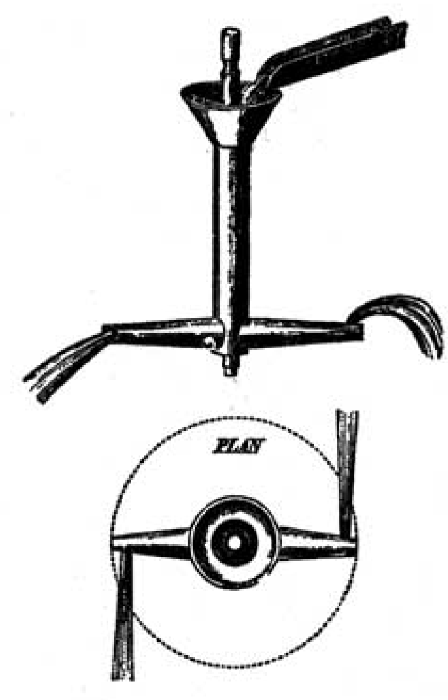


Fig. 2.1 Barker's mill

Instead of making the arms “in the form of round tubes, which has been heretofore done”, Foster and Avery made their cross-section a profile consisting of two circular arcs of the same large radius. They thought that this shape, with its sharp edges, would give “the least possible resistance”, but they noted that tubes of elliptical, or oval, cross-section would reduce the resistance almost as much. Although such aerodynamic intuition might seem quite remarkable for mechanics in 1830, it is a somewhat dubious basis for a patent.

From 1835 to 1837, several notices about “Avery’s Engine” appeared in local newspapers and in mechanics’ magazines with wider circulation. Foster’s name was seldom mentioned. These sources said that the engine not only provided power for all sorts of machinery in Lynds’s shop, but also for several sawmills in Central New York. The salt industry in the nearby town of Salina required considerable amounts of firewood for salt boilers and lumber for solar drying sheds, and lumber was also needed or construction in the burgeoning city of Syracuse. There was need for steam-powered mills in the flat northern reaches of Onondaga County.

The light-speed Avery engine was well-matched to sawing lumber. There was no need to gear it up to match the saw speed, as was necessary with the low-speed reciprocating steam engines of the time. (Oliver Evans’s high-pressure engine drove some Sawmills in Pennsylvania).

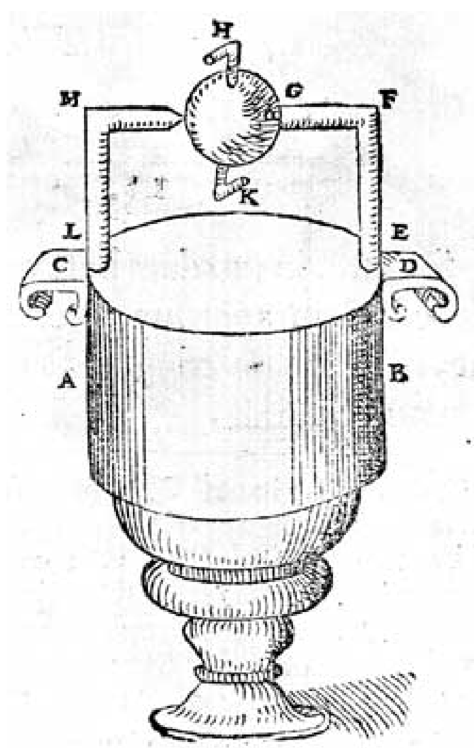


Fig. 2.2 Hero’s steam-driven device

In early 1837, the last year of its existence, mechanics' magazine printed several letters about the use of Avery's engine in gristmills and sawmills. One, from Avery himself, described a gristmill in Cayuga County driven by an engine with a rotor 12 feet in diameter, which made about 1,000 rpm at a steam pressure of 120 psi. Four men from Ithaca wrote a letter about a local sawmill driven by an Avery engine "estimated at 20 horse power". This letter showed a diagram of the sawmill, which had two reciprocating vertical saws connected to opposite ends of a rocking beam driven by a connecting rod from a drum belted to the engine. [Lyman, F.A. 2004]

It's too bad that we have only newspaper and magazine accounts of Avery's turbine. They rely too heavily on the claims of the inventor, manufacturer, and their friends and customers.

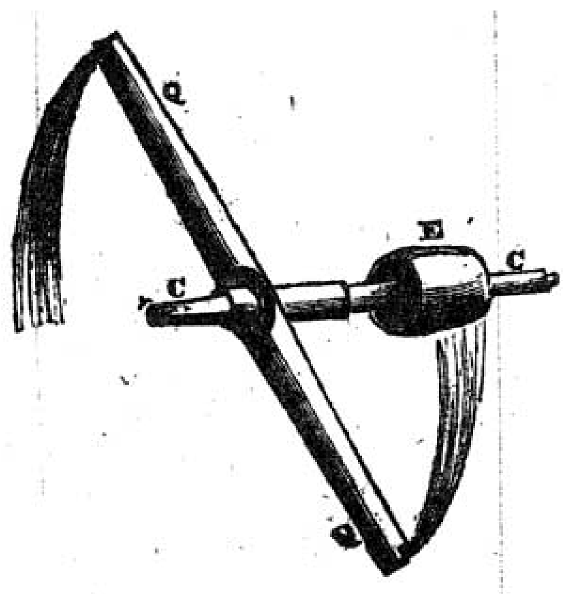


Fig. 2.3 Avery's engine

Sir Charles Parsons, the British engineer who in 1884 patented what was to become the first commercially successful steam reaction turbine, gave the Rede lecture at Cambridge University in 1911. The printed lecture has a photograph of the rotor of Avery's turbine, which was 5 feet in diameter and had a tip speed of 880 feet per second. In Parsons' view, "These wheels were inefficient, and it is not so obvious that an economical engine could be made on this principle". In 1882, Dr. Gustav de Laval had tried to use a Hero-type reaction turbine to drive his high-speed cream separator, but he rejected it and instead developed the impulse turbine that bears his name. [Lyman, F.A. 2004]

The Avery engine probably had other problems; noise, vibration, the difficulty in sealing the rotary coupling, and the problem of speed regulation. These problems would have been difficult to solve with 1830s technology.

A short retrospective on the Avery engine in the Scientific American of November 19, 1864, described two other problems. The cast rotor of one Avery engine had flown apart, and a piece of it had gone "up through two or three floors with a force equal to that of a common shot". The sides and edges of the rotor arms tended to become furrowed and jagged after long use. [Lyman, F.A. 2004]

The article noted that, nevertheless, an Avery engine drove a Saw-mill in New York City for 20 years, and its proprietors later regretted replacing it with a reciprocating engine.

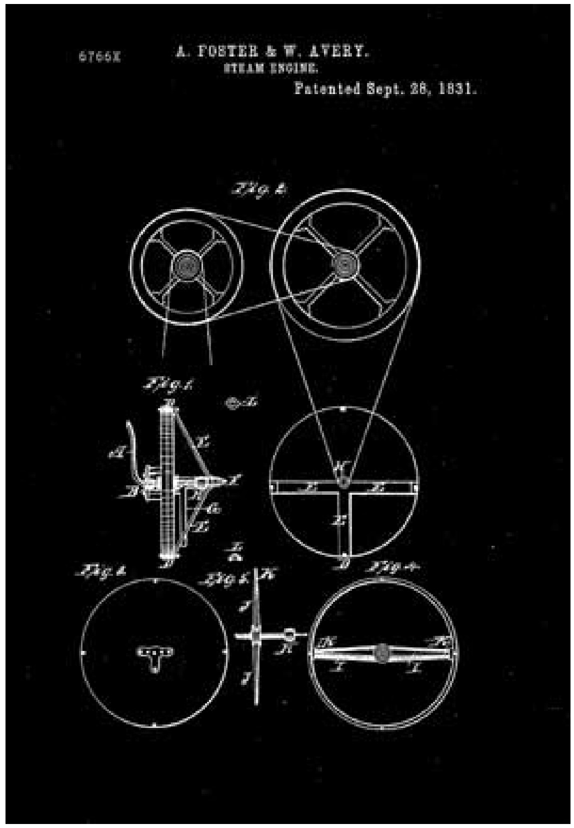


Fig. 2.4 W. Avery and Ambrose Foster steam engine

Avery also built the machinery for the first steamboat on Lake Ontario, as well as for several other lake steamers. [Lyman, F.A. 2004]

William Avery apparently left no descendants, but one of his nephews, John E. Sweet, gained prominence as a Mechanical Engineer and served as the third president of ASME (1884-85)

Sweet established that 50 to 75 Avery engines were built and used to run cotton gins and equipment in Saw-mills and woodworking shops. He also said that the rotor of one of these engines was in the Museum of ASME in New York City. [Lyman, F.A. 2004]

2.3.2 Impulse turbine

The impulse turbine is very easy to understand. A nozzle transforms water under a high head into a powerful jet. The momentum of this jet is destroyed by striking the rotor, which absorbs the resulting force. If the velocity of the water leaving the rotor is nearly zero, all of the kinetic energy of the jet has been transformed into mechanical energy, so the efficiency is high.

A practical impulse turbine was invented by Lester A. Pelton (1829-1908) in California around 1870. There were high-pressure jets there used in placer mining, and a primitive turbine called hurdy-gurdy, a mere rotating platform with vanes, had been used since the `60's, driven by such jets. Pelton also invented the split bucket now universally used, in 1880. Pelton is a trade name for the products of the company he originated, but the term is now used generically for all similar impulse turbines. [Calvert J.B (2003).]

A diagram of a Pelton wheel is shown below. The wheel of pitch diameter D has buckets around its periphery, so spaced that the jet always strikes more than one at a time. The buckets have the form shown at the upper left, where the water enters at a splitter and is diverted to each side, where the velocity is smoothly reversed. [Calvert J.B (2003).]

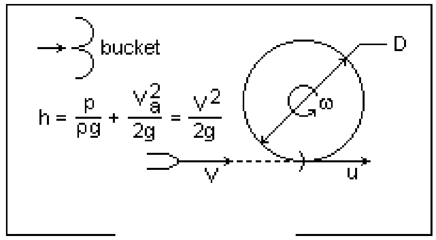


Fig. 2.5 Impulse Turbine

The conduit bringing high-pressure water to the impulse wheel is called the penstock. This was strictly just the name of the valve, but the term has been extended to the conduit and its appurtenances as well, and is a general term for a water passage and control that is under pressure, whether it serves an impulse turbine or not. [Calvert J.B (2003).]

2.3.3 Reaction turbine

By contrast with the impulse turbine, reaction turbines are difficult to understand and analyze, especially the ones usually met with in practice. The modest lawn sprinkler comes to our aid, since it is both a reaction turbine, and easy to understand. It will be our introduction to reaction turbines. In the impulse turbine, the pressure change occurred in the nozzle, where pressure head was converted into kinetic energy. There was no pressure change in the rotor, which had the sole duty of turning momentum change into torque. In the reaction turbine, the pressure change occurs in the rotor itself at the same time that the force is exerted. The force still comes from rate of change of momentum, but not as obviously as in the impulse turbine. In 1826, Benoit Fourneyron (1802-1867) developed an outward-flow turbine that was efficient, but the mechanical arrangements were poor, since the rotor was on the outside. Jean V. Poncelet (1788-1867) designed an inward-flow turbine in about

1820. S.B. Howd took the design to the U.S. and patented it in 1838. In 1849, James B. Francis (1815-1892) improved Howd's turbine, where the water entered horizontally through guide Vanes that gave them a whirl, then passed into the rotor and was diverted downwards. His hydraulic experiments on turbines at Lowell, Massachusetts, are famous. James Thomson of Belfast, brother of Lord Kelvin, made important improvements to the inward radial flow turbine, in the shape of Vanes, control. The rotor of a Francis turbine is illustrated below. Its basic dimension is the diameter D. The shape of the Vanes cannot be well represented, but they are designed for smooth flow at the design speed and head of the turbine. [Calvert J.B (2003).]

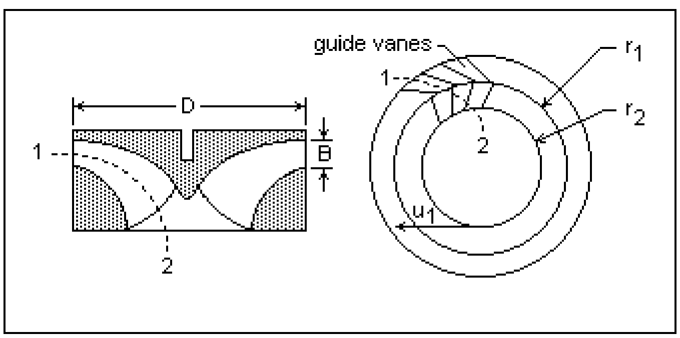


Fig. 2.6 Francis Rotor

The plan view at the right shows how the guide vanes in the stator direct the water onto the moving rotor, acting like nozzles.

The water follows the dotted path in space from the inlet at 1 to the outlet at 2. Relative to the rotor, it flows parallel to the vanes, exerting the force that creates the output torque. In this diagram, it is easy to imagine the velocity triangles at input and output, which will be similar to those for the lawn sprinkler. The vertical section at the left shows that the flow is not completely radial, as it was in the earliest Francis turbines. During its passage through the rotor, the water is diverted axially, and exits at the bottom of the rotor. This, of course, complicates our analysis, but nothing is really fundamentally changed. The mixed flow allows a more efficient turbine by making the exit smoother [Calvert J.B (2003).]

.

2.3.4 Axial flow

Hydraulic turbines can be made that are almost completely axial flow; the rotor taking the form of vanes perpendicular to the axis is the propeller turbines. An example is the Kaplan turbine, invented by Victor Kaplan (1876-1934) and first put into service in 1912-13, with movable blades that rotate, or “feather”, to handle different conditions, the key to making an efficient propeller turbine. A turbine without such adjustments will work efficiently only at its design speed and head. The water is given a swirl at the top of a Kaplan turbine that is taken out by the propeller. [Calvert J.B (2003).]

2.4 Construction of outward radial-flow reaction water turbine

Umoren, I.H (2004) designed an outward radial-flow reaction water turbine for the purpose of driving a 6W electric dynamo. The turbine-dynamo combination is also expected to serve as a water sprinkler to be driven by the water mains pressure. The device is to generate electricity for lightening the places they are to be used whenever there is sufficient water pressure. Umoren's project was limited to the design of reaction water turbine.

My contribution to this work was therefore to review the design of the water turbine, construct it fully and test it as was the aim of the work. The water sprinkler is a good example of a reaction water turbine whose energy output served only to move the sprinkler head.

CHAPTER 3 GENERAL DESIGN THEORY AND CALCULATIONS

3.1 The Euler Turbine Equation

The real flow through a rotor is three dimensional, that is to say the velocity of the fluid is a function of three positional coordinates, say, in the cylindrical system, r, θ and z, as shown in Fig. 3.1. Thus, there is a variation of velocity not only along the radius but also across the blade passage in any plane parallel to the rotor rotation, say from the upper side of one blade to the underside of the adjacent blade, which constitutes an abrupt change – a discontinuity. Also, there is variation of velocity in the meridional plane, i.e. along the axis of the rotor. The velocity distribution is, therefore, very complex and dependent upon the number of blades, their shapes and thicknesses, as well as on the width of the rotor and its variation with radius.

The one-dimensional theory simplifies the problem very considerably with the following assumptions.

(i) The blades are infinitely thin and the pressure difference across them is replaced by imaginary body forces acting on the fluid and producing torque.

(ii) The number of blades is infinitely large, so that the variation of velocity across blade passages is reduced and tends to zero. This assumption is equivalent to stipulating axisymmetrical flow, in which there is perfect symmetry with regard to the axis of rotor rotation. Thus, $\frac{\delta V}{\delta \theta} = 0$

(iii) Over that part of the rotor where transfer of energy takes place (blade passages) there is no variation of velocity in the meridional plane, i.e. across the width of the rotor. Thus,

$$\frac{\delta V}{\delta z} = 0$$

The result of these assumptions is that whereas, in reality,

$$V = V(r, \theta, z),$$

For the one-dimensional flow

$$V_\infty = V(r)$$

Note, that the suffix ∞ stipulates the assumption of an infinite number of blades and, hence, axisymmetry.

As a result, the flow through, say, a centrifugal rotor may be represented by a diagram such as Fig. 3.2. Although finite blades are shown, they are not taken into account in the theory. Furthermore assumption (ii) implies that the fluid streamlines are confined to infinitely narrow inter blade passages, and, hence, their paths are congruent with the shape of the inter blade centerline, shown by a chain-dotted line. Thus, the flow of fluid through a rotor passage may be regarded as a flow of fluid particles along the centerline of the inter blade passage.

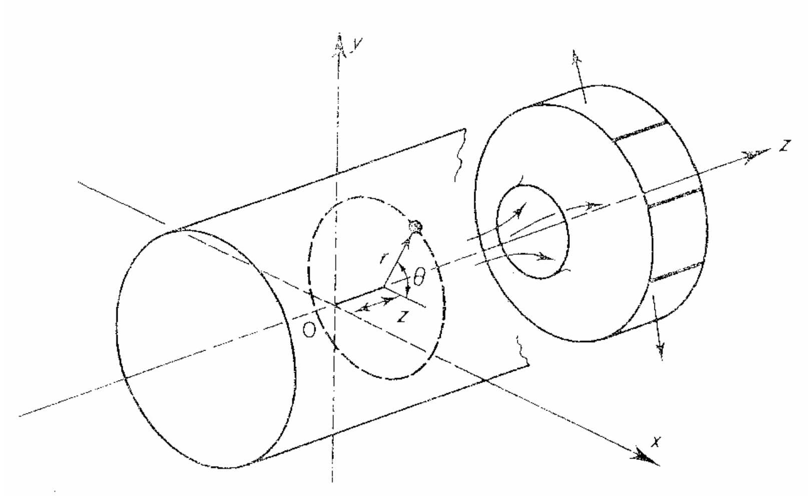


Fig. 3.1 A Centrifugal rotor

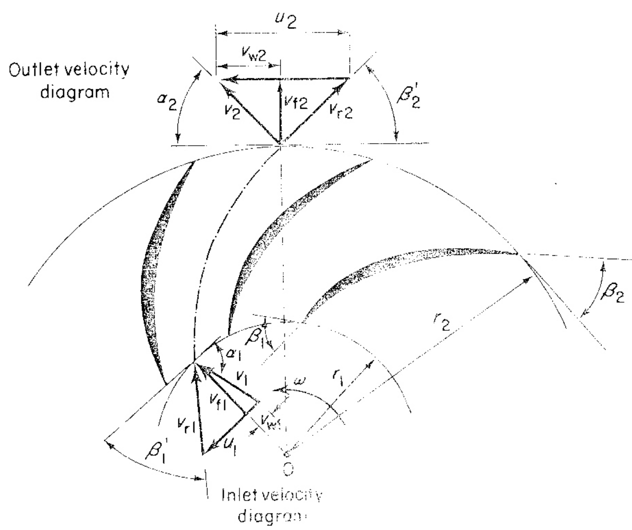


Fig. 3.2 One-dimensional flow through a centrifugal rotor

The assumptions of the theory enable us to limit our analysis to changes of conditions which occur between rotor inlet and rotor outlet without reference to the space in between, where the real transfer of energy takes place. The space is treated as a ‘black box’ having an input in the form of an inlet velocity triangle and an output in the form of the outlet velocity triangle. [Douglas et al (1985)] Velocity triangles for a centrifugal rotor rotating with a constant angular velocity ω, is shown in Fig. 3.2. At inlet, the fluid moving with an absolute velocity V_1 enters the rotor through a cylindrical surface of radius r_1 and may make an angle α_1 with the tangent at that radius. At outlet, the fluid leaves the rotor through a cylindrical surface of radius r_2, absolute velocity V_2 inclined to the tangent at outlet by the angle α_2.

The general expression for the energy transfer between the rotor and the fluid, based on the one-dimensional theory and usually referred to as Euler’s turbine equation, may be now derived as follows: From Newton’s second law applied to angular motion,

Torque = Rate of change of angular momentum,

Now, Angular momentum = (Mass) (Tangential Velocity) x (Radius)

Therefore,

Angular momentum entering the rotor per second $= \dot{m} V_{w_1} r_1$,

Angular momentum leaving the rotor per second $= \dot{m} V_{w_2} r_2$,

Therefore, Rate of change of angular momentum = $\dot{m} V_{w_2} r_2$ - $\dot{m} V_{w_1} r_1$,

So that, Torque transmitted $= \dot{m}\left(V_{w_2} r_2 - V_{w_1} r_1\right)$.

Since the work done in unit time is given by the product of torque and angular velocity,

Work done per second = $(Torque)\omega = \dot{m}\left(V_{w_2} r_2 - V_{w_1} r_1\right)\omega$

$U = \omega r$, $\omega = \dfrac{U}{r}, \omega r_2 = U_2, \omega r_1 = U_1$. Hence, on substitution,

Work done per second, $E_t = \dot{m}\left(U_2 V_{w_2} - U_1 V_{w_1}\right)$ ……………… 3.1

The SI units of the above expression are joules per second or watts.

Since the work done per second by the rotor on the fluid, such as in this case, is the rate of energy transfer, then

Rate of energy transfer/unit mass of fluid flowing, $Y = gE = \dfrac{E_t}{\dot{m}}$

The product $gE = Y$ known as specific energy is of significance in the case of pumps and fans. The units of Y are joules per kilograms.

From the specific energy, the Euler's head E is given by:

$$E = \frac{1}{g}\left(U_2 V_{w_2} - U_1 V_{w_1}\right) \quad …….. \quad 3.2$$

The units of this equation are joules per kilograms divided by m/s^2. This, of course, simplifies to meters and, therefore, is the same as all the terms of Bernoulli's equation and, consequently, E may be used in conjunction with it. Equation (3.2) is known as Euler's equation. From its mode of derivation it is apparent that Euler's equation applies to a pump (as derived) and to a turbine. In the latter case, however, since $U_1 V_{w_1} > U_2 V_{w_2}$, E would be negative, indicating the reversed direction of energy transfer. It is, therefore, common for a turbine to use the reversed order of terms in the brackets to yield positive E. Since the units of E reduce to meters of the fluid handled, it is often referred to as Euler's head and, in the case of pumps or fans; it represents the ideal theoretical head developed H_{th}.

It is useful to express Euler's head in terms of the absolute fluid velocities rather than their components. From velocity triangles of fig. 3.2,

$$V_{w_1} = V_1 \cos\alpha_1, V_{w_2} = V_2 \cos\alpha_2$$

So that $E = \frac{1}{g}\left(U_2 V_2 \cos\alpha_2 - U_1 V_1 \cos\alpha_1\right)$ ……. 3.3

But, using the cosine rule,

$$V_{r_1}^2 = U_1^2 + V_1^2 - 2U_1 V_1 \cos\alpha_1,$$

So that $U_1 V_1 \cos\alpha_1 = \frac{1}{2}\left(U_1^2 - V_{r_1}^2 + V_1^2\right)$

Similarly, $U_2 V_2 \cos\alpha_2 = \frac{1}{2}\left(U_2^2 - V_{r_2}^2 + V_2^2\right)$

Substituting into 3.3

We get $E = \frac{1}{2g}\left(U_2^2 - U_1^2 + V_2^2 - V_1^2 + V_{r_1}^2 - V_{r_2}^2\right)$

And for a turbine, since $U_1 V_{w_1} > U_2 V_{w_2}$

$$E = \frac{1}{2g}\left(V_1^2 - V_2^2\right) + \frac{1}{2g}\left(U_1^2 - U_2^2\right) + \frac{1}{2g}\left(V_{r_2}^2 - V_{r_1}^2\right) \quad \text{……… 3.4}$$

In this expression, the first term denotes the increase of the kinetic energy of the fluid in the turbine or dynamic head or pressure. The second term represents the energy used in setting the fluid into a circular motion about the rotor axis (forced vortex) which gives rise to pressure difference. The third term is the regain of static head or pressure due to reduction of relative velocity in the fluid passing through the rotor. The 2nd and 3rd terms represent a change in static head or pressure. [Shepherd (1956)]

In the radial flow, the velocity triangles are as shown in Fig. 3.2 and, in addition, the following relationship hold. Since, in general, $U = \omega r$, it follows that the tangential blade velocities at inlet and outlet are given by:

$$U_1 = \omega r_1, U_2 = \omega r_2 \text{ .... } 3.5$$

Since the flow at inlet and outlet is through cylindrical surfaces and the velocity components normal to them are V_{f_1} and V_{f_2} , the continuity equation applied to inlet and outlet for the mass flow $\dot{m}$

$$\dot{m} = \rho_1 A_1 V_{f_1} = \rho_2 A_2 V_{f_2} \text{ .... } 3.6$$

For incompressible flow, equation 3.6 Simplifies to

$$A_1 V_{f_1} = A_2 V_{f_2} \text{ .... } 3.7$$

Now, assuming that m, ω , r_1 and r_2 are known, the following arguments are usually employed in order to draw the velocity triangles.

At inlet the usual assumptions are as follows:

(i) The absolute velocity is radial, therefore $V_1 = V_{f_1}, V_{w_1} = 0$,

Hence, V_1 is calculated from equation 3.6 and $\alpha_1 = 90^0$. If this condition does not apply, which only occurs if there is a prewhirl (V_{w_1}) component present, perhaps due to inlet vanes or unfavorable inlet conditions, then V_{f_1} is calculated from equation (3.6) and α_1 can be determined only if V_{w_1} is known.

(ii) The blade angle at inlet β_1 is such that the blade meets the relative velocity tangentially. Thus, $\beta_1 = \beta_1^{'}$ this assumption is known as the 'no shock' condition and is applied in determining the blade inlet angle during design in order to minimize the entry loss.

Thus, the inlet triangles may be drawn.

For the outlet triangles, it is assumed that the fluid leaves the rotor with a relative velocity tangential to the blade at outlet. Thus, $\beta_2^{'} = \beta_2$ and, in order to draw the outlet velocity triangles, β_2 must be known. The direction of V_{r_2} is then drawn, as well as the V_{f_2} vector, which is radial and whose magnitude is calculated from equation (3.6). It is, thus, possible to draw the $U_2 (= \omega r_2)$ vector perpendicular to V_{f_2} and starting from the intersection with the direction of V_{r_2}. The absolute velocity V_2 is then obtained by completing the triangle, from which,

$$\cot \beta_2 = {(U_2 - V_{w_2})}\Big/{V_{f_2}} ,$$

So that $V_{w_2} = U_2 - V_{f_2} \cot \beta_2$.

Substituting this into Euler's equation, and remembering that $V_{w_1} = 0$, the following expression is obtained:

$$E = \left(\frac{U_2}{g}\right)(U_2 - V_{f_2} \cot \beta_2) \quad \ldots\ldots\ldots \, 3.8$$

The total amount of energy transferred by the rotor is, thus,

$$E_t = \dot{m}\, gE = \dot{m} U_2 (U_2 - V_{f_2} \cot \beta_2) \quad \ldots\ldots.. \, 3.9$$

3.2 Departures from Euler's Theory and Losses

There are two fundamental reasons why the actual energy transfer achieved by a hydraulic machine is smaller than that predicted by Euler's equation. The first reason is that, the velocities in the blade passages and at the rotor outlets are not uniform due to the presence of blades and the real flow being three – dimensional. This results in a diminished velocity of whirl component and, hence, reduces the Euler's head. This effect is not caused by friction and, therefore, does not represent a loss but follows from ideal flow analysis of pressure and velocity distributions. The second reason is that in a real rotor there are losses of energy due to friction, separation and wakes associated with the development of boundary layers.

Finally, there are mechanical losses of energy such as in the bearings and sealing glands, which must be accounted for. It is normal practice in hydraulic machines to include within this category losses due to disc friction, sometimes referred to as "windage" loss. This is the power required to spin the rotor at the required velocity without any work being done by the rotor or on the rotor by the fluid. This would be possible only if the rotor did not have any blades. Thus, windage loss accounts for the friction between the outer surfaces of the rotor rotating in the fluid surrounding it within the casing.

3.3 Degree of Reaction

A parameter which describes the reaction turbines is the degree of reaction or reaction. It is derived by the application of Bernoulli's equation to the inlet and outlet of a turbine, assuming ideal flow (no losses). Thus, if the conditions at inlet are denoted by the use of suffix 1 and those at outlet by suffix 2, then,

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}=E+\frac{P_2}{\rho g}+\frac{V_2^2}{2g},$$

Where E is the energy transferred by the fluid to the turbine per unit weight of the fluid. Thus,

$$E=\frac{(P_1-P_2)}{\rho g}+\frac{(V_1^2-V_2^2)}{2g}$$

In this equation, the first term on the right-hand side represents the drop of static pressure in the fluid across the turbine, whereas the second term represents the drop in the velocity head. The two extreme solutions are obtained by making either of these two terms equal to zero. Thus, if the pressure is constant, so that $P_1 = P_2$, then $E=\frac{(V_1^2-V_2^2)}{2g}$, and such a turbine is purely impulsive. If on the other hand, $V_1 = V_2$ then $E=\frac{(P_1-P_2)}{\rho g}$ and this represents pure reaction. The intermediate possibilities are described by the degree of reaction (R), define as

$$R=\frac{\text{Static Pressure Drop}}{\text{Total Energy Transfer}} \quad \ldots\ldots\ldots \quad 3.10$$

But the static pressure drop is given by $\frac{(P_1-P_2)}{\rho g}=E-\frac{(V_1^2-V_2^2)}{2g}$,

So that $R=\frac{\left\{E-\frac{(V_1^2-V_2^2)}{2g}\right\}}{E}=1-\frac{(V_1^2-V_2^2)}{2gE}$

Substituting now from Euler's equation for $E=\frac{V_{w_1}U_1}{g}$, we obtain

$$R=1-\frac{(V_1^2-V_2^2)}{2V_{w_1}U_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 3.11$$

From Euler turbine equation

$$R = \frac{\left(U_1^2 - U_2^2\right) + \left(V_{r_2}^2 - V_{r_1}^2\right)}{\left(V_1^2 - V_2^2\right) + \left(U_1^2 - U_2^2\right) + \left(V_{r_2}^2 - V_{r_1}^2\right)}$$

A machine with any degree of reaction must have the rotor enclosed in order that the fluid cannot expand freely in all directions [Shepherd (1956)]. A machine with reaction is exemplified by the lawn sprinkler, in which water issues at high velocity from the rotor in a tangential direction. The feature of the rotor is that the water enters at a high pressure or under high head and the pressure energy is transformed into velocity energy in a nozzle which is part of the rotor itself. In the impulse turbine, the nozzle is stationary and its function is only to transform pressure energy to kinetic energy, this kinetic energy being transferred to the rotor by pure impulse action. The change of momentum of the fluid in the nozzle gives rise to a reaction force but as the nozzle is stationary, no energy is transferred by it. While in reaction turbine, the nozzle is part of the rotor and is free to move, although constrained to a circular path, and it rotates due to the reaction force caused by the change of momentum.

Many machines are part impulse and part reaction, and for exact classification, the amount or degree of reaction should be explicitly stated. In general, however, the term reaction alone is used whenever the machine is not purely impulse, but for the special case of steam turbines it has come to imply 50% or half-degree reaction. In radial-flow machines, the centrifugal action is a very important factor in the degree of reaction, sometimes predominantly so, but it is more difficult to demonstrate the above effects in simple fashion. However, the reaction can always be calculated from equation (3.10) even if the physical means cannot always be clearly elucidated. It will now be appreciated how a certain amount of reaction in a given type of machine immediately gives some important information on the form of machine, such as open or closed rotor and shape of flow passages.

3.4 Efficiency and Utilization Factor

There are a great many 'efficiencies" of turbo machines. The definition of efficiency is the ratio of useful energy delivered to the energy supplied, and here it is necessary only to refine this with respect to the point of application. Two efficiencies are considered, a hydraulic efficiency η_h between fluid and rotor and an overall efficiency η between fluid and shaft. The difference between the two represents the energy absorbed by bearings, glands, couplings etc., or, in general, by purely mechanical effects which occur between the rotor and the point of actual power input or output.

For a turbine

$$\eta_{hydraulic} = \eta_h = \frac{\text{Mechanical energy supplied by rotor}}{\text{Hydrodynam ic energy available from fluid}} \quad \text{.......... 3.12}$$

$$\eta_{overall} = \eta = \frac{\text{Mechanical energy in output shaft at coupling}}{\text{Hydrodynamic energy available from fluid}} \quad \text{............... 3.13}$$

The ratio of rotor and shaft energy is represented by the mechanical efficiency η_m

So $$\eta_m = \frac{\eta}{\eta_h} \quad \text{.................................... 3.14}$$

Turbines are those types of turbo machines which are designed to convert fluid energy into mechanical energy and the major performance factor of interest is the ratio of the actual shaft work produced to the energy supplied. Overall efficiency η is the product of the mechanical efficiency η_m and the hydraulic efficiency η_h. For a turbine however, the hydraulic efficiency is again the product of two 'efficiencies", one relating purely to the geometrical arrangement of the ideal blading as it affects energy transfer and the other relating to the hydraulic behavior of the blading as affected by the real fluid effects of friction and eddies. Even for an ideal fluid, not all the energy supplied can be converted into useful work, because there must be some residual discharge kinetic energy due to the finite exit velocity, this energy being wasted as far as the rotor is concerned. The ratio of ideal work to energy supplied is sometimes called the "diagram" efficiency but here will be termed 'utilization factor' and given the symbol ε, as it is preferred to restrict efficiency to cases involving a dissipation of energy due to viscous fluid effects and mechanical friction. The value of the utilization factor may be found from the ideal velocity diagrams and the energy equations. The energy available to the rotor is the sum of the absolute inlet Kinetic energy and the energy available from pressure drop in the rotor (reaction); thus;

$$E_{available} = \frac{1}{2g}\left[V_1^2 + \left(U_1^2 - U_2^2\right) + \left(V_{r_2}^2 - V_{r_1}^2\right)\right]_{id}$$

The energy utilized by the rotor in the absence of fluid friction is

$$E_{utilized} = \frac{1}{g}\left[U_2 V_{w_2} - U_1 V_{w_1}\right]_{id}$$

$$= \frac{1}{2g}\left[\left(V_1^2 - V_2^2\right) + \left(U_1^2 - U_2^2\right) + \left(V_{r_2}^2 - V_{r_1}^2\right)\right]_{id} \quad \text{............... 3.15}$$

Hence

$$\text{Utilization factor,} \quad \varepsilon = \frac{2\left(U_2 V_{w_2} - U_1 V_{w_1}\right)_{id}}{\left[V_1^2 + \left(U_1^2 - U_2^2\right) + \left(V_{r_2}^2 - V_{r_1}^2\right)\right]_{id}} \quad \text{................ 3.16}$$

This can also be expressed as

$$\varepsilon = \frac{E_{id}}{E_{id} + \frac{V_2^2}{2g}} \quad \text{.................................. 3.17}$$

The ideal values are those obtained from the ideal velocity diagrams. If now the actual energy transferred to the rotor is taken for the numerator and the ideal energy transferred as the denominator, there results a vane efficiency (η_{vane}), thus

$$\eta_{vane} = \frac{\left(U_1 V_{w_1} - U_2 V_{w_2}\right)_{actual}}{\left(U_1 V_{w_1} - U_2 V_{w_2}\right)_{ideal}} \quad \text{......................} \ 3.18$$

Finally the product of ε and η_{vane} gives another efficiency, which is the ratio of the actual energy transferred to the rotor in the presence of fluid friction, etc., to the energy available to the rotor, the hydraulic efficiency η_h.

3.5 Design Theory of the Radial Outward Flow Reaction Water Turbine (Lawn Sprinkler)

3.5.1 Introduction

Water is fed from below to balance the weight of the rotor and reduce friction. The flow of water in a lawn sprinkler is radially outward. Water under pressure is introduced at the center, and jets of water that can cover the area necessary, issue from the ends of the arms at zero gauge pressure. The pressure decrease occurs in the sprinkler arms. Though the water is projected at an angle to the radius, the water from an operating sprinkler moves almost along a radius. The force on the rotor must act in reaction to the creation of the momentum, which is, of course, the origin of the name of the reaction turbine.

A two-armed rotor of a rotating lawn sprinkler is shown below.

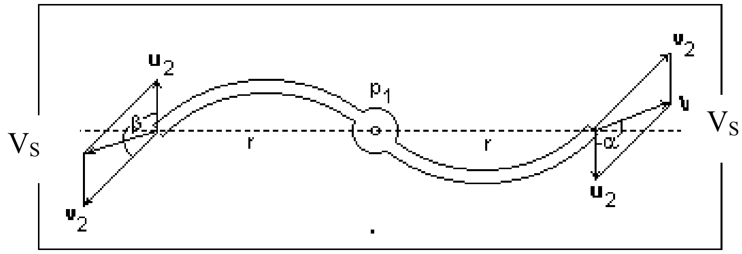


Fig. 3.3 Two-armed Lawn Sprinkler

Conditions at the ends of the two arms are the same. The jet at the end of an arm is projected at an angle β with a perpendicular normal to the radius from the center of rotation, in the direction of the rotational velocity $U_2 = \omega r$. The space velocity V_s, is the sum of V_2 and U_2, V_s makes an angle α_2 with U_2. When the rotor is stopped, $V_s = V_2$. As the rotor speeds up, V_s moves closer to a radial direction. When it reaches the radial direction, there is no longer a component normal to the radius and, therefore, no accelerating torque. It is easy to see that the torque will be a maximum when the rotor is stalled. To find V_2 in terms of P_1, we shall use Bernoulli's theorem. However, energy is not conserved between the axial point 1 and point 2 at the end of the arm, since the water does work in passing from one point to the other. There is a reaction force of magnitude ρV_2 in the opposite direction to V_2. The movement of point 2 is in the direction of U_2, so the rate of doing work is $\rho V_2 U_2 \cos\alpha$. Dividing by ρg to express this work as head, we find that a head of $\frac{V_2 U_2 \cos\alpha}{g}$ must be subtracted from the difference of the heads at point 2 and 1. Since $h_2 = h_1$ because the arms are lying horizontally, and $V_1 = 0$ at point 1, and $P = 0$ at point 2 (we are using gauge pressures), we get.

$$\frac{V_2^2}{2g} = \frac{P_1}{\rho g} - \frac{V_2 U_2 \cos\alpha}{g}$$

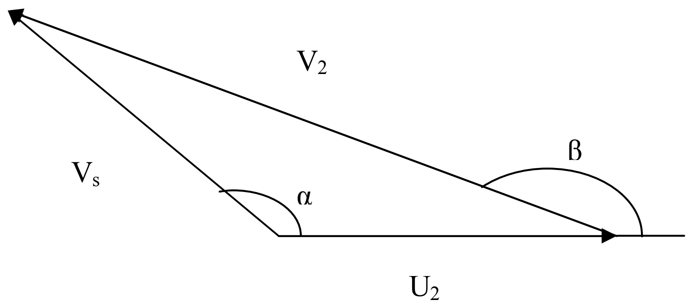


Fig. 3.4 Vector Triangle

From the vector triangles, we find that (dropping the subscript 2 for the moment) $V_s^2 = U^2 + V^2 - 2UV\cos(180 - \beta) = U^2 + V^2 + 2UV\cos\beta$ and also that $V_s \cos\alpha = V\cos\beta + U$. Substituting for V_s^2 and $V_s^2 \cos\alpha$ in the above equation, we find that $\frac{(V^2 - U^2)}{2g} = \frac{P_1}{\rho g} = h$, which simplifies to

$$V^2 = 2gh + U^2 \quad \text{………………... 3.19}$$

In this equation, h is the supply head, which may include approach velocity if it is to be considered.

Now, we can find everything we need as a function of U_2, or of the angular velocity ω. In particular the component of V perpendicular to the radius is $V' = V_s \cos\alpha = V\cos\beta + U$. The corresponding reaction force is obtained by multiplying by ρ, and by the total discharge AV, where A is the total area of the jets on all arms. The torque then, is

$$T = -\rho AVr(V\cos\beta + U) \quad \text{……………. 3.20}$$

Where $V = \sqrt{2gh + U^2}$ and $U = \omega r$.

If we are more interested in power than watering, β could be made 180^{o}, and the area of the jets could be increased, partly by multiplying the number of jets. [Daugherty et al (1965)]. If the angular velocity of the rotor could be such that, $V = U$, the water would drop directly down, and the efficiency of the turbine would be a maximum. However, we must have $V^2 = 2gh + U^2$, so this condition cannot exist. All that can be done is to make U as large as possible, but this is not very satisfactory. This is the reason Baker's mill are not often seen these days. [Daugherty et al (1965)]

In the hydraulic field, the rotating lawn sprinkler is an elementary reaction turbine. The ideal maximum, or runaway, speed is when T = 0, and this will be the case when $U_2 = -V_2 \cos\beta_2$ and when $V_2 \cos\alpha_2 = 0$ or $\alpha_2 = 90^0$. Because of mechanical friction, this condition will never be realized. Of the total power supplied to the sprinkler, the greater part is lost in the kinetic energy of the jets. The total power developed by the sprinkler is used in overcoming its own friction. If there were more arms, with larger orifices, so as to discharge more water, there could be a surplus of power which would be useful power delivered. A primitive turbine constructed in this manner was the Barker's mill. A continuing increase in the number of arms terminated in a complete wheel with passages separated by vanes, but the device was not very efficient, until in 1826, a Frenchman by the name of Fourneyron added stationary guide vanes in the central portion. These guide vanes gave the water a definite tangential component, thereby imparting angular momentum to the fluid entering the rotor. This outward-flow turbine was efficient, but the mechanical construction was not good because the rotating element was on the outside with the fixed guide vanes near the axis. [Daugherty et al (1965)].

The inward-flow turbine permits a better mechanical construction since the rotor and shaft form a compact unit in the center, while the stationary guide vanes are on the outside. Several crude inward-flow turbines were constructed around 1838, but the first to be well designed was built in 1849 by the eminent engineer James B. Francis, who made an accurate test of this turbine. [Daugherty et al (1965)]

3.5.2 Determination of design angular speed of the turbine

Since the turbine is intended to drive a bicycle dynamo, an experiment was carried out to determine the speed of the dynamo that gave maximum power. The speed obtained was used as the design speed of the turbine.

The results obtained from the experiment were as follows:

The maximum angular speed of the output shaft of the electric motor, $N_m = 2000rpm$

The diameter of the pinion on the shaft of electric motor, $D_m = 0.0355\text{m}$

The diameter of the pinion of the dynamo, $D_d = 0.0224\text{m}$

Let $N_d =$ the maximum angular speed of the dynamo

The tangential velocities of these pinions in contact are the same. This implies that,

$$N_m D_m = N_d D_d$$

Therefore, $$N_d = \frac{N_m D_m}{D_d} \qquad \text{…………………… 3.21}$$

Let D_T = Diameter of pinion of the output shaft of the turbine.

N_T = Angular speed of the turbine

Also, the tangential velocity of the pinion of the dynamo and that of the output shaft (when the two are in mesh) are the same.

Therefore $D_T N_T = N_d D_d$

This gives,

$$N_T = \frac{N_d D_d}{D_T} \qquad \text{………………... 3.22}$$

Equation (3.28) gives the maximum design angular speed of the simple reaction turbine.

3.5.3 Determination of rotor arm diameter

Let A_{in} = the area of the mains inlet pipe

V_{in} = the velocity of fluid in the main inlet pipe.

The number of rotor arms are initially taken as four (4)

The volume flow rate is constant

Therefore,

$$A_{in} V_{in} = 4 A_1 V_1$$

Where, V_1 = velocity at inlet to one of the arms

Assuming $A_1 = \frac{A_{in}}{4}$, then $A_{in} = 4A_1$

Therefore, $\frac{\pi d_{in}^2}{4} = 4 \times \frac{\pi d_1^2}{4}$

This gives $d_1 = \sqrt{\frac{d_{in}^2}{4}}$

$$d_1 = \frac{d_{in}}{2} \qquad \text{……………. 3.23}$$

This implies that the rotor arm diameter must be half the main inlet pipe diameter

Also, from Volume flow rate,

$$Q = A_{in}V_{in} = 4A_1V_1$$

But, $A_{in} = 4A_1$

This gives $V_{in} = V_1$ ………….. 3.24

This implies that the velocity of the fluid in the main inlet pipe is the same with that in the rotor arm

And $V_{in} = \frac{Q}{A_{in}}$ ………… 3.25

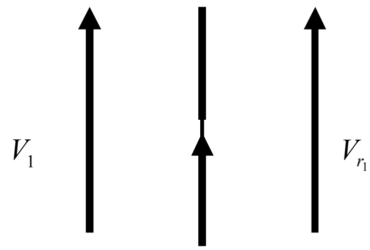
3.5.4 Work Done on the arm by the fluid

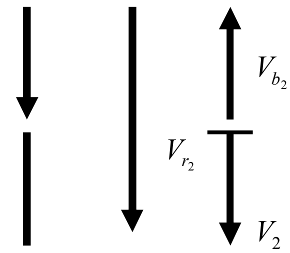
From a one-dimensional radial flow turbine in Fig. 3.1

For outward radial flow, there is no whirl velocity component at the inlet

I.e. $V_{w_1} = 0$ and $\alpha_1 = 90^0$

For a simple reaction turbine in the form of liquid sprinkler shown above





$V_1 = V_{r_1}$

Fig. 3.5 (a): Inlet velocity diagram $(r = 0)$ **Fig.3.5 (b): Exit velocity diagram** $(r = R)$

At the inlet, $U = r_1\omega$

But $r_1 = 0$

Therefore $U_1 = 0$

Hence, $V_{r_1} = V_1$

At the exit, $\alpha_2 = 0$

And, $V_{r_2} = U_2 - (-V_2)$

Therefore, $V_{r_2} = U_2 + V_2$ ……………. 3.26

From equation 3.2, the work done on the arm by the fluid is given by,

$$Y = U_1 V_{w_1} - U_2 V_{w_2}$$

But $V_{w_1} = 0$

Therefore $U_1 V_{w_1} = 0$

Equation 3.2 becomes,

$$Y = -U_2 V_{w_2} \quad \text{……………… 3.27}$$

From the general exit velocity diagram

$$V_2 \cos\alpha_2 = V_{w_2}$$

But, $\alpha_2 = 0$

Therefore, $V_{w_2} = V_2$

And V_2 is in the negative direction

Hence, $Y = -U_2(-V_2)$

Thus, $Y = U_2 V_2$ ………….. 3.28

This implies that work done on the arm by the fluid, Y, increases as the exit velocity, V_2 increases. To increase the exit velocity, a converging nozzle is used. This is because the working fluid is water, which is incompressible. If the working fluid is compressible e.g. gas, steam; then a converging – diverging nozzle will be used.

3.5.5 Determination of tangential force

The following quantities are measurable

Q = volume flow rate (m^3/s) in the turbine

d_2 = diameter of the nozzle at exit (m)

r = distance between the axis of nozzle and axis of rotation of the rotor (m)

N_T = rotational speed of the rotor $(r.p.m)$

For continuity, $Q = 4A_2V_2 = 4A_1V_1$

Where A_2 = area of the nozzle at exit (m^2)

V_2 = exit velocity through the nozzle (m/s)

But $A_2 = \dfrac{\pi d_2^2}{4}$ ………………….. 3.29

Therefore,

$$Q = 4\frac{\pi d_2^2}{4}V_2 = 4\frac{\pi d_1^2}{4}V_1$$

And $V_2 = \dfrac{Q}{\pi d_2^2}$ …………………… 3.30

Or $V_2 = \left(\dfrac{d_1}{d_2}\right)^2 V_1$ …………………… 3.31

Equations (3.30) and 3.31) give the exit velocity of a single jet,

Also $U_2 = r\omega$

Where ω = angular speed in rev/s

And $\omega = \dfrac{2\pi N_T}{60}$

Therefore, $U_2 = \dfrac{2\pi N_T r}{60}$ ………….. 3.32

Equation (3.32) gives the tangential velocity at exit

But the work done per unit mass of flow per arm was obtained as,

$$Y = V_2 U_2$$

Substituting equations (3.30) and (3.32) gives work done per kg per arm as

$$Y = \frac{Q}{\pi d_2^2} \times \frac{2\pi N_T r}{60}$$

$Y = \dfrac{r N_T Q}{30 d_2^2}\left(J/kg\right)$ …………………. 3.33

Equation (3.33) gives the work done per unit mass of flow for a single jet. If the mass flow rate of fluid through the machine is $\dot{m}$, then the power developed by the turbine (theoretical output power) is given by,

$P_{oth} = 4Y\dot{m}$ ……………… 3.34

Since there are four jets,

But, $\dot{m} = \rho Q$ ………………. 3.35

Where ρ = density of the fluid

Substituting equations (3.33) and (3.35) into (3.34), gives

$$P_{oth} = \frac{2\rho r N_T Q^2}{15 d_2^2}(watts) \qquad \ldots\ldots\ldots\ldots\ldots 3.36$$

From equation (3.36), the volume flow rate through the turbine is given by,

$$Q = \sqrt{\frac{15 d_2^2 P_{oth}}{2\rho r N_T}}\left(m^3/s\right) \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad 3.37$$

The input water power into the turbine is given by,

$$P_{in} = \Delta P \times Q \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 3.38$$

Where ΔP = mains pressure

From equations (3.12), (3.36) and 3.38, the hydraulic efficiency can be expressed as,

$$\eta_h = \frac{2\rho r N_T Q}{15 d_2^2 \Delta P} \ldots\ldots\ldots \quad 3.39$$

Equations (3.38) and (3.39) give the significance of the mains pressure in the design. Recall from (3.13) and (3.14) that the mechanical and overall efficiencies are respectively given by: $\eta_m = \frac{P_{oA}}{P_{oth}}$ and $\eta_{ov} = \eta_m . \eta_h$

Where P_{OA} = actual output power

Substituting for hydraulic efficiency from equation (3.39) into overall efficiency gives;

$$\eta_{ov} = \frac{2 r N_T \rho \eta_m Q}{15 d_2^2 \Delta P} \qquad \ldots\ldots\ldots\ldots 3.40$$

Consequently, by using $\eta_{ov} = \eta_m . \eta_h$, gives

$$d_2 = \sqrt{\frac{2rN_T\rho}{15\eta_h}\left(\frac{Q}{\Delta P}\right)} \quad \text{………… 3.41}$$

The theoretical power developed can also be expressed as,

$$P_{oth} = T\omega \quad \text{………………. 3.42}$$

Where, T = applied torque (Nm)

But T = Fr

Where F = tangential force (N)

Therefore $P_{oth} = Fr\omega$

But $r\omega = U_2$ = tangential velocity at exit

And $P_{oth} = FU_2$

The tangential force is given as,

$$F = \frac{P_{oth}}{U_2} \quad \text{………… 3.43}$$

3.6 Tolerance

After determining the load, stresses and selecting the proper materials, it is necessary to have complete assembly and detail drawings to convey all necessary information to the Chop man. Tolerances or dimensional variation must be considered carefully, as an important phase of design. Tolerances must be placed on the dimensions of a drawing to limit the permissible variations in size because it is impossible to manufacture a part exactly to a given dimension. It is easy for the designer to specify a certain dimension say 10mm but for the workman or machine to turn it out exactly 10mm is quite another thing. In general, the closer the dimension must be to precisely 10mm which we will call basic size, the more costly will be the manufacture. Since the expense of manufacturing varies directly with the accuracy required, the tolerance is chosen to be as large as the design permits without affecting the function of the component.

The tolerance chosen is based on the fact that the dimensions of the components should permit interchangeability in case of replacement of components. For the designation of tolerances, the hole-basis system has been chosen. In this system of fit, the basic diameter of the hole is constant while the shaft size varies with different types of fit.

The British Standard BS4500, IS0 limits and fits, gives a selection of hole and shaft tolerances to cover a wide range of engineering applications. Therefore, the BS 4500 ISO limits and fits will be used in specifying the tolerances needed in this work.

With the basic functional requirement expected of the turbine, the suitable clearance fit required for this work is the closed fit (H7/g6) and will be used accordingly.

3.7 Calculations

S/No	Initial Data	Calculations	Results
1	$N_m = 2000rpm$ $D_m = 0.0355$m $D_d = 0.0224$m (from section3.5.2)	Maximum angular speed of dynamo from equation (3.21) $N_d = \frac{N_m D_m}{D_d} = \frac{2000 \times 0.0355}{0.0224}$	$N_d \approx 3170rpm$
2	$N_d = 3300rpm$ $D_d = 0.0224m$	Maximum angular speed of turbine from equation (3.22) $N_T = \frac{N_d D_d}{D_T}$	

S/No	Initial Data	Calculations	Results
	$D_T = 0.052\text{m}$ (chosen)	$N_T = \frac{3300 \times 0.0224}{0.052}$	$N_T = 1421 rpm$
3	$d_{in} = 0.0254m(chosen)$	Diameter of a single rotor arm From equation (3.23) $d_1 = \frac{d_{in}}{2} = \frac{0.0254}{2}$	$d_1 = 0.0127m$,Nozzle arm inlet diameter, See Drawing no: 11
4	$d_{in} = 0.0254m$	Area of the mains inlet pipe $A_{in} = \frac{\pi d_{in}^2}{4} = \frac{\pi \times 0.0254^2}{4}$	$A_{in} = 5.067 \times 10^{-4} m^2$
5	$\eta_m = 0.80(assumed)$ $P_{OA} = 5.5W$	Theoretical Output Power of the Turbine, From equation 3.14 $\eta_m = \frac{P_{OA}}{P_{oth}}$ $\therefore P_{oth} = \frac{P_{OA}}{\eta_m} = \frac{6}{0.8}$	$P_{oth} = 7.5W$
6	$P_{oth} = 7.5W$ $\eta_h = 0.6$ (assumed)	Input Power to the turbine From equation 3.12 $\eta_h = \frac{P_{oth}}{P_{in}}$ $P_{in} = \frac{P_{oth}}{\eta_h} = \frac{7.5}{0.6}$	$P_{in} = 12.5W$
7	$P_{in} = 12.5W$ $\Delta P = 3.7 \times 10^5 N/m^2$ (Greater than minimum pressure of $3.5 \times 10^5 N/m^2$)	Volume flow rate through the turbine From equation (3.38) $P_{in} = \Delta PQ$ $\therefore Q = \frac{P_{in}}{\Delta P}$ $Q = \frac{12.5}{3.7 \times 10^5}$	$Q = 3.38 \times 10^{-5} m^3/s$

S/No	Initial Data	Calculations	Results
8	$\eta_m = 0.8$ $\eta_h = 0.6$ $\Delta P = 3.7\times10^5 \, N/m^2$ $Q = 3.38\times10^{-5}\, m^3/s$ $N_T = 1421rpm$ $r = 0.1m$ $\rho = 1000\, kg/m^3$	Diameter of the nozzle From equation 3.41 $d_2 = \sqrt{\frac{2rN_T\rho}{15\eta_h}\left(\frac{Q}{\Delta P}\right)}$ $d_2 = \sqrt{\frac{2\times0.1\times1421\times1000\times3.38\times10^{-5}}{15\times0.6\times3.7\times10^5}}$	$d_2 \approx 0.002m$,Nozzle exit diameter, see Drawing no: 2
9	$r = 0.1m$ $N_T = 1421rpm$ $Q = 3.38\times10^{-5}\, m^3/s$ $d_2 = 0.002m$	Work Done by the turbine, Y_T For Single jet, and from equation 3.33 $Y = \frac{rN_TQ}{30d_2^2}$ $Y = \frac{0.1\times1421\times3.38\times10^{-5}}{30\times0.002^2}$ For four 4 jets $Y_T = 4Y = 4\times40$	$Y = 40\frac{J}{kg}$ $Y_T = 160\frac{J}{kg}$
10	$Q = 3.38\times10^{-5}\, m^3/s$	Exit velocity of a single jet From equation 3.30 $V_2 = \frac{Q}{\pi d_2^2}$ $V_2 = \frac{3.38\times10^{-5}}{\pi\times0.002^2}$	$V_2 = 2.66\, m/s$
11	$Q = 3.38\times10^{-5}\, m^3/s$ $A_{in} = 5.067\times10^{-4}\, m^2$	Velocity of fluid in the mains inlet pipe From equation 3.25 $V_{in} = \frac{Q}{A_{in}}$	

S/No	Initial Data	Calculations	Results
		$V_{in} = \frac{3.38 \times 10^{-5}}{5.067 \times 10^{-4}}$	$V_{in} = 0.067 \frac{m}{s}$
12	$V_{in} = 0.067 \frac{m}{s}$	Velocity of fluid in the rotor arms From equation 3.24 $V_1 = V_{in}$ $= 0.067 \frac{m}{s}$	$V_1 = 0.067 \frac{m}{s}$
13	$N_T = 1421 rpm$ $r = 0.1m$	Tangential velocity of fluid from jet From equation 3.32 $U_2 = \frac{2\pi N_T r}{60}$ $= \frac{2 \times \pi \times 1421 \times 0.1}{60}$	$U_2 = 14.88 \, m/s$
14	$U_2 = 14.88 \, m/s$ $P_{OA} = 6W$ $\eta_m = 0.8$	Tangential force From equation 3.43 and 3.14 $F = \frac{P_{oth}}{4U_2}$ $= \frac{P_{OA}}{4U_2\eta_m} \quad = \frac{6}{4 \times 14.88 \times 0.8}$	$F = 0.126N$

CHAPTER 4 CONSTRUCTION OF THE OUTWARD RADIAL FLOW REACTION WATER TURBINE

4.1 Introduction

Machine design consists of the application of scientific principles to the practical constructive act of engineering with the object of expressing original ideas in the form of drawing.

A designer is a person who solves problems. A good designer needs a wide knowledge of the subject of:

i) Applied mechanics to find forces, speeds and acceleration;

ii) Strength of materials to calculate dimension, Stiffness and stability;

iii) Metallurgy for selection of materials;

iv) Engineering manufacture to determine methods of casting, forging, and heat treatment;

v) A knowledge of economics and comparative cost;

vi) Engineering drawing;

vii) Machine tools.

4.2 Material Selection

In designing a machine member the selection of a material and manufacturing process by which it is to be made should be considered together.

Factors affecting the selection of materials are;

i) Availability and cost of materials,

ii) Strength and rigidity;

iii) Resistance to fatigue should be the basis for the design of members when subjected to cyclic loading;

iv) Toughness should be considered when the member is subjected to shock loads;

v) Weight should be considered very carefully.

Other factors which influence the selection of materials are electrical and thermal properties, resistance to wear and corrosion, ease of casting or forging, ease of machining etc.

Mild steel was used for the shaft and dynamo holder support, tripod stand, main inlet pipe, supply end, base plate, dynamo holder and brackets because it is readily available, it is relatively cheap, it can be machined and it is weld able. The outer part was painted to minimize corrosion. For the rotor output shaft, driving pinion and the nozzles aluminium alloy was used to meet the following requirements. It is relatively cheap, it is not heavy, it is readily available, machinable and it is not very corrosive.

Stainless steel was used for the rotor arms because of its relatively non corrosive nature, it could be machined and it was available.

4.3 Methods of Production

The following methods were used to fabricate the outward radial flow reaction turbine.

S/No	Components	Materials	Production methods	Equipment
1	Driving pinion. Drawing no: 6	Mild Steel rod	Cut, face and turn to size. Then drill through and slot the key way	Power saw machine, Lathe machine, drilling machine and radial drilling machine
2	Rotor. Drawing no:3	Aluminium alloy	Produce wooden pattern, mould it and then cast. Face to square the ends, then drill holes, bore hole for bearing sitting and finally turn to the given size	Lathe machine, radial drilling machine and milling machine
3	Dynamo bracket carrier. Drawing no:8	Mild Steel Plate	Cut with gas from the plate, grind the rough edges, turn to size, drill holes and then bore hole for bearing sitting. Finally tap some holes	Acetylene and gas welding equipment, off-hand grinding machine, Lathe machine, drilling machine and taps
4	Tripod Stand. Drawing no:4	Mild Steel Plate	Cut to size file, bend and then drill	Guillotine machine, hand files, bending machine and the drilling machine
5	Rotor arm. Drawing no: 11	Stainless Steel rod	Cut to size, face the ends, drill and then screw cut	Power hacksaw, lathe machine
6	Nozzles. Drawing no: 2	Aluminium alloy	Produce wooden Pattern, mould it and then cast. Mill to square to size, drill holes and then tap.	Milling machine, drilling machines and taps
7	Main inlet pipe. Drawing no: 13	Mild steel rod	Cut to size, drill, turn, drill and then tap a hole	Power hack-saw, Lathe machine, drilling machine and taps

S/No	Components	Materials	Production methods	Equipment
8	Supply end. Drawing no: 12	Mild steel rod	Cut to size, face, turn and drill accordingly	Power hack-saw, Lathe machine, drilling machine
9	Base Plate. Drawing no:5	Mild steel plate	Cut with gas from the plate, grind the rough edges, turn to size, drill holes and then bore hole for supply end sitting and then finally tap	Acetylene and oxygen gas welding equipment, off-hand grinding machine, Lathe machine, drilling machine and taps
10	Dynamo bracket. Drawing no: 1	Mild steel plate	Cut to size, file, bend and then drill	Guillotine machine, hand files, bending machine and drilling machine
11	Supply end brackets. Drawing no: 14	Mild steel plate	Cut to size, file, bend and then drill	Guillotine machine, hand files, bending machine and drilling machine.

4.4 Cost of Production of the Water Turbine

The cost of the materials used, direct labour, over heads, administrative and other expenses in the production of the water turbine as obtainable currently in Zaria and Kaduna markets are as given below.

S/No	Item	Quantity	Unit Price (₦)	Total Price (₦)
1	Driving pinion Φ50mm x 30mm Aluminium rod	1	200.00	200.00
2	Supply end Φ 80mm x 300mm mild steel rod	1	2500.00	2500.00
3	Nozzles 35mm x 35mm Aluminium casting	4	125.00	500.00
4	Base plate 350mm x 20mm mild steel plate	1	3000.00	3000.00
5	Dynamo carrier Φ 200mm x 20mm mild steel plate	1	3000.00	3000.00

S/No	Item	Quantity	Unit Price (₦)	Total Price (₦)
6	Tripod stand 412mm x 33x3mm mild steel plate	3	1000.00	1000.00
7	Rotor arm Φ25.4mm x 110mm Stainless steel rods	4	250.00	1,000.00
8	Main inlet pipe Φ35mm x 60mm mild steel	1	200.00	200.00
9	Dynamo bracket 190mm x 45mm x 6.6mm mild steel plate	1	400.00	400.00
10	Roller bearing	1	2,800.00	2,800.00
11	Rotor	1	500	500
12	6W Bicycle dynamo	1	500	500
13	Bicycle head lamp (with 12 V bulb)	1	400	400
14	Oxygen gas	$\frac{1}{2} bottle$	1,700.00	1,700.00
15	Hose	1	450	450
16	Clip	1	20	20
17	Soluble oil	1 gallon	1000	1000
18	Aluminium Paint, thinner and Primer		530	530
19	Grinding disc	1	500	500
20	Power Hacksaw blade	1	500	500
21	Labour cost		9400.00	9400
22	Overheads, Administrative and other expenses		5,080.00	5,080.00
23	Total		34,255.00	35,380.00

4.5 Bearing Selection

The choice between plain and rolling bearings is usually made on the basis of practical, common sense considerations rather than on the basis of a technically involved appraisal. For guidance, some of the particular advantages of each type are given below.

4.5.1 Rolling bearings

These are usually easier for a draughtsman to incorporate into a design, because all he needs do is to select them from a catalogue which gives full details of load capacities, appropriate speeds, housing fits and tolerances, anticipated life, etc. when spares are available, rolling bearings are easier to replace because there are no wearing surfaces to be reconditioned. Although there is usually no detectable difference between the running frictional losses of the two types, rolling bearings do tend to have much less 'stiction' than most types of plain bearing when starting from rest. For many applications in which the load and speed conditions are not too severe, rolling bearings may be adapted to run in a grease-packed sealed construction that will require no further lubrication or maintenance throughout their life.

4.5.2 Plain bearings

Plain bearings, on the other hand, do not always have to be made by specialist, and in their traditional forms can often be renewed or repaired with no more than the usual workshop facilities available almost anywhere in the world, including ships' engine-rooms. Even in their more advanced forms, plain bearings are often easier to replace than rolling bearings, particularly since most types can readily be made of split construction, enabling them to be removed without withdrawing the shaft. They are also less susceptible to damage from water and dirt, can be built to carry many times higher loads than rolling bearings, and are more suitable for operation at very high speeds. Plain bearings tend to wear out more suitable for operation at very high speeds. Plain bearings tend to wear out more gradually and are much less prone to sudden collapse without warning. Even if they fail, it is usually possible to keep the machine running long enough to avoid any hazards of an immediate shutdown. Plain bearings are quieter and when necessary, can be designed to provide and maintain a more precise concentricity of the shaft within its housing. They do not 'brinell' when subjected to vibration from an external source, and operate reliably under conditions of high cyclic loading such as occur in reciprocating machinery. In special forms arranged to use readily available pressurized gas (e.g. air) as a lubricant, they have a lower coefficient of friction (both when at rest and in motion) than any other known form of support. They also tend to be much cheaper.

4.5.3 Rotor bearings

Sealed rolling bearing (RLS 922 SKF) was selected for the rotor to reduce leakage to the barest minimum.

4.6 Description of a fully built outward radial flow reaction water turbine

Plate I: Components of the Turbine before Assembly

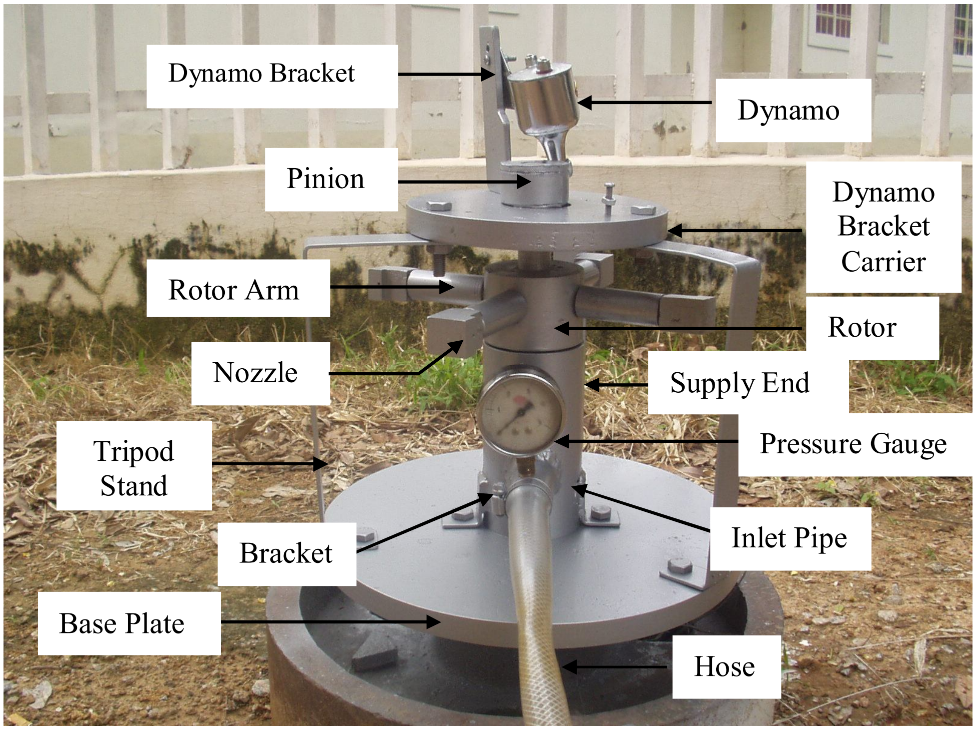


Plate II: Fully built reaction water turbine

The main inlet pipe (with a provision for bolting a pressure gauge to it) and two brackets were brazed onto the supply end. Two bolts were used to fasten supply end to the base plate.

A sealed bearing was force fitted into position created for it on the rotor. Then the four arms were screwed onto the rotor. Four nozzles were also screwed onto the rotor arms. This rotor sub-assembly was then force fitted onto the supply end.

Three tripod stands were bolted to the base plate and the dynamo bracket carrier was then bolted to the tripod stands. The dynamo bracket was bolted on the dynamo carrier and then the dynamo was bolted to the bracket.

A piece of tube was wound round the driving pinion and then the pinion was then connected to the shaft of the rotor by means of a key. The turbine was then sprayed with aluminium paint.

CHAPTER 5 TESTS AND RESULTS

5.1 The Outward Radial Flow Reaction Water Turbine (Sprinkler)

The plate below shows the turbine under test

Plate III: Turbine under Operating Condition

Load and No load tests were carried out on the turbine. Below is the plate showing the set up fully labeled for tests and taking of measurements.

5.2 Procedure for taking Measurement

5.2.1 Measurement of pressure and speed taken with turbine not loaded

The dynamo was not connected to the pinion of the turbine. Water under pressure was introduced at the centre of turbine and jets of water that can cover the area necessary; issue from the ends of the arms at zero gauge pressure. The force on the rotor acted in a reaction to the creation of the momentum. The maximum gauge pressure was then determined to be $1.8 \times 10^5 \, N/m^2$ (gauge pressure) using the pressure gauge attached to the inlet pipe. Then the pressure of the water was varied from $0.5 \times 10^5 \, N/m^2$ to $1.8 \times 10^5 \, N/m^2$ using the valve and the corresponding speeds to the pressures were taken using a tachometer. Below is plate showing readings being taken and the readings taken and a corresponding graph.

Plate IV: Measuring Speed and Pressure under no Load Condition

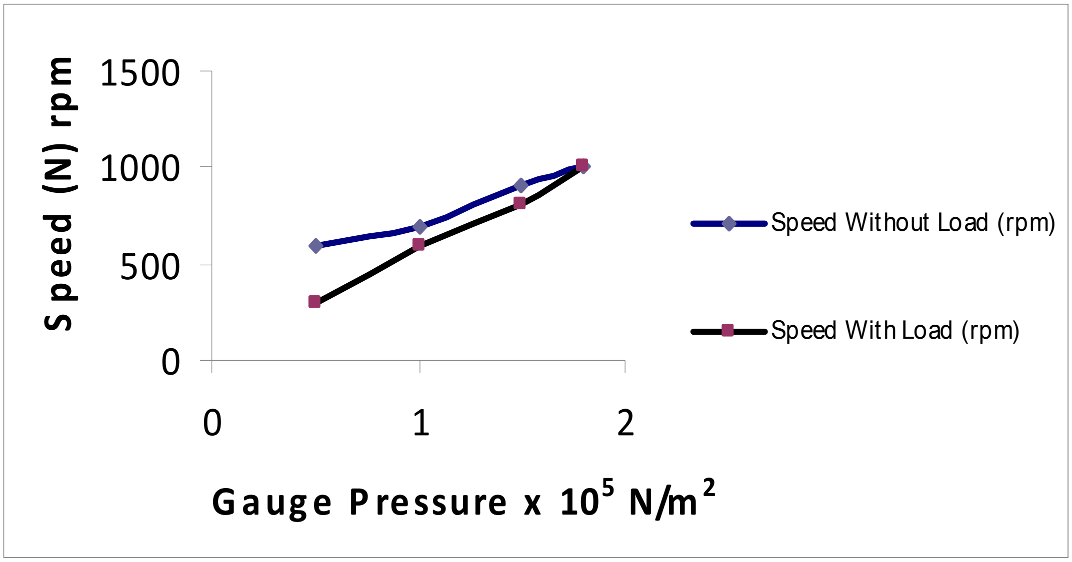


Figure 5.1: Speed versus Pressure for Turbine with and without Dynamo

5.2.2 Measurement of pressure, speed, current, and voltage and flow rate taken with turbine loaded with dynamo

The dynamo was then connected to the pinion of the turbine. A circuit was constructed to take readings of current, voltage and speed corresponding to the pressures considered in section 5.2.1. An Ammeter was connected in series, a voltmeter in parallel, and a bicycle head lamp was also connected in parallel. At each pressure, the hose was disconnected and water was collected for 10 seconds. The volume of the water was measured and the volume and mass flow rates calculated. There are plates below illustrating how the measurements were taken and there are also tables and graphs showing the results obtained from this exercise. The power output, power input, mass flow rate and over all efficiency were calculated and tabulated are shown below. The corresponding graphs are as shown in Figs 5.2-5.7

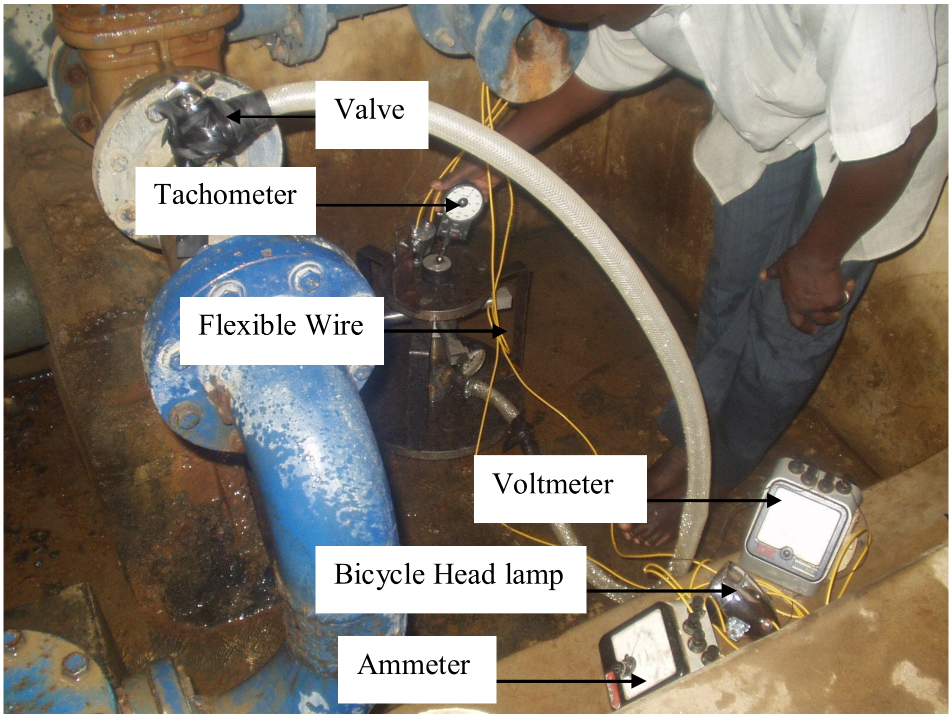


Plate V: Measuring speed, current and voltage at varying pressure under load condition

Plate VI: Measuring flow rate at varying pressure

Plate VII: Measuring flow rate at varying pressure

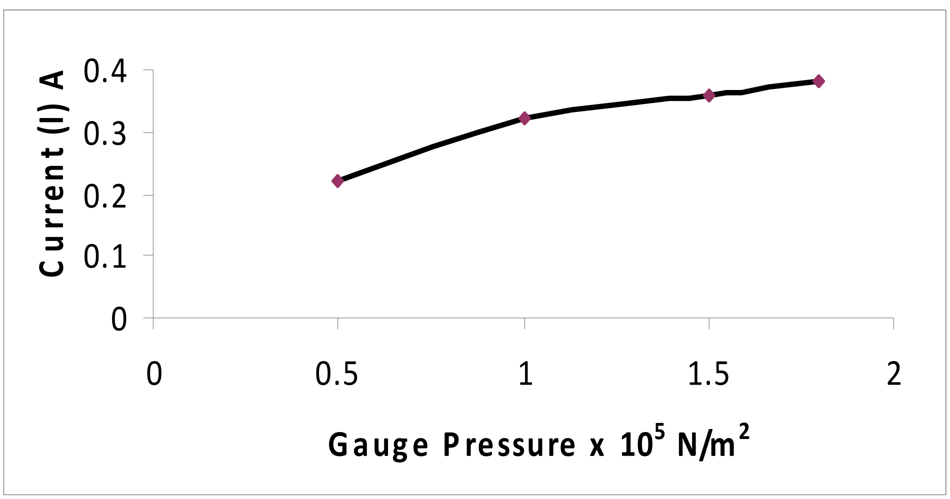


Figure 5.2: Current versus pressure

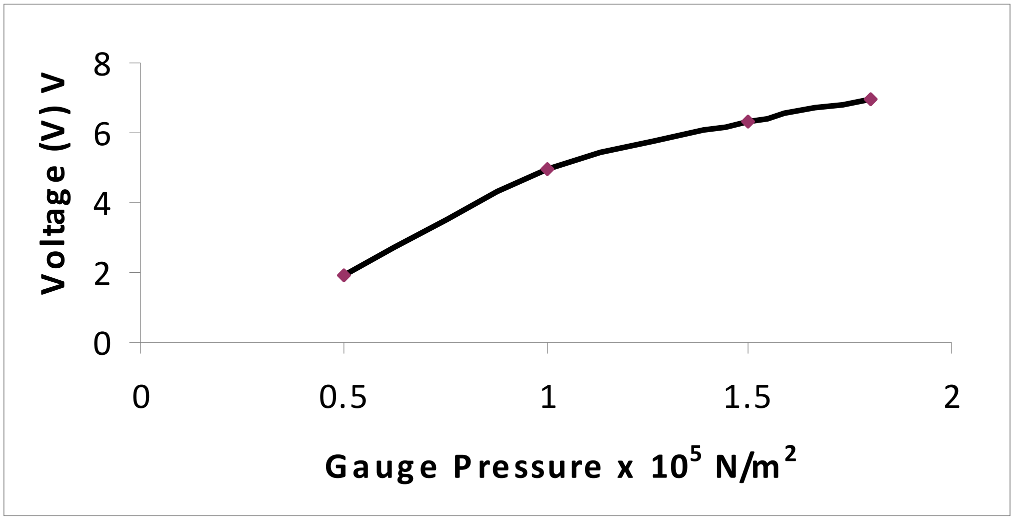


Figure 5.3: Voltage versus Pressure

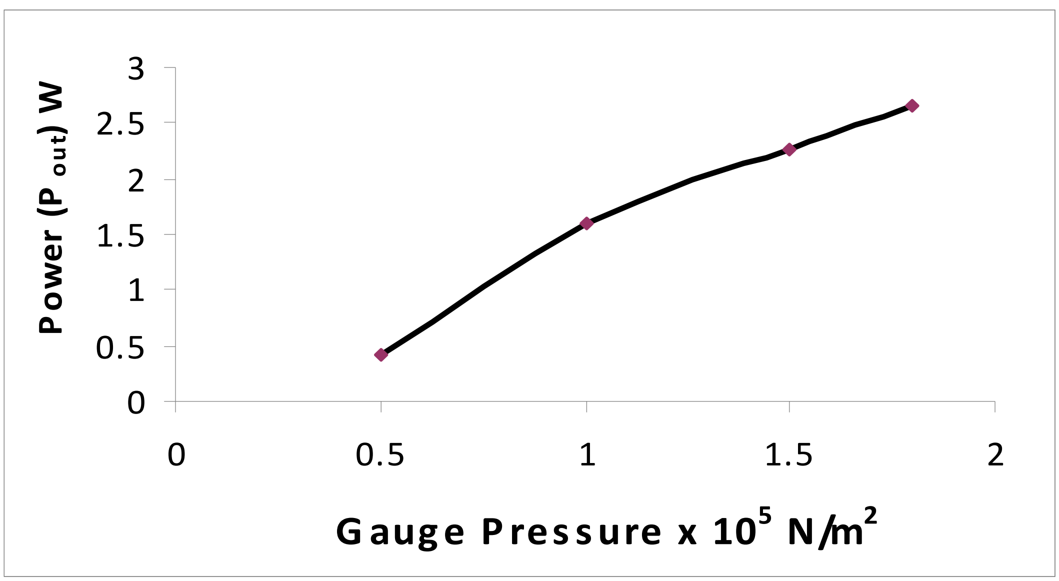


Figure 5.4: Power versus Pressure

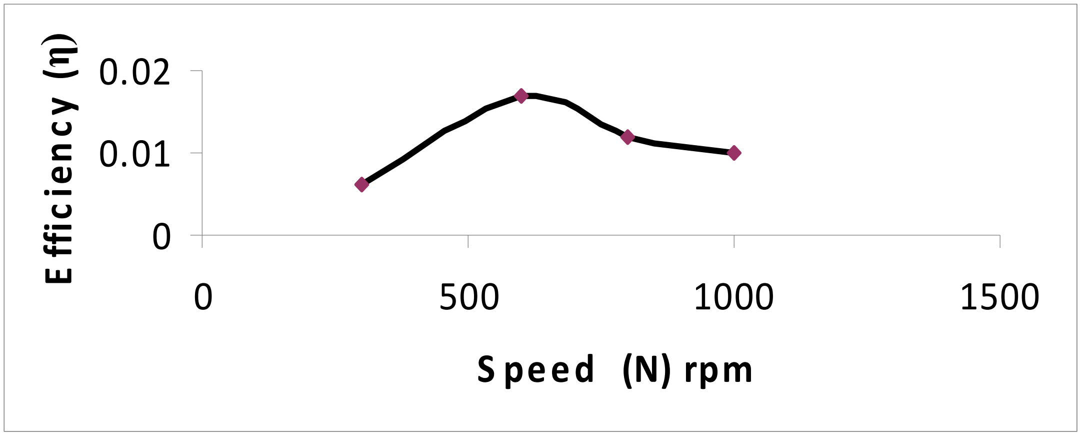


Figure 5.5: Efficiency versus speed

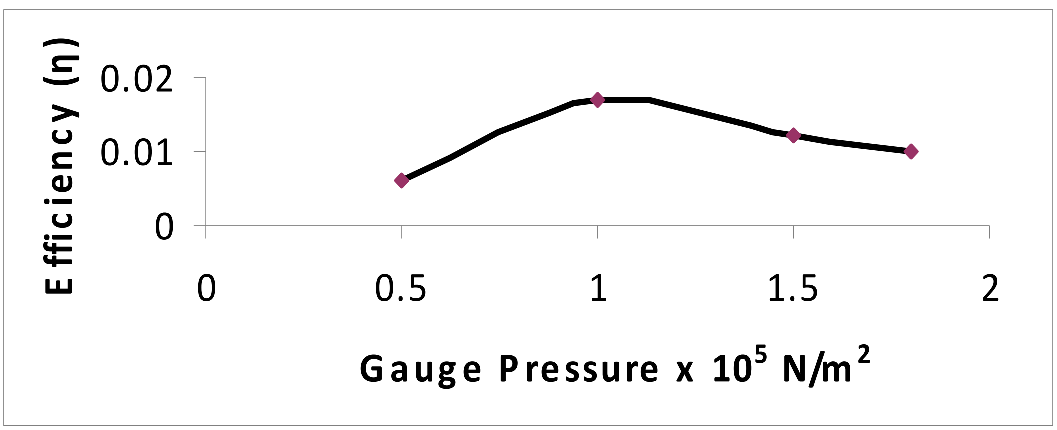


Figure 5.6: Efficiency versus Gauge Pressure

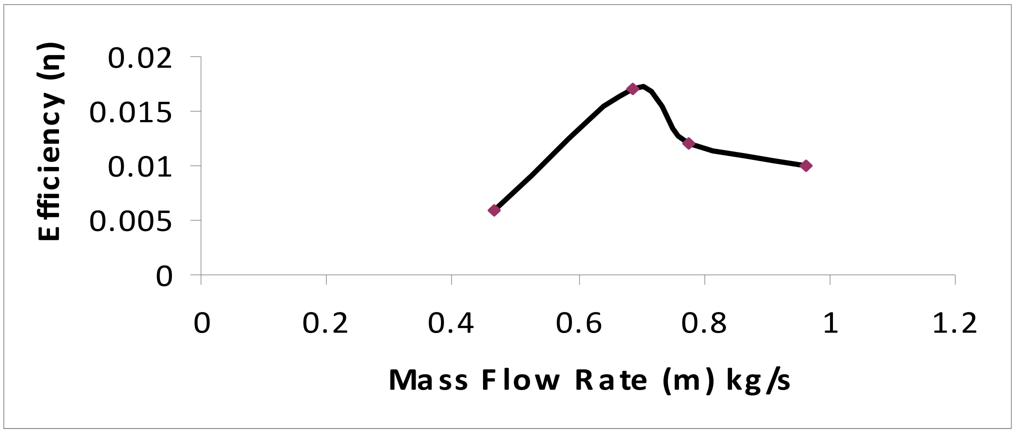


Figure 5.7: Efficiency versus Mass flow rate

5.3 Discussion of Results

An outward radial - flow reaction water turbine was designed, constructed and tested at the Ahmadu Bello University, Zaria Water Works Booster Station. It drove a 6W bicycle electric dynamo. In Fig 5.1, it was observed that when the turbine was loaded with the dynamo under the same gauge pressures as when it was not loaded, there was a reduction in speed at $0.5\times10^5\, N/m^2$, $1.0\times10^5\, N/m^2$, $1.5\times10^5\, N/m^2$ but at $1.8\times10^5\, N/m^2$, the same speed was obtained, this was because the torque was a maximum at the starting point and it continued to reduce to a minimum and tended towards zero at maximum gauge pressure. At this pressure, the torque had very little effect.

It was also observed from Figs. 5.7, 5.6 and 5.5 that the optimum performance of the turbine obtained at a mass flow rate of 0.685kg/s, a gauge pressure of $1.0\times10^5\, N/m^2$, and a speed of 600rpm. Increasing the values of these parameters, the corresponding change in Power delivered was insignificant.

Considering Figs. 5.4, 5.3 and 5.2, the maximum gauge pressure obtained at the Ahmadu Bello University booster station was $1.8 \times 10^5 \, {}^{N}\!/_{m^2}$. Given that the optimum gauge pressure was $1.0 \times 10^5 \, {}^{N}\!/_{m^2}$, increasing the gauge pressure beyond $1.0 \times 10^5 \, {}^{N}\!/_{m^2}$ and even if gauge pressure higher than the maximum pressure here was attained anywhere, the corresponding change in power output, current and voltage would continue to decrease until it would be insignificant or even zero because more and more losses would continue to occur within the rotor without corresponding output.

CHAPTER 6 SUMMARY, CONCLUSION AND RECOMMENDATION

6.1 SUMMARY

Considering where reaction water turbines (i.e. sprinklers) are used, lighting the places will greatly enhance the working time and ease working at the places at night. The residual power head was used to rotate a dynamo and generate electricity thereby eliminating the use of power from other sources that are costlier and not as environmental friendly.

Literature showed that Pelton impulse turbine, Francis Turbine and Kaplan turbine had been satisfactorily designed based on Euler theory and accepted universally.

This research focused more on power generation than watering, four rotor arms were therefore used for the design of this water turbine based on the Euler theory. The design angular speed of the turbine, the rotor arm diameter, the work done on each arm by the fluid, the tangential force and tolerances were determined.

Mild steel, cast aluminium alloy and stainless steel were sourced and bought around Zaria and Kaduna markets. Then the machine tools of workshops in some government organizations in Zaria and a local foundry also in Zaria were used to fabricate this turbine.

Finally, the turbine was tested at the Ahmadu Bello University, water works booster station. It drove a 6W bicycle electric dynamo. The maximum power obtained at maximum mains pressure of $2.8\times10^{5} N/m^{2}$ was 2.66W, this was about half the power rating of the electric dynamo. This was so partly because the designed pressure was not reached and the leakage could not be blocked completely.

The optimum performance occurred at $2\times10^{5} N/m^{2}$, a speed of 600 rpm and a mass flow rate of 0.685kg/s.

6.2 CONCLUSION

The objective of this research to design, construct and test an outward radial flow reaction water turbine was achieved.

In producing this turbine, materials were sourced and bought locally. Fabrication and assembly of various parts of the turbine were carried out at machine tools shop and foundries within Zaria. The cost of materials used in fabricating the turbine stood at thirty five thousand, three hundred and eighty naira only (N 35,380.00). This cost would be greatly reduced when this turbine is mass produced.

The turbine was tested at the Ahmadu Bello University, water works booster station. It drove a 6W bicycle electric dynamo. The optimum performance occurred at a mains pressure of $2\times10^{5} N/m^{2}$ (i.e. gauge pressure plus atmospheric pressure of $1\times10^{5} N/m^{2}$ a speed of 600 rpm and a mass flow rate of 0.685kg/s. The maximum power output from Fig. 5.4 at maximum mains pressure of $2.8\times10^{5}\,{}^{N}\!/_{m^{2}}$ was 2.66W, this was about half the power rating of the electric dynamo because the designed pressure was not reached, the leakage could not be blocked completely and there were losses within the dynamo.

This research revealed the possibility of generating power to light the places that sprinklers would be used from the sprinkler head.

6.3 RECOMMENDATION

In view of the fact that there is continuous need to look for other sources of energy and perseverance is the key to getting better results knowing that quality is a journey not a destination, the following are recommended:

(i) A speed multiplying mechanism could be introduced between the pinions of the rotor and the dynamo to get higher speeds with the same pressure to increase the power output.

(ii) The rotor arms could be made from aluminium alloy casting, the base plate and dynamo carrier support made from 10mm mild steel plate to reduce the weight and cost of materials in order to produce a cheaper water turbine.

(iii) The leakage within the rotor bearing and the supply end assembly should be reduced as much as possible to minimize losses in power output.

(iv) The design could be simulated using available computational fluid dynamics software to get optimum performance at lower pressures and to also to validate the experimental result obtained.

(v) Digital measuring equipment with high sensitivity could be used to reduce error in results to the minimum.

(vi) The number of arms could be increased and the diameter of the jets widened to deliver more power.

(vii) A bigger generator could be used to get higher power output.

REFERENCES

Calvert, J.B. (2003), Turbines: http://www.du.edu/Njcalvert/tech/fluid/turbines.htm. Date visited: 17th January, 2007.

Daugherty R.L and Franzini J.B (1965) *Fluid Mechanics with Engineering Applications.* Sixth edition, McGraw-Hill Book Company Inc. U.S.A pp. 141 – 173, pp. 486 – 514.

Donald F.Y, Bruce R.M, Theodore H.O. (1997). *A brief Introduction to Fluid Mechanics.* John Wiley and Sons Inc. pp. 139 – 201.

Douglas J.F, Gasiorek J.M, Swaffield J.M (1985). *Fluid Mechanics* Second Edition Longman Group Ltd., England. pp. 555-56.

Grigorova D.I, (1988), *Machine Design*, Mech. Eng. Dept., ABU Lecture Series, Unpublished.

Janna W.S (1993). *Introduction to Fluid Mechanics* Third Edition. PWS Publishing Co., Boston U.S.A.

Kearton, W.J. (1951) *Steam Turbine, Theory and Practice.* Sixth Edition Pitman Press, BATH, UK.

Lyman, F.A. (2004). *A Practical Hero Feature Article*.

Nelkon M. and Parker P. (1982) *Advanced Level Physics,* Fifth Edition. Heinemann Educational Books Ltd. London pp. 726-747

Parker M.A and Pickup F. (1976). *Engineering Drawing with worked examples 1*, Third Edition, Rombic Concept Ltd.

Parker M.A and Pickup F. (1981). *Engineering Drawing with worked example 2*, Third Edition, Rombic Concepts Ltd.

Rao S., Parulekar B.B. *Energy Technology*, (1999) Khanna Publishers, Delhi pp. 965 – 993, pp. 994 – 1020.

Rogers G.F.C and Mayhew Y.R. (1980). *Engineering Thermodynamics*. Longman Group Ltd.

Shepherd, D.G. (1956), *Principles of Turbo-machinery*. The Macmillan Company, U.S.A, pp. 49 – 97.

Theraja B.L. (1978), *Electrical Technology.*
Seventeenth Revised Edition. S. Chand and Company ltd. Ram Nagar,
New Delhi 110055. pp. 529-539

Umoren, I.H (2004), *Design of an 8W Outward radial – flow reaction water turbine.* Unpublished B.Eng. Thesis, Ahmadu Bello University, Zaria, Nigeria

Welsh R.J (1983). *Plain Bearing Deign Handbook*. The Thetford Press Ltd., Thetford, Norfolk. (123 – 124).

Yusuf, W.O. (1991). *Design, Construction and Testing of a Horinzontal Rotating Water Sprinkler.* Unpublished B.Eng. Thesis, Ahmadu Bello University (10 – 11).

APPENDIX

Table 5.1: Gauge Pressure versus Speed with and without Dynamo Loaded

$Gauge\Pr essure \times 10^5 \, N/m^2$	Speed Without Load (rpm)	Speed With Load (rpm)
0.5	600	300
1	700	600
1.5	900	800
1.8	1000	1000

Table 5.2: Gauge Pressure versus Current

$Gauge\Pr essure \times 10^5 \, N/m^2$	Current (A)
0.5	0.22
1	0.32
1.5	0.36
1.8	0.38

Table 5.3: Gauge Pressure versus Voltage

$Gauge\Pr essure \times 10^5 \, N/m^2$	Voltage (V)
0.5	1.9
1	5.0
1.5	6.3
1.8	7.0

Table 5.4: Gauge pressure versus power

$Gauge\Pr essure \times 10^5 \, N/m^2$	Power (W)
0.5	0.42
1	1.60
1.5	2.27
1.8	2.66

Table 5.5: Efficiency versus Speed

Efficiency, (η)	Speed (rpm)
0.006	300
0.017	600
0.012	800
0.010	1000

Table 5.6: Efficiency versus Mains Pressure

Efficiency, (η)	$Gauge\Pr essure \times 10^5\ N/m^2$
0.006	0.5
0.017	1.0
0.012	1.5
0.010	1.8

Table 5.7: Efficiency versus mass flow rate

Efficiency, (η)	Mass Flow Rate (kg/s)
0.006	0.466
0.017	0.685
0.012	0.778
0.010	0.963