INTEGRAL PROPELLER TURBINE - INDUCTION GENERATOR UNITS FOR VILLAGE HYDROELECTRIC SCHEMES

by

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I dedicate this thesis to my parents,

Μανωλης (*Manolis*) and Χαριθεα (*Charithea*)

who have instilled in me many of the qualities needed

for such an undertaking.

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Abstract

Integral propeller turbine-induction generator units for village hydroelectric schemes

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Stand alone micro - hydroelectric schemes, especially where grid extension is not feasible, have long been acknowledged as an appropriate way of village electrification in developing countries.

This thesis describes an investigation into *propeller turbines* that are suitable for low head micro hydroelectric schemes due to their high specific speeds and their compact design which allows them to be easily transported. In addition, their direct coupling to generator sets offers several advantages over belt drive systems and gearboxes such as improved system efficiency, increased turbine and generator bearing lifetime, reduced costs, lower maintenance requirements and simplicity of installation.

A review of existing low head turbine technology shows that no design is appropriate for use in developing countries, either because of high costs or the technology is unsuitable for local manufacture and/or maintenance.

The thesis presents the design, development and testing of an integral propeller turbine-induction generator unit, which utilises as much of the local skills and materials as possible. Pipe sections can be incorporated into the design as a result of tailoring turbine dimensions to standard sizes which therefore minimise fabrication costs. The runner blades are made of constant thickness metal sheets, bent and twisted. Test results show that the performance of the turbine is sensitive to the geometrical accuracy of the blades which were difficult to form to the design angles. Moreover, a systematic loss analysis using the experimental data pinpointed inaccuracies in the original design assumptions. Despite this, a turbine efficiency of 53% was achieved, which compares favourably with existing designs.

Based on the knowledge and experiences gained from this research project, the thesis includes proposals for use of criteria and procedures which are likely to lead to a more successful design of propeller turbines for village hydroelectric schemes.

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Publications resulting from the research presented in the thesis

Demetriades, G. M.; Williams, A. A. and Smith, N. P. A. 1995a. The use of propeller turbines in low head stand alone micro hydroelectric power generation units. *International Journal of Ambient Energy*. **16**(3), pp. 165-168.

Demetriades, G. M. and Williams, A. A. 1995b. The design of a low - cost propeller turbines for stand alone micro - hydroelectric power generation units. *In : Proceedings of the 10th Conference on Fluid Machinery*. Akatémiai Kiadó, Budapest, Hungary. pp. 116-124.

Demetriades, G. M., Williams, A. A. and Smith, N. P. A. 1996. A simplified propeller turbine runner design for stand alone micro - hydro power generation units. *International Journal of Ambient Energy*. **17**(3) pp. 151-162.

Demetriades, G. M. and Williams, A. A. 1997. A simplified propeller turbine design for direct coupling to an induction generator for village hydroelectric schemes. *In : Proceedings of the Up and Coming in Fluid Machinery Seminar*, Organised by the Power Industries Division's Fluid Machinery Committee of the Institution of Mechanical Engineers. London, November 1997.

The above papers are included in Appendix I.

Nomenclature

Symbol		Units
	Lower case letters	
b_g	Guide vane height	m
c	Chord length of blade	m
d	Diameter	m
f	Frequency	Hz
	Maximum camber	m
	Systematic uncertainty	%
	Friction coefficient	
g	Gravitational constant	m/s^2
h	Height	m
k	Ratio of angular velocity of the fluid to that of a rotating disc	
	Roughness	mm
k	Velocity coefficient	
m	Mass flow rate	kg/s
	Hydraulic mean depth	m
n_s	Dimensional specific speed, $\frac{N\sqrt{Q}}{H_N^{3/4}}$	min^{-1}
p	Number of poles of a generator/motor	
q	Flow rate through a section	m^3/s
r	Radius	m
s	Slip of an induction motor/generator	
	Gap width	m
t	Blade spacing	m

Symbol		Units
t	Blade thickness	m
u	Peripheral velocity	m/s

Upper case letters

A	Surface area	m^2
C_D	Drag coefficient	
C_L	Lift coefficient	
C_P	Pressure coefficient	
D	Diameter	m
	Drag force	N
F	Resultant force	N
H	Head	m
	Form factor	
L	Lift force	N
	Length	m
M	Moment	Nm
N	Speed	rpm
N_{sp}	Non dimensional specific speed, $\dfrac{\omega\sqrt{P_{out}}}{\sqrt{\rho}\,[g\,H_N]^{5/4}}$	
N_s	Dimensional specific speed, $\dfrac{N\sqrt{P_{out}}}{[H_N]^{5/4}}$,	$\dfrac{rpm\sqrt{kW}}{m^{5/4}}$
P	Power	W
	Pressure	Pa
	Load	N
Q	Flow rate through turbine	m^3/s
R	Radius	m

Symbol		Units
Ra	Relative average	
Re	Reynolds number	
T	Torque	Nm
V	Absolute velocity of blade	m/s
	Average velocity	m/s
	Voltage	V
W	Relative velocity	m/s
X	Radial load factor	
	X axis dimension	
Y	Axial load factor	
	Y axis dimension	
Z	Height	m
	Number of runner blades or guide vanes	

Greek letters

α	Fluid angle in the absolute velocity direction	°
α'	Blade angle in the absolute velocity direction	°
β	Fluid angle in the relative velocity direction	°
β'	Blade angle in the relative velocity direction	°
β^*	Corrected blade angle in the relative velocity direction	°
γ	Blade setting or stagger angle	°
δ	Deviation angle	°
δ^*	Boundary layer displacement thickness	m
Δ	Denotes change in a quantity	
ε	Gliding angle	°

Symbol		Units
ε	Absolute uncertainty	
ζ	Loss coefficient	
η	Efficiency	
ϑ	Draft tube angle	$^\circ$
	Boundary layer momentum thickness	m
θ	Blade camber angle	$^\circ$
ι	Incidence angle	$^\circ$
λ	Angle measured in the spiral casing	$^\circ$
κ	Weinig coefficient for flat plates in a cascade	
μ	Weinig factor for circular arc blades	
	Bearing coefficient of friction	
	Dynamic viscosity	kg m^{-1} s^{-1}
ν	Kinematic viscosity	m^2/s^2
ξ	Head loss	m
π	Constant	
ρ	Density of water	kg/m^3
σ	Solidity ratio	c/t
σ_{Th}	Thoma cavitation constant	
σ_P	Permissible direct stress	Pa
σ_{max}	Maximum stress	Pa
τ	Shearing stress	Pa
υ	Percentage leakage loss	%
Φ	Flow coefficient, $\frac{Q}{\omega D^3}$	

Symbol		Units
Ψ	Head coefficient, $\dfrac{g\,H_N}{\omega^2\,D^2}$	
ω	Rotational speed	s^{-1}

Subscripts

a	Axial direction
	Atmospheric
av	Average
A	Stream level
b	Barometric
B	Turbine level
bep	Best efficiency point
c	Cascade
	Casing end wall
ce	Casing entrance
C	Tailrace level
dt	Draft tube
exp	Expansion
ext	Exit
E	Euler
fr	Friction
g	Denotes the gap between a rotating and stationary disc
	Guide vane
	Generator
gi	Guide vane section inner diameter
go	Guide vane section outer diameter

Symbol		Units
G	Gross	
H	Hub	
	Hydraulic	
in	Input	
	Into the turbine	
L	Leakage	
m	Meridional or axial direction of flow	
	Mean	
	Motor	
max	Maximum	
mid	Middle section	
M	Mechanical	
N	Net	
out	Output	
p	Penstock	
	Pipe	
r	Radial direction	
R	Runner	
s	Setting	
	Specific	
	Shaft	
	Specific speed	
sp	Spiral casing	
synch	Synchronous	
T	Tip section	

Symbol		Units
T	Turbine	
u	Tangential (or circumferential) direction of flow	
v	vapour	
V	Volumetric	
1	Inlet section of the runner	
2	Outlet section of the runner	
	Entry to the draft tube	
3	Exit from the draft tube	
o	Denotes the overall efficiency of the turbine	
∞	Average direction of flow	
x	X axis direction	
	Denotes quantity	
y	Y axis direction	

Chapter 1

Introduction

"There is no substitute for energy; the whole edifice of modern life is built upon it. Although energy can be bought and sold like any other commodity, it is not just another commodity, but the precondition of all commodities, a basic factor equally with air, water and earth"

E. F. Schumacher

1.1 Micro hydro and rural electrification

1.1.1 What is micro hydro ?

Lack of electricity is a serious constraint to development in most of the rural communities in developing countries [Brown, 1992 and Khatib, 1993]. The extension of the national grid to remote communities in developing countries is prohibitively expensive. According to the Intermediate Technology Development Group[1] (ITDG) office in Colombo,

> *"Approximately 70% of Sri Lanka's population live in rural areas. Only 30% of the population, mostly in the urban areas have access to grid electricity"* [ITDG, 1993].

The potential sites for *small scale*, *stand - alone*, i.e. not connected to the grid, hydroelectric schemes, coincide quite closely with the locality of scattered rural villages and permits power to be generated near the end users at low transmission and distribution costs [Inversin, 1986; Meier, 1985 and Scott, 1991]. Such schemes with power output less than 100kW are usually referred to as ***micro hydro schemes***. More recently, a new classification of *pico hydro* has been defined for schemes in the 0.5 to 5kW power range [Harvey, 1993 & 1995].

Pico hydro schemes, which are also referred to as *domestic hydroelectric schemes*, supply electrical power only to a single household or a small group of houses. *Pico hydro schemes* have been very successful due to the development and use of low cost[2], locally made Pelton turbines directly coupled to induction generators known as a Peltric sets [Smith 1992a; Waltham 1993 and Williams, 1989]. According to Harvey [1996], sales of Peltric sets in Nepal, which are available off-the-shelf, are in the order of 50 to 100 per year and increasing year by year.

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[1] Intermediate Technology Development Group is a UK based charity that was founded in the 1960's by the economist E. F. Schumacher [1973]. It's purpose [Miles, 1982] is to facilitate the introduction of appropriate technologies to developing countries. They have offices in Bangladesh, Kenya, Peru, Sri Lanka, Sudan Nepal, UK and Zimbabwe.

[2] Details of the cost breakdown of a Peltric set made and sold in Nepal are available in Appendix A.

The power produced can be used to drive *electrical machines*, and/or *mechanical machines*. Micro hydro schemes can therefore be classified according to the end use of the generated power into the following categories :

- *Electrical schemes*, where *electrical machines*, such as vehicle alternators, synchronous and induction generators, supply electrical power for a range of uses.

- *Mechanical schemes*, where *mechanical machines* are used in small workshops and for agro - processing.

- A combination of the two systems described above where *mechanical power* is used to drive *mechanical machines* during the daytime and *electrical machines* provide *electrical energy* during the evening.

Electrical power used during the evening can replace kerosene and liquified petroleum gas for lighting [Meier, 1985 and Munasinghe, 1989]. The benefits from lighting as perceived by the recipient household are improved conditions in the evenings for household work, reading and studying. During the daytime, electrical power can be used for battery charging, refrigeration (which may involve vaccines as well as food storage) and to operate small radio and television sets [Carrasco, 1993; Inversin, 1986 and Woollacott, 1996]. Electricity can also be used for small businesses.

The dependence of the rural communities on fuel wood for cooking is contributing to deforestation and soil erosion [Mackay, 1990]. Moreover, smoke from indoor fires frequently contributes to respiratory problems [Waltham, 1991]. Micro hydro electric power can therefore be used for cooking with the use of the recently developed low wattage electric cookers [Young, 1989] and provide pre - heated water for washing.

Figure 1.1 shows the world wide use of micro hydro power for rural electrification[3].

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[3] Specific aspects of the development of micro hydro, with reference to Nepal, Sri Lanka and Peru, are available in Appendix B.

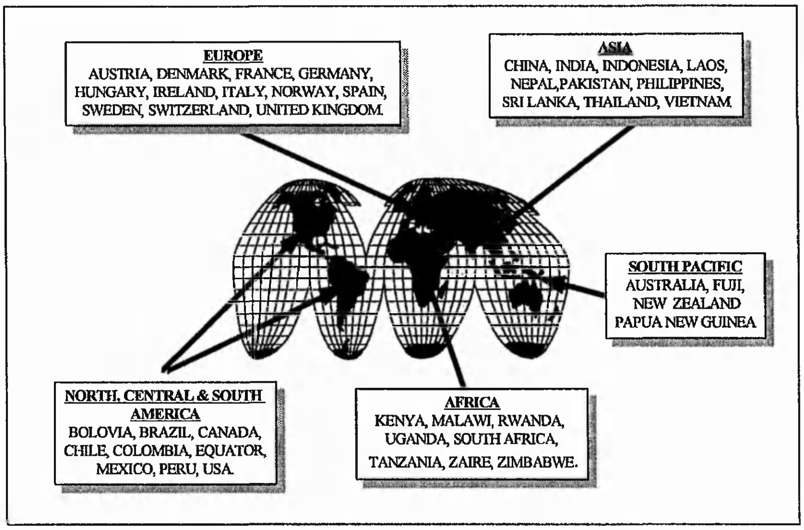

Figure 1.1 : World Wide micro hydro electric schemes.

1.1.2 Comparison of micro hydro with other energy sources

The most common technological options for stand - alone electricity generation in remote areas other than grid connection are :

- ❑ Household photovoltaic (PV) units.
- ❑ Micro hydro power generation schemes.
- ❑ Diesel driven generator sets.

Biogas gasifiers, biomass technology and wind power are also used, though to a smaller extent [Rijal, 1995]. A cost comparison between PV units, micro hydro power generation, diesel generators and grid connection for a Nepalese village consisting of 100 households[4] [Inversin, 1994] is given in *Figure 1.2.* For each option, the per consumer *life cycle cost* (LCC) over a period of 15 years was calculated allocating 50W per household.

Transportation costs have not been considered during the analysis.

[4] Details presented here are based on the information published by Inversin [1994] and are available in Appendix A.

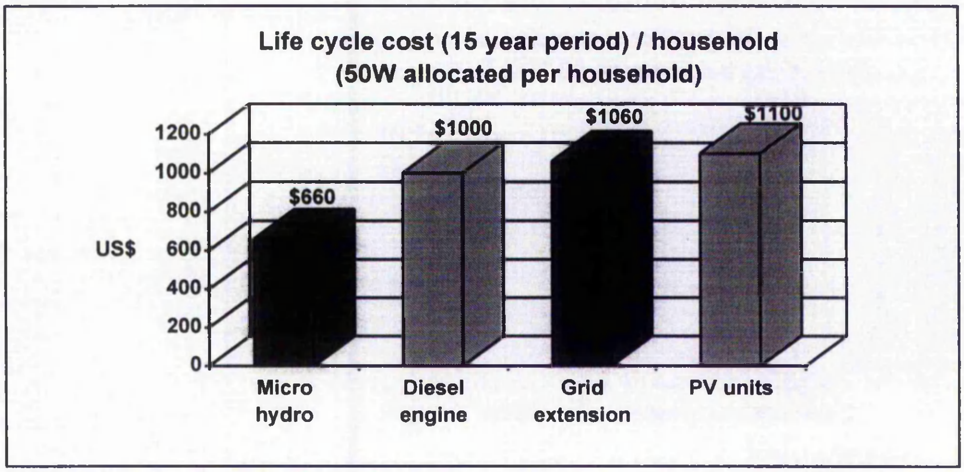

Figure 1.2 : Summary of cost breakdown for various electricity supply options to serve a Nepalese village with 100 consumers, [data adopted from Inversin, 1994].

Expensive transmission and distribution networks are required to bring grid power to the consumers. This situation is costly and is economically feasible only where large load centers exist i.e. urban areas. For this reason, electrical energy in remote and rural areas of developing countries has traditionally been derived from stand - alone, mass produced diesel driven alternators. However, despite their low capital cost, diesel driven generator sets are characterized by high running costs, high maintenance requirements and low efficiencies [Phillips, 1993].

Environmental considerations and the cost of fossil fuels alongside the increasing demand by the rural communities for electricity and the benefits associated with it, have generated a great interest in renewable[5] energy sources and technologies for village electrification. Solar PV units are the costliest of all and can only be used to power energy efficient lighting, small radio and television sets. Their performance relies on the weather conditions and their technology does not allow local manufacturing in developing countries. The use of batteries can also be a problem since their life expectancy depends on how well they are maintained.

Micro hydro schemes are generally the least cost electricity source where suitable sites exist. The capital cost per kW installed varies between and sometimes within countries.

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[5] The term renewable energy can be defined in several ways; for example, Twidell [1986] defines renewable energy as "*the energy obtained from continuous or repetitive currents of energy recurring in a natural environment*".

Schemes that are over designed or use imported equipment can cost in excess of US$10,000/kW. Costs in the range of US$ 1200-2000/kW [Ariyabandu, 1994 and Pandey, 1994] can be achieved by :

- The use of suitable turbine designs that can be manufactured, installed and maintained in developing countries utilizing materials and skills[6] locally available [Cromwell, 1992; Kristoferson, 1991; Meier, 1989 and Waltham, 1993 & 1996].

- The small size and relative simplicity of micro hydro schemes allows the involvement of local communities in all aspects, from initiation and implementation to operation, maintenance and management. When the local communities contribute labour and materials, the costs incurred are lower, and when the villagers are committed to a properly planed and executed project, the possibility of its long term success increases significantly [Inversin, 1986 and Meier, 1989].

- Considerable technological developments in the area of electronics and control systems have made the control of the generator output much easier and less expensive compared to using complex and bulky mechanical governors [Henderson, 1993 and Smith, 1994a, b & c].

1.2 Components of micro hydro schemes

All micro hydro schemes are run-of-river type. A typical layout is shown by *Figure 1.3.* Unlike large hydroelectric schemes, which have generating power from a few to hundreds of MW capacity, the development of micro hydro schemes is not associated with massive capital investment. Moreover, run-of-river schemes do not require the construction of reservoirs, which are environmentally hazardous [Jobin, 1986] and quite often require the resettlement of a large number of people and loss of productive land. Micro hydro electric schemes are often described in terms of head. They can be classified as :

---

[6] A typical workshop is mainly involved with fabrication using steel, and is usually equipped with an arc welding machine, a lathe, a power saw, a pillar drill and a grinder [Oram, 1995].

- ❑ **Low head** **less than 5 m.**
- ❑ **Medium head** **5 to 30 m.**
- ❑ **High head** **above 30 m.**

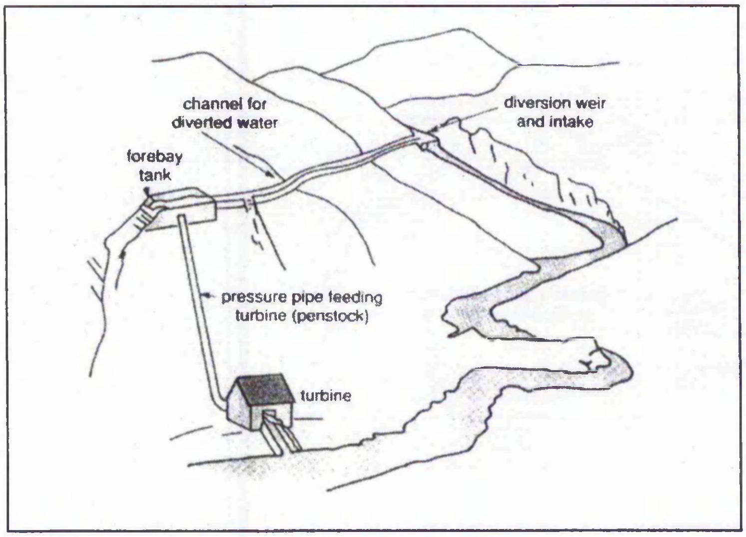

Figure 1.3 : A typical micro hydro site.

The major components of micro hydro schemes fall into two categories : *civil works* which are buildings and structures and *electromechanical components*, such as *water turbines*, *driven machines* and *controllers*. The cost breakdown of a typical medium head scheme is given by *Figure 1.4*.

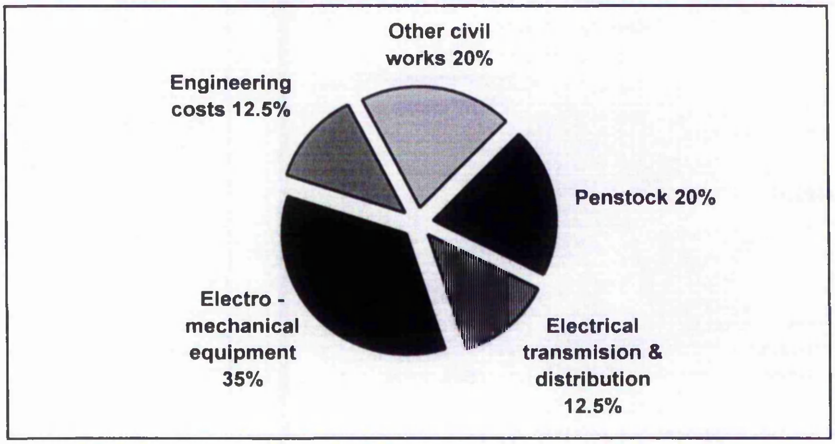

Figure 1.4 : Cost breakdown of a typical medium head micro hydro power generation scheme, [adopted from Cromwell, 1989 and Smith 1994a].

1.2.1 Classification of water turbines

Water turbines are classified according to the following :

- ❑ The direction of the water through the runner.
- ❑ The action of the water on the runner blades.
- ❑ Their specific speed.

1.2.1.1 Direction of water flow through the runner

Water turbines are classified into : (a) *tangential or peripheral flow*, (b) *radial - flow*, *inward* or *outward*, (c) *mixed* or *diagonal flow*, and (d) *axial flow* as shown by *Figure 1.5* a), (b), (c) and (d) respectively.

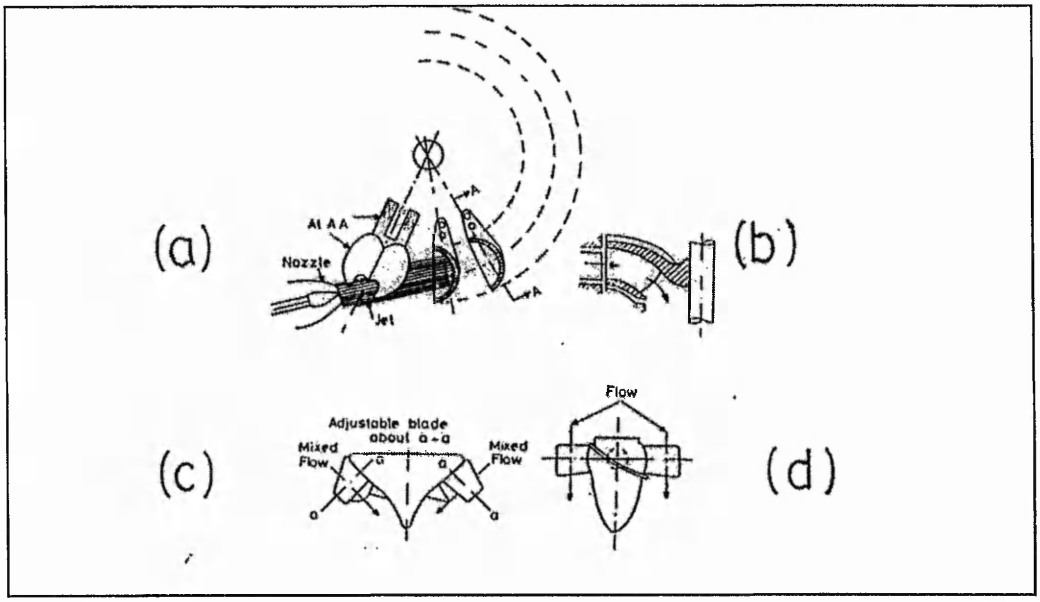

Figure 1.5 : Schematic diagrams of turbine runners.

(a) Peripheral - flow, Pelton Wheel.

(b) Radial - flow, Francis turbine.

(c) Mixed flow, Dériaz turbine.

(d) Axial flow, Kaplan turbine.

1.2.1.2 The action of water on the blades

There are two basic types of turbines, denoted as *impulse* and *reaction*. In an *impulse turbine*, the available head is converted to kinetic energy before entering the runner, the power available being extracted from the flow at atmospheric pressure. In a *reaction turbine*, the runner is completely submerged and both the pressure and the velocity decrease from the inlet to the outlet. Reaction turbines in general rotate at faster speeds than impulse turbines.

1.2.1.3 The specific speed of the machine

Turbines are also classified by their *specific speed*, a non-dimensional term [Douglas, 1995; Sayers, 1982 and Turton, 1984] which does not contain the linear dimension of the turbine. The specific speed of a turbine is given by :

$$N_{SP} = \frac{\omega \sqrt{P_{out}}}{\sqrt{\rho}\,\left[gH_N\right]^{5/4}} \qquad \{1.1\}$$

All geometrically similar turbines have the same specific speed irrespective of their size. Low specific speed machines work under a high head and need only small flow rates per unit power output. On the other hand, high specific speed machines work under relatively low heads and need high flow rates for the same power.

A more detailed explanation of the use and importance of the specific speed term is given in Chapter 3. For the purposes of this thesis, a dimensional specific speed term is used in order to enable the comparison of the proposed design with information available in literature [Raabe, 1985 and Bohl, 1991]. This dimensional term of specific speed is

$$n_s = \frac{N\sqrt{Q}}{\left(H_N\right)^{3/4}} \qquad \{1.2\}$$

Table 1.1 shows the specific speed ranges of various reaction and impulse type turbines.

TURBINE NAME	N_S	TURBINE TYPE
Pelton Wheel		
• single jet	4 - 35	IMPULSE
• twin jet	17 - 50	
• four jet	24 - 70	
Cross flow turbine	20 - 150	IMPULSE
Turgo turbine	20 - 80	IMPULSE
Francis turbine	70-430	REACTION
Propeller turbine (axial)	300 - 675	REACTION

Table 1.1 : Turbine specific speeds, [adopted from Kadambi, 1977 and Harvey, 1993].

1.2.2 Turbine selection

The selection of the most suitable turbine for any particular hydro site depends on the site characteristics, the dominant factors being the *head* and *flow rate* available and *power required* [Holland, 1983]. *Table 1.2* shows the operating head range of various turbines. Selection also depends on the *operating speed* of the turbine. High operating speeds enable the direct coupling of turbines to generators with no need for speed increasing mechanisms such as belt drives or gearboxes [Demetriades, 1995a]. An extensive review of turbine technology for low heads is given in Chapter 2.

Type	**Operating head range (m)**
Propeller turbine	2-10
Cross flow turbine	3-50
Francis turbine	4-100
Centrifugal pumps as turbine	10-100
Turgo turbines	20-100
Pelton wheel	30-200

Table 1.2 : Range of operating heads of various turbine types, [adopted from Hothersall, 1984; Holland, 1986 and Williams, 1995a].

1.2.3 Electrical machines

Three types of electrical machines are widely used in micro hydro schemes. These are : *vehicle alternators*, *synchronous generators* and *induction generators*.

1.2.3.1 Vehicle alternators and battery charging

Vehicle alternators are often used in very small installations (under 1 kW), designated for battery charging. They are widely available and well understood, but not designed for continuous operation. The advantages of vehicle alternators are :

- ☑ Low cost and easily available spare parts.
- ☑ Repair services.
- ☑ A voltage regulator is an integral part of the alternator.

The disadvantages of vehicle alternators are :

- ☒ They are designed to operate at high speeds (2000 rpm and above) and are therefore rarely suitable for direct coupling to turbines.
- ☒ They have poor long term reliability.
- ☒ They are of open construction to aid cooling and any attempts to cover them in order to prevent entry of dust and water may cause overheating and failure.
- ☒ Electricity is generated as alternating current (AC) and then rectified to direct current (DC) at either 12V or 24V, ratings that limit the applications to low voltage appliances, and make transmission over more than few meters uneconomic and impractical.

Batteries offer a means of transporting small quantities of electrical energy over distances that would be uneconomical with cable transmission [Frankel, 1991]. However, in a recent report by Smith [1995] the high capital cost of batteries is emphasized along with the additional charging and transport costs.

Moreover, the life expectancy of batteries is often short due to the fact that they are rarely well maintained by their users.

1.2.3.2 Synchronous generators

This is the conventional approach for any alternating current system which is not connected to the grid. Synchronous generators are so called because the frequency generated is directly related to the speed of the shaft. The synchronous speed is given by

$$N_{synch.} = \frac{120}{p} f \qquad \{1.3\}$$

According to Smith,

> *"Advice must be taken from literature and turbine or alternator manufacturers to make sure that the alternator is suitable for use with the over speed (the runaway speed of the turbine), humidity, altitude and control system of a proposed installation"* [Smith, 1994b].

Synchronous generators are expensive for sites where power generation is less than 30 kW [Smith, 1994b]. In addition, smaller machines tend to have slip-rings and brushes which require routine maintenance. Machines that are specifically designed for hydroelectric generation are very expensive and unsuitable for small power outputs. Another limitation of the synchronous generators is that they are only available as 4 and 2 pole machines running at 1500 and 3000rpm respectively. Direct coupling to turbine shafts is therefore impossible with slow speed turbines.

1.2.3.3 Induction motors used as generators

The cost and reliability of micro hydro schemes can be further improved when induction motors run as generators are used instead of synchronous generators [Smith, 1994a & 1996 and Williams, 1992]. Induction generators are also referred to as asynchronous generators and run at speeds slightly higher than synchronous speed due to slip[7]. Capacitors are needed to convert motors into generators, an added cost of about 20%.

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[7] Appendix C more information on slip and its effect on motor and generator speed.

Induction machines have the following advantages :

- ☑ They are robust and for powers less than 20 kW are less expensive than synchronous generators and therefore more suitable for micro hydro electric schemes [Holmes, 1987].

- ☑ They are widely available as 2, 4 and 6 pole machines and sometimes as 8 pole machines.

Induction generators have two disadvantages as well :

- ☒ Whilst synchronous generators can be purchased ready for use, the induction machine will not work unless capacitors are fitted for self excitation.

- ☒ As induction machines are not always available with suitable voltage ratings for use as generators, modifications to the winding connections may be required, [Bell, 1995].

1.2.4 Controllers

One of the significant advances that has been made in recent years is the adoption of a different approach to turbine-generator speed control. Formerly, complex mechanical water flow governors were used to control the turbine speed and thereby the frequency of the electrical output. However, their mechanical complexity and unreliability contributed greatly to the cost of micro hydro electric schemes. Today, electronic controllers are available and are acknowledged as the most reliable method for controlling micro hydro plants due to a number of advantages :

- ☑ Almost maintenance free and easy to operate.

- ☑ They allow the use of simpler turbine designs with fewer moving parts, giving an overall reduction in the cost of the plant.

- ☑ They can be made or assembled locally.

When synchronous generators are used, the frequency is generally controlled by an electronic load controller (ELC) [Henderson, 1993 & 1996 and Smith, 1996]. Voltage regulation is not required because nearly all synchronous generators contain an in-built voltage regulator [Smith, 1996]. *Figure 1.6* shows a schematic arrangement of an ELC.

With reference to *Figure 1.6*, the generator produces the same amount of electricity all the time. The controller directs electricity which is temporarily in excess of demand to a resistive load, known as the *ballast* load, thereby maintaining a balance between the total electric load and the hydraulic input power from the turbine.

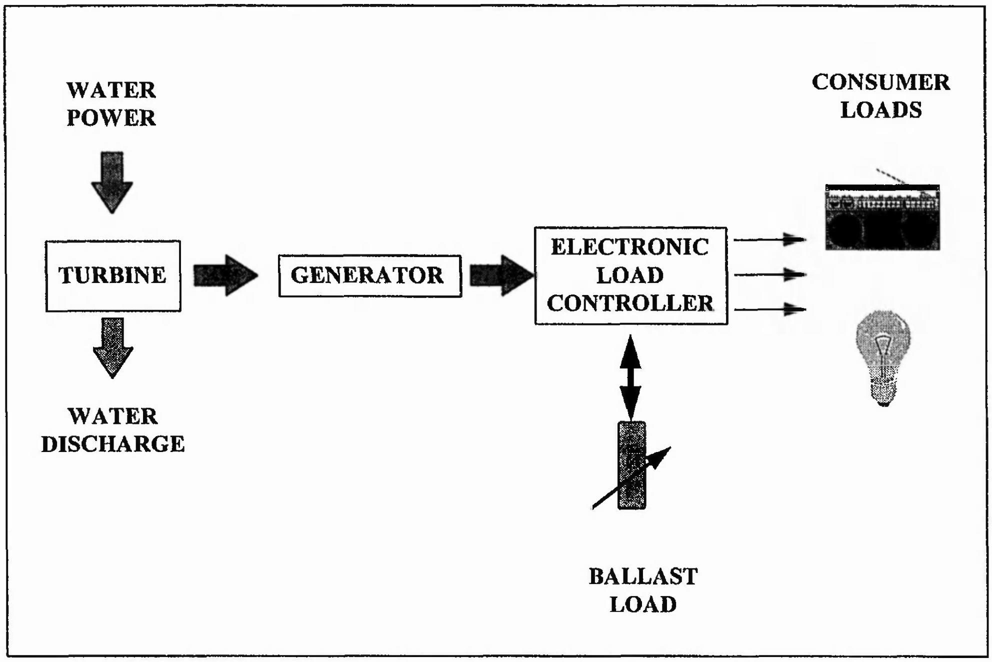

Figure 1.6 : Principles of electronic load control.

Until recently, the main drawback with using induction motors as generators was the need for separate control units for voltage and frequency regulation. However, a recently developed electronic load controller, the Induction Generator Controller (IGC) [Smith, 1992b] allows more reliable and easy control of the induction generator. The major technological advantage of this controller is that one unit is required for the frequency and voltage control. Like all other components of a micro hydro unit, the controller can be made locally.

A cost comparison between induction and synchronous generators with the various control systems available for a 10 kW unit is given by *Table 1.3*.

Approximate cost comparison for a 10 kW unit (US$)		
Synchronous generator with mechanical governor	• Generator	1224
	• Frequency control	2856
	• TOTAL	4080
Synchronous generator with ELC	• Generator	1224
	• Frequency control	408
	• TOTAL	1632
Induction generator with load and voltage control	• Generator	408
	• Frequency control	408
	• Voltage control	816
	• TOTAL	1632
Induction generator with IGC	• Generator	408
	• IGC	408
	• TOTAL	816

Table 1.3 : Cost comparison between synchronous and induction generators and their control methods, [adopted from Holland, 1989].

1.3 Low head micro hydro schemes

This section examines the potential for low head micro hydroelectric schemes, the technology available in developing countries and the reasons that have so far limited the wider implementation of low head schemes.

1.3.1 The potential for low head micro hydro

Most micro hydro schemes are located in the mountainous regions of developing countries, such as those of the Himalayas on the Indian subcontinent and the Andes mountains of Latin America. Pelton wheels are used for high head sites, whereas cross flow turbines are most common for medium head sites, [Smith, 1992a and Waltham 1993].

Although there is a large number of people who live in the hilly and mountainous areas, population density is greater in low lying areas where rivers have large flow rates but with more gentle head drops. In a recent report to the Overseas Development Agency (ODA), it is suggested that it is in these areas that micro hydro has the greatest potential [Waltham, 1996]. Potential low head sites are also found in countries with extensive irrigation canals with small drops, such as Egypt, India, Indonesia and Pakistan. When a low head scheme is part of irrigation works, capital costs are generally lower because much of the civil works are already in place. Moreover, there is better control of the flow rate.

1.3.2 Low head micro hydro in developing countries

People have harnessed the potential of falling water to produce power for centuries. Low head sites have generally been developed for agro-processing using traditional wooden water wheels, known as *pani ghattas*, which have been developed and manufactured locally. *Pani ghattas* have flat blades and run on a vertical axis, driving the millstones located directly above it as shown by *Figure 1.7*, [Hislop, 1987a & b and Brown, 1992]. In Nepal, it is estimated that there are 20,000 pani ghattas still in use.

The advantages of using *pani ghattas* are :

- ☑ Simplicity and availability of designs and ease of manufacturing.
- ☑ Ease of repair.

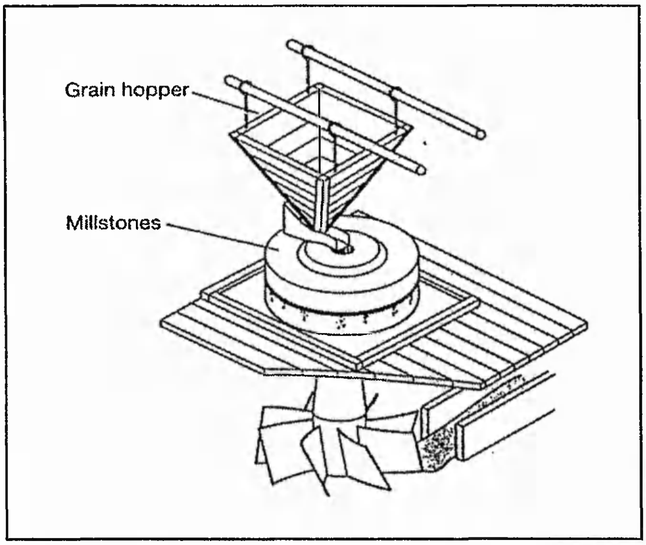

Figure 1.7 : The Nepalese pani ghatta.

Due to the low efficiencies of the pani ghattas, *cross flow* turbines are used for applications that require higher power outputs. Cross flow turbines have become very popular with micro hydro, mostly because of the ease of manufacture and access to designs. No casting is needed for most of the designs, and machines are commonly built in basic workshops. The runner blades can be fabricated from lengths of pipe cut into strips or sheet metal plates. *Figure 1.8* shows a typical cross flow turbine.

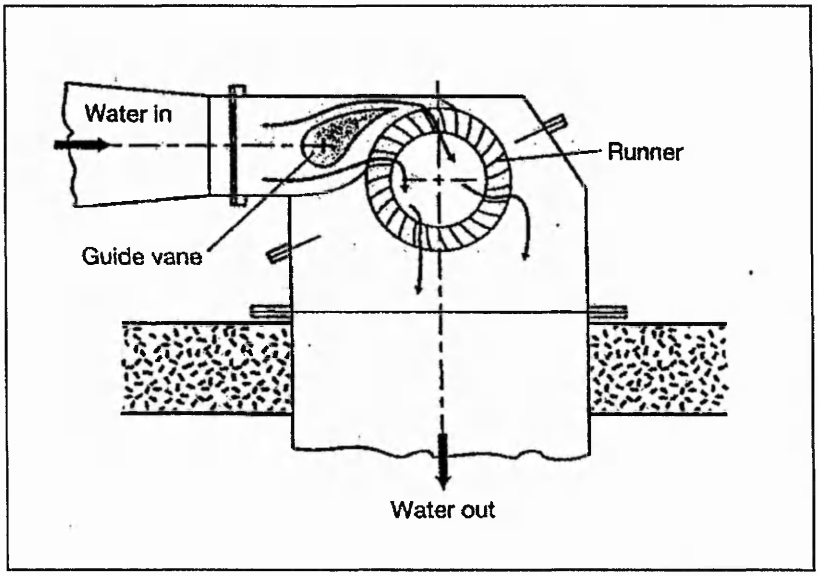

Figure 1.8 : Cross flow turbine.

Cross flow turbines tend to have moderate efficiencies at peak flow, with part - flow efficiencies dependent on the particular design [Harvey, 1993]. However, cross flow turbines are considered inappropriate for low head pico hydro electric schemes [Demetriades, 1995a and Waltham, 1996] for the following reasons :

- ☒ Speed increasing mechanisms are required for driving a generator.
- ☒ The size of cross flow turbines becomes large at such low heads, expensive to produce and difficult to carry over rough terrain.

With a few exceptions, *Francis turbines* have never been a serious option for heads lower than 10 m [Waltham, 1996]. Though *propeller turbines* are ideal for low head micro hydroelectric schemes, their development has so far been at an experimental stage with the only exception being China. Detailed consideration of low head turbine technology is given in Chapter 2.

1.3.3 Barriers to the wider implementation of low head micro hydro electric schemes

With reference to the report on low head micro hydro potential by Waltham [1996], it is believed that the development of low head schemes has been hindered by the following :

- ☒ The lack of suitable turbine technology for low head micro hydro power generation in developing countries.
- ☒ High costs. The cost per installed kW capacity tends to increase as the *head* is reduced. Hence, low head micro hydro power generation is the least economically feasible situation.
- ☒ High silt content in water, a frequent condition in developing countries due to deforestation and frequent storms, requires extensive civil works as silt traps tend to be bigger for large flow schemes. Silt is a common cause of bearing and seal failure.

☒ The risk of flooding is a more serious problem with low head sites. A very large proportion of available head must be wasted to spare the system from flood damage.

1.4 Project origins and objectives

So far, the vast potential of low head micro hydro schemes has remained untapped for the reasons explained in the previous section. The aims outlining the course of this thesis are as follows :

> "*...to develop a low cost propeller turbine, suitable for low head pico hydro power generation for village electrification in developing countries. The design of the propeller turbine will enable local manufacturing in developing countries utilizing the skills and materials available...*"

For the purposes of this thesis, an integral *propeller turbine-induction generator unit* will be referred to as a ***propeltric unit***. The objectives of this research project are :

❑ Investigation and analysis of past and present low head propeller turbine designs for village electrification in developing countries.

❑ Design of a propeltric unit for the power range 1-5kW that would operate at reasonable efficiencies in the range of 55 to 65%.

❑ Manufacture the proposed design within the UK under conditions that are normally found in developing country workshops.

❑ Use the knowledge and experiences gained from field trips and pilot projects within the UK and abroad[8] to develop strategies for the successful technology transfer of propeltric units to developing countries.

❑ Carry out a series of tests at design and off design conditions and optimize the turbine performance.

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[8] Details of the visits can be found in Appendix B.

1.5 Contribution to knowledge

In the author's opinion, the following points presented in this thesis are contributions to knowledge :

- Production of a general propeller turbine design method that will allow field workers in developing countries to design propeller turbines suitable for direct coupling to induction generators, Chapter 3.
- Development of a design that will be easily fabricated utilizing materials and skills locally available and easily transported over rough terrain and installed in remote areas with minimum requirement of tools, Chapter 3.
- Development of a non-contact seal for continuous maintenance free operation in silty environments, Chapter 3.
- Development of a systematic analysis for estimating the head losses and therefore the performance of low head propeller turbines, Chapter 4.
- Development of a method that allows the flow distribution in a spiral casing to be determined as well as the loss in momentum through the guide vanes to be calculated, Chapter 4.

1.6 Report organization and study objectives

The main objectives and achievements of the work to date and an explanation of how these are dealt with and presented in the chapters to follow, are given below.

Chapter Two : Literature Review.

The main subject of this chapter is a review of innovative low head equipment and pilot project attempts to harness hydropower from low head sites, with emphasis being given to propeller turbine technology. The advantages and disadvantages of each design are identified and discussed and conclusions are drawn with regard to the design features of a successful low head propeller turbine design.

Chapter Three : Design / Fabrication of the Propeltric Unit.

The various design layouts for a propeltric unit are explained. Conventional methods of axial flow water turbine design are compared with the proposed design method. The design of the runner blades using constant thickness metal sheets is introduced as well as the design features of the guide vanes, the spiral inlet system and the draft tube. Problems with fabrication are analyzed and possible alternatives are introduced and discussed. The chapter also introduces the design of a non contact seal.

Chapter Four : Experimental Work and Evaluation of Design.

The set up of an appropriate test rig for the performance evaluation of the propeltric unit is described alongside the test procedures followed during the course of the experimental work. The performance and test results of the propeller turbine are discussed and a detailed analysis of hydraulic losses and the mechanisms associated with them is presented. The design approach used in Chapter 3 is evaluated and solutions to particular problems related to fabrication and decrease in performance are given. The chapter concludes with a cost analysis of the prototype propeltric unit and a cost/kW comparison is made with a Pelton wheel manufactured by a Nepali workshop.

Chapter Five : Conclusions and Recommendations for Further work.

Based on the discussion of results and design evaluation, conclusions are drawn for the methodology used and the achievements of this research project. In addition, suggestions for further work based on the findings of the experimental work are given.

Chapter 2

Literature Review

"The first precept was never to accept a thing as true until I knew it as such without a single doubt".

Rene Descartes

2.1. Chapter brief

Before considering the development of a prototype propeltric unit it is useful to review the body of knowledge that has built up from the design and development of low head propeller turbine technology across the world. Though there are country specific variations, there are also common findings which can guide decision making for further developments. The work described here is a review of novel devices and particularly propeller turbine designs that have been developed for low head micro hydro power generation in various parts of the world. The reasons that these devices and turbines are attractive for utilizing low head micro hydro potential are discussed as well as the major drawbacks associated with their design and use in developing countries.

2.2. Review of innovative low head equipment

Attempts to develop low head micro hydro equipment for power generation are numerous. The most recent innovations in small scale hydropower systems are in the use of composite materials and plastics, and the development of more efficient turbine-generator systems to bring viable power generation to areas where a lower potential of water head is available. The aim of this section is to discuss some of the most outstanding and novel designs that have been developed to utilize hydropower from sites of less than 5m of head and relate them to this research project.

2.2.1 Water current turbines

Water current turbines use the kinetic energy of fast flowing streams or rivers. Power output is dependent on the *area* of the turbine impeller (swept area) and is given by :

$$\textit{Power output} = \eta_o \times \left\{ \frac{1}{2} \times \rho \times (\textit{swept area}) \times (\textit{velocity})^3 \right\} \qquad \{2.1\}$$

Equation {2.1} is similar to the one used for wind turbines but with the density 1000 times bigger and with the cube of velocities approximately 1000 times smaller. In other words, the power density is as for a wind turbine [Hulscher, 1994].

Output powers are usually modest (less than 1kW) due to the low speeds of the streams and therefore, water current turbines are often used for water lifting or battery charging [Hulscher, 1994].

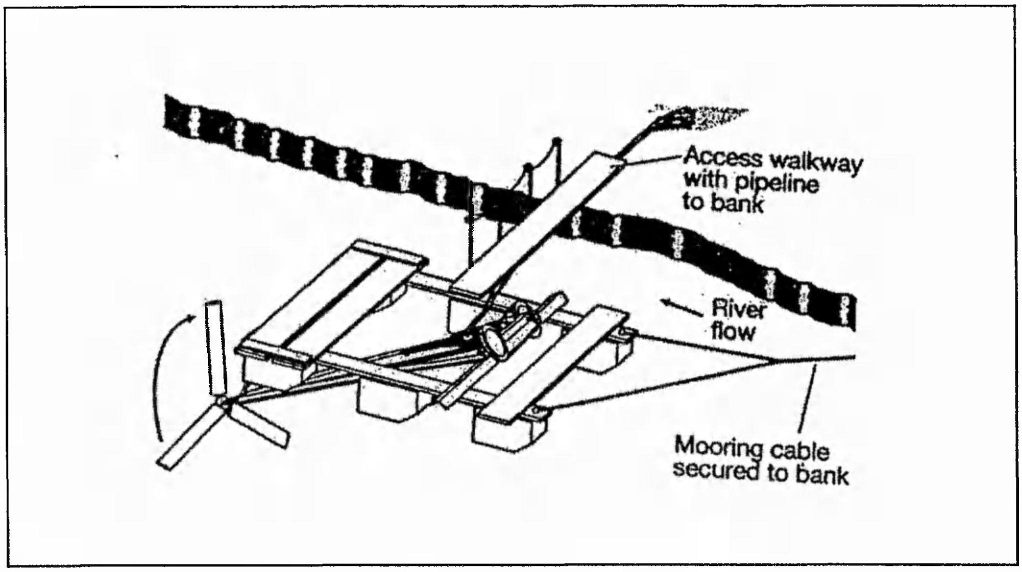


Figure 2.1 : Water current turbine.

Harnessing the power from fast flowing rivers is an attractive idea, but its appeal may be somewhat mitigated with the realization that :

- ☒ The power density of a river flow is very low.
- ☒ Performance is heavily dependent on water velocity.
- ☒ Very high costs.
- ☒ Problems of speed matching between generator and turbine due to the varying water speeds.

2.2.2 Axial flow pumps as turbines

The use of *axial flow pumps* for micro hydro schemes has been investigated by a number of researchers [Cooper, 1981 and Yang, 1983]. Despite their high specific speeds and direct coupling to generators, axial flow pumps are considered to be unsuitable for low head micro hydro schemes for the following reasons :

- ☒ Availability of designs is very limited for schemes less than 20kW [Alatorre - Frenk, 1990 & 1994 and Williams, 1992].

- ☒ Their use is prohibitively expensive for small sizes (15 to 30kW)

- ☒ With vertical axial flow pump designs, the long shafts may be unreliable [Wong, 1987].

2.2.3 Marine Thrusters

Marine thrusters are large diameter shrouded propellers with variable pitch blades and are primarily used for the propulsion and manoeuvering of ships and other off shore applications, such as oil rig supports. According to Huetter,

> *"Thrusters are built for variable speeds in hostile salt water, dirty harbour, flot sam and jet sam - cluster operating environments so they are likely to prove extremely reliable in a constant flow, fresh water application"* [Huetter, 1981].

A study by the Department of Energy and the Energy Research & Applications (ER & A) Institute in the United States aimed to identify the suitability of commercial thrusters for micro hydro schemes and determine their practicality and performance as turbines [Huetter, 1981]. The task was not made easier due to the lack of data with regard to the turbine performance of thrusters. The study published by Huetter [1981], which has used information from Harbormaster and Schottel thruster manufacturers, concluded the following :

- ❑ The predicted average efficiency of a thruster unit would be 65% for a power range of 40 to 537kW.

- ❑ The cost for a thruster assembly (suitable for a 3m scheme), transmission system, support structure and generator was US$258/kW installed.

- ❑ The plant overall installation costs are US$1287/kW installed, including civil works.

- Though the gearboxes, which are an integral part of the thruster unit, could be used to drive electrical generators, there would be a great difficulty in defining the speed of a thruster in power generation mode that can be used to generate electricity.

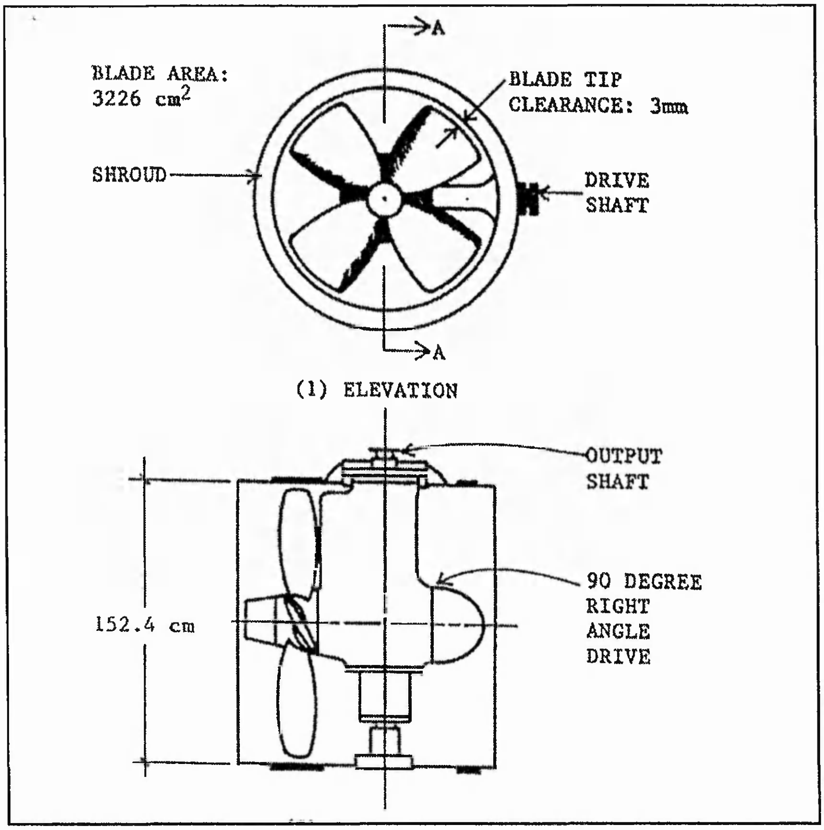


Figure 2.2 : Marine thruster [DeLano, 1984].

Instead of making tests on a model thruster, an attempt was made to install a prototype scheme by Modesto Irrigation District (MID) who were advised by the ER & A Institute. A Harbormaster thruster of 1.52m in diameter with 4 Kaplan type blades was used. The generated power from this unit was predicted to be 428kW at 4.9m of head and a flow rate of 15.65m^3/s. However, tests have shown that the thruster has produced only about half of the power, 225kW at 3.96m of head and a flow rate of 10.93m^3/s. Modifications to the runner blades and guide vanes of the unit only increased the power to 230 kW [DeLano, 1984].

Initial economic analysis of marine thrusters has shown that they are a very attractive solution for low head micro hydro schemes. However, poor performance meant that more research was required into the design of blades and guide vanes. Bearing failure, shaft misalignments, excessive vibration and cavitation lead to the judgment that long term reliability of the package is questionable [DeLano, 1984].

Marine thrusters are unsuitable for use in developing countries for the following reasons:

- ☒ They have to be imported as there is little or no ship building industry in developing countries, especially in countries that are land locked like Nepal.

- ☒ The technology of marine thrusters, based on Kaplan turbines, does not allow manufacturing in developing countries. Moreover, local communities are unlikely to cope with their maintenance, especially with the routine maintenance required by the gearboxes that have to be used as marine thrusters operate at slow speeds.

- ☒ Marine thrusters are unsuitable for direct coupling to generators and from what has been described in literature, matching their operating speed to that of a generator can be a difficult task [Huetter, 1981 and DeLano, 1984].

- ☒ The sizes that are commercially available is also a problem. According to Huetter [1981], their overall dimensions often exceed 1m. Transportation of a machine of this size could be a very difficult and costly task over rough terrain.

2.2.4 The Helical turbine

This newly patented design, has been developed at the Hydro Pneumatic Power Laboratory of the Northeastern University of Boston in the USA. The helical turbine is a reaction turbine capable of providing a high speed unidirectional rotation under a reversible ultra low head (less than 3m) and/or high flow velocities [Gorlov, 1995a].

The turbine was designed for use in hydro pneumatic, hydro, wind and wave power systems. The cost of a mass produced fibre glass or mild steel helical turbine is estimated not to exceed US$120 to 150 per kW installed.

The power range of such units is not defined in the literature provided. However, it is suggested that the helical turbine will be ideal for schemes of up to 1kW [Gorlov, 1995b]. The turbine consists of a wheel with aerofoil - shaped blades (using the NACA - 0021 symmetrical aerofoil) mounted transversely to the direction of the fluid flow for rotation in a plane parallel to the direction of the flow, *Figure 2.3*.

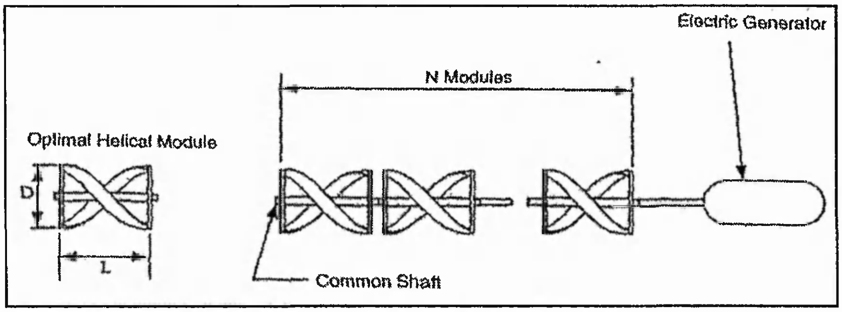


Figure 2.3 : The Helical turbine.

The helical configuration of the blades ensures that a portion of the blades is always positioned perpendicular to the fluid pressure, thereby creating a maximum thrust to spin the turbine and ensuring a continuous speed of rotation with no acceleration or deceleration. The turbines tested, were 0.15m long and had a diameter of 0.09m. The angle of the helix was varied from 90° to 180° and the number of blades from two to seven. The performance of the first models was very poor averaging at about 40%. Though there were many problems faced (mainly excessive vibration) during the testing of the first 20 models that have been developed, according to Gorlov,

> *"Scaled up helical turbines will have efficiencies of at least 50 to 70% running at 1200rpm with no problems of vibrations, in both straight and reversing water flow rates. Though these efficiencies are lower than expected, they are quite good for low head applications*" [Gorlov, 1995b].

Despite the novelty of this design and their intended high rotational speed, the design of these machines is based on aerofoil profiled blades whose dimensional accuracy is very crucial for their efficient operation. In addition, the test results have yet to prove their trouble free operation and with their efficiencies being low, more research is required before considering their use in the remote communities of developing countries.

2.2.5 Hydro pneumatic devices

Novel water to air power conversion systems have been developed specifically for low head hydroelectric power generation. The working principle of such devices is based on water power conversion to air power. The use of air avoids the need for gearboxes and allows the use of low cost air turbines, known as Wells turbines, to drive high speed generators. A typical Wells turbine is shown by *Figure 2.4*.

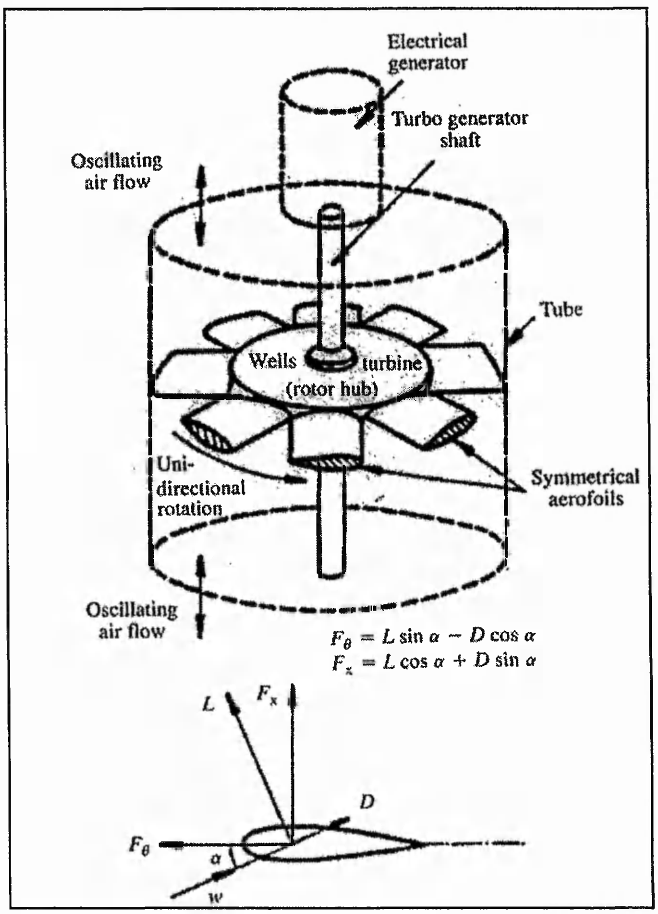


Figure 2.4 : Schematic diagram of the Wells turbine, [Raghunathan, 1995].

The Wells turbine was developed by the Department of Civil Engineering at Queen's University of Belfast and was originally used for wave power generation [Gould, 1989 and Raghunathan, 1995]. In its simplest form, the air turbine rotor consists of several symmetrical aerofoil blades positioned around a central hub, *Figure 2.4*.

With reference to classical aerofoil theory, for a symmetrical aerofoil, the direction of the tangential force, is the same for both positive and negative angles of incidence. So, if such aerofoil blades are positioned around the axis of rotation, they will always rotate in the same direction, regardless of the air flow direction. Wells turbines have been used in three novel hydro pneumatic devices, the *box*, the *bag* and the *syphogen systems*.

The pneumatic box system [Gould, 1989 and Jennings, 1988] consists of an enclosed box or chamber that is filled with water through the inlet valve and drives air out through the Wells turbine. *Figure 2.4* shows the schematic of a pneumatic box system.

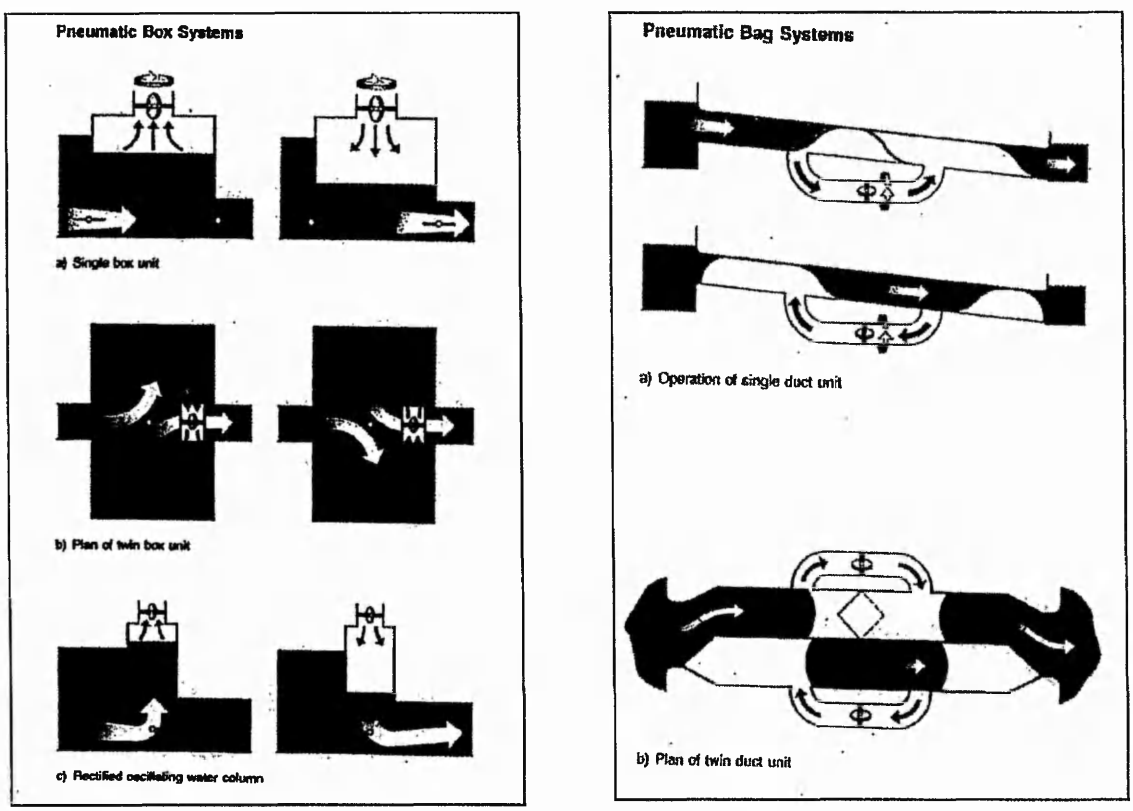


Figure 2.5 : The pneumatic box and pneumatic bag system.

When the box is nearly filled up with water the inlet valve is shut and the exit valve is opened. Air is therefore drawn back into the chamber through the turbine, which will still rotate in the same direction. However, due to the mechanical complexity of the valves and the varying power output of the system, the *pneumatic bag system* was developed [Bellamy, 1989] which uses flexible membranes instead of mechanical valves, which are placed inside an inclined duct.

A demonstration prototype *pneumatic bag system* was installed at Borrowash on the River Derwent in the late 1980's. However, reliability problems with the membrane and the variations in power output have led to its replacement by a Kaplan turbine.

The *Syphogen*, a more recent development, is claimed to be a much better design with regard to simplicity of manufacture, cost and has no moving parts in water [Bellamy, 1995]. In this system, air is drawn into a venturi siphon structure within a water duct to provide air power that drives an air turbine which is coupled to a generator as shown by *Figure 2.6*. Despite the simplicity and originality of the design, the major drawback of this idea that if too much air is drawn into the system, the venturi flow is disrupted and the siphon effect no longer exists. Moreover, the system is not self starting. Tests of a 22 kW prototype *Syphogen*, with a theoretical water to air efficiency of 50%, have shown a maximum efficiency of 30% [Bellamy, 1995].

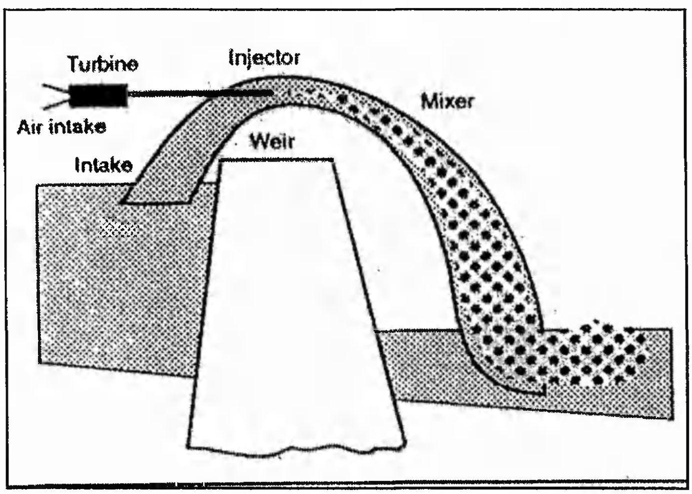


Figure 2.6 : Schematic configuration of the Syphogen hydro system.

The devices described in this section are not appropriate for use in developing countries. First of all, their design is very sophisticated and complex. Due to the high standards required by the design, advanced manufacturing skills and methods with very high costs are also required. Wells turbines make use of aerofoil shaped blades that are difficult to make. For most of these designs, there is no evidence of use other than the prototypes described in literature. In addition, with the variety of reliability problems mentioned in literature, they are clearly unsuitable for use in developing countries.

2.2.6 Positive displacement low head hydropower machines

Two types of positive displacement low head hydropower machines have been commercially developed by AUR Hydropower Ltd. These machines are designed to operate at heads in the range of 0.5-3m.

2.2.6.1 The AUR water engine

Named after its inventor (Mr. Allister Ure Reid), this new machine, shown in *Figure 2.7* is able to harness power available in rivers, canals and estuaries at very low heads.

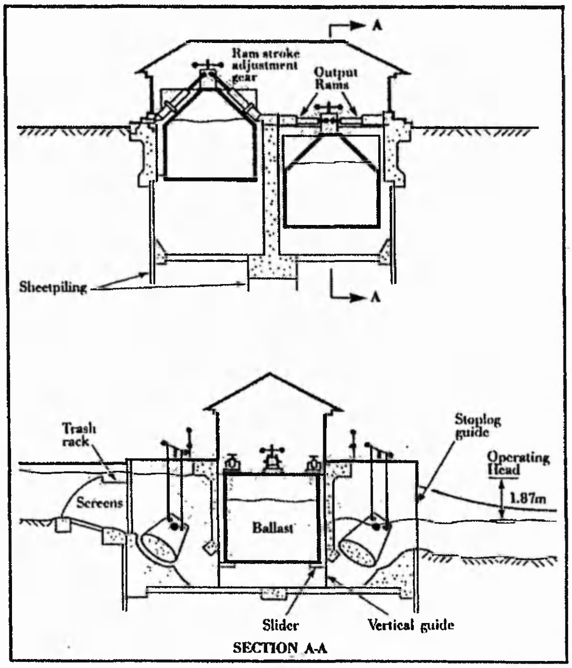


Figure 2.7 : Commercial AUR engine - general arrangement.

Energy is extracted from the water by harnessing the work done against an applied resistance, by the vertical displacement of a float in a chamber [Wilson, 1984]. Water flow into the chamber from the higher level and out to the lower level is controlled by a gate system at the base of the chamber. The float is linked to a power take off mechanism which incorporates a number of hydraulic rams. As the float responds to changes of water level in the chamber, so the rams deliver a supply of pressurized fluid to the load. A prototype 70kW AUR water engine designed by Salford University, was installed and tested over an extended period of time with no serious mechanical failure [AUR, 1991]. The mean overall efficiency during the test program was measured at 55% with an estimated cost of US$3000/kW installed.

2.2.6.2 The STO (Salford Transverse Oscillating) machine

This was invented during the development work on the AUR water engine by Drs. G. N. Bullock and E. M. Wilson [Wilson, 1984]. Like the AUR water engine, the STO is a positive displacement machine for the production of energy at sites with 0.5-3m of head. The patent rights have been acquired by AUR Hydropower Ltd., and the development work has been carried out at Salford University.

The device consists of a parallel double wall barrage built across the direction of flow, to make use of or create a hydraulic differential. At regular intervals across the barrage are gaps called cells. The purpose of the cells is to direct water form the higher level to the lower level. Paddles are placed in the cells to prevent free passage of water from upstream to down stream. There is thus a net hydraulic force on the paddle in the direction the water moves, causing it to be pushed to the end of the cell. By linking the paddles at the top to a carriage and synchronizing gate action, it is possible to make the carriage and the paddles oscillate from end to end of the cells, along the barrage transversely to the direction of the water flow. A power take off rod links the paddles, which is linked to a double acting hydraulic ram. A hydraulic motor can be used to drive a generator.

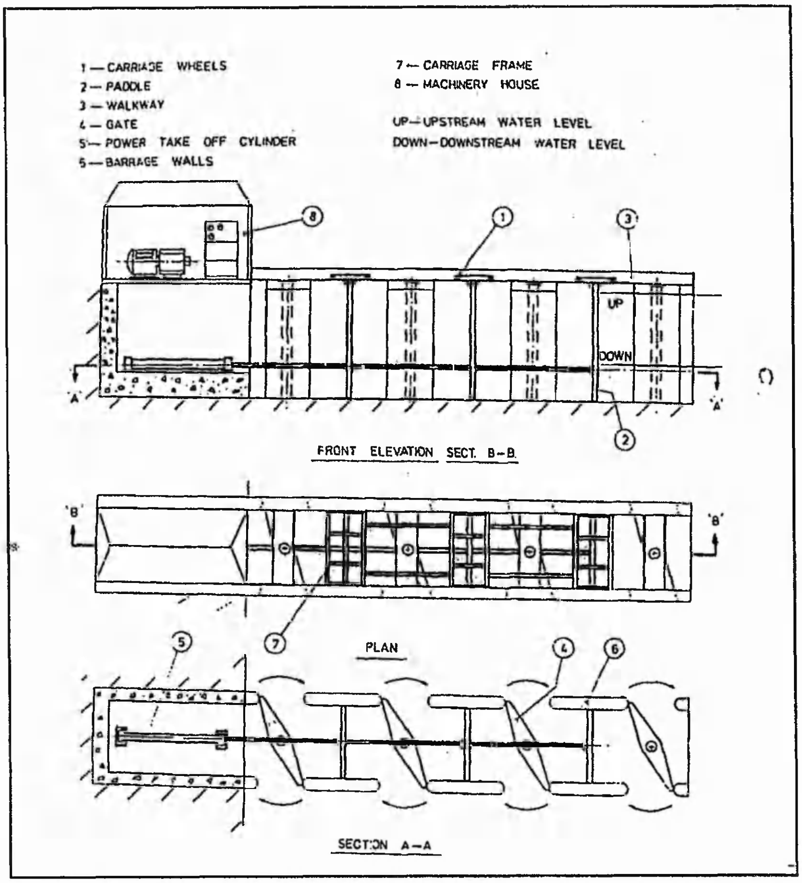


Figure 2.8 : Salford Transverse oscillator.

2.2.6.3 Conclusions on the positive displacement machines

The devices described in this section are novel and unique from an engineering point of view. However, they are unsuitable for use in developing countries for the following reasons :

- ☒ Considerable engineering background and skills are required for their development and production.
- ☒ The range of application that is described in literature is not suitable for *domestic schemes*.
- ☒ The civil works involved for the installation of such machines is considerable. This is a major drawback especially to those areas where material and equipment have to be transported over rough terrain and long distances.

2.3. Review of low head propeller turbine designs.

The designs that are described in this section, cover a wide range of design and manufacturing approaches. The majority of them were one off examples and no evidence exists of their successful implementation by field workers.

2.3.1 Chinese micro hydro generators

The leading market and manufacturer of small, low head hydropower units is China [Waltham, 1996 and Xuein, 1989]. According to Xuein,

> "*Micro hydropower systems, in the 650W to 12kW range, are produced in about 50 factories in seven provinces in Southern China. The installation of very small units to serve a single family is said to be spreading in most of the provinces of China especially of the small family size units, with power output up to 2 kW, which can be carried home on a bicycle*" [Xuein, 1989].

These units come in a package form, with axial guide vanes and a fixed geometry turbine runner directly coupled to a single phase permanent magnet synchronous generator. Their operating head range is between two and four meters. They weigh no more than 30 to 50kg and cost around US$300/kW[1] installed. Small turbine-generator units are exported to many countries of the Far East for the electrification of rural households.

The designs of these units are characterized by low manufacturing quality and technical standards and their mechanical reliability is questionable [Green, 1993a & b and Waltham, 1996]. Problems are associated with one or more of the following :

- ☒ Silt erosion and cavitation damage.
- ☒ Poor hydraulic design of the flow passages.
- ☒ Poor rotor efficiency.

---

[1] This price is for the rated power, refer to section 2.3.2.

2.3.2 Family Hydro in Vietnam

In the early days of micro hydro development, Vietnam was one of the major consumers of Chinese technology. According to Green,

> *"Family hydro installations, which comprise of a 50W to 1kW turbine - generator sets, are being widely used in Vietnam, where approximately 3000 sets have been installed. They have the advantage of being easy to set up and having a low capital price. They can be purchased in the market place for as little as US$28 for a turbine and generator set (US$300/kW installed) which will supply in the region of 80W"* [Green, 1993a].

Today, similar equipment is being manufactured in Hanoi by the Renewable Energy Research Center (RERC). RERC is presently manufacturing 200W and 1kW family hydro sets which are sold locally for US$75 and US$350 respectively [Green, 1993a]. The 200W units operate at 1.5 to 2.5m of head and require a flow rate of about $0.02m^3/s$ whereas the 1kW units have a head range between 2 and 4m and a flow rate of around $0.08m^3/s$. Though more expensive than those imported from China, the Vietnamese units are more efficient and provide greater power outputs [Nguyen, 1991].

The Chinese equipment is often rated at a capacity higher than actually delivers. According to Green,

> *"The rated capacity of one Chinese set tested was 200W, though the maximum output obtained in the test was 84W"* [Green, 1993a].

The fixed geometry units made by RERC have 4 runner blades which are cast as a whole piece onto the runner hub. The blades have aerofoil shapes resembling those used for symmetrical profiles. A 300W unit, the MTD 300 Family Hydro Power Unit, has recently been tested here in the UK. The basic unit, turbine and generator, stands at 0.6m and comes with a draft tube adding between 1.5 and 2m of extra length, *Figure 2.9*. The peak efficiency of the unit was only 40% [Scott, 1995].

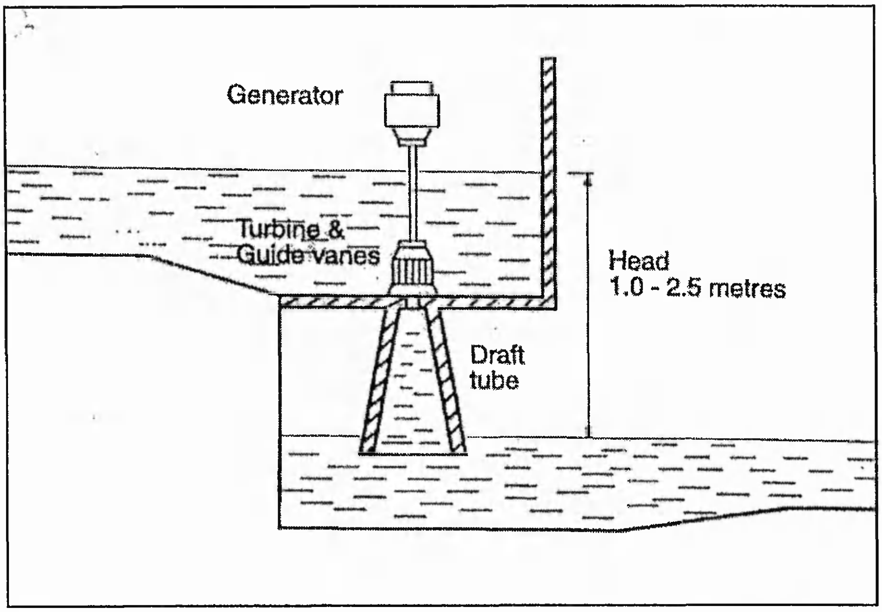


Figure 2.9 : Schematic of a Vietnamese Family hydro unit.

2.3.3 Conclusions on the Family Units from China and Vietnam

The development of these units is a significant achievement towards the efforts for electrification of the rural areas of developing countries. The units described have a number of advantages which can be summarized as follows :

- ☑ Low cost.
- ☑ Compact design.
- ☑ Direct coupling of generators to the turbine shaft.
- ☑ Ease of installation and operation.

The major drawbacks of these units can be summarized as follows :

- ☒ Design problems associated with :
    - ♦ Low efficiencies [Green, 1993a and Scott, 1995].
    - ♦ Reliability problems with bearings and seals [Green, 1993a].

☒ Problems associated with local manufacture :

- Standard sizes are not used.

- The runner hub and blades are cast as one piece. This is a problem with small workshops that are not capable of manufacturing large pieces or where the required skills are not available.

- The use of blades with aerofoil shapes requires highly skilled people and advanced manufacturing techniques that are often not available in small workshops in developing countries.

2.3.4 Axial flow turbine in Indonesia

Indonesia has approached propeller turbines in two ways : through imports of Chinese units; and through the development of innovative propeller turbine designs by the Energy Research Laboratory. According to a recent report, the Chinese 1kW units are currently gaining attention by many rural communities [Sunarto, 1991], but due to its high initial cost (US$1700/kW installed for the imported units), potential users can only benefit when local authorities and governments subsidize the units.

The Energy Research Laboratory of the Republic of Indonesia, has developed a propeller turbine for low head, small capacity hydropower plant for village electrification shown in *Figure 2.10.* The turbine runner of four blades was designed to run at 1000rpm, an operating head of 2m, 0.15m^3/s of flow rate and producing 2.66kW at 86% hydraulic theoretical efficiency. The turbine blades were made of the Göttingen - 428 profile [Susanto, 1981].

The major drawbacks of this design can be summarized as follows :

☒ Cast iron was used for the prototype development. This technique requires a considerable level of skills and a relatively well equipped foundry.

☒ Cast iron runners are heavier than mild steel fabricated runners and therefore difficult to carry over rough terrain.

- The complexity of the runner blade profiles prohibits fabrication and requires workshops equipped with milling machines and highly skilled people.

This particular turbine design was developed with the intention to use it as part of a village electrification scheme, but there has been no further published evidence on design improvements, testing or installation of such a unit.

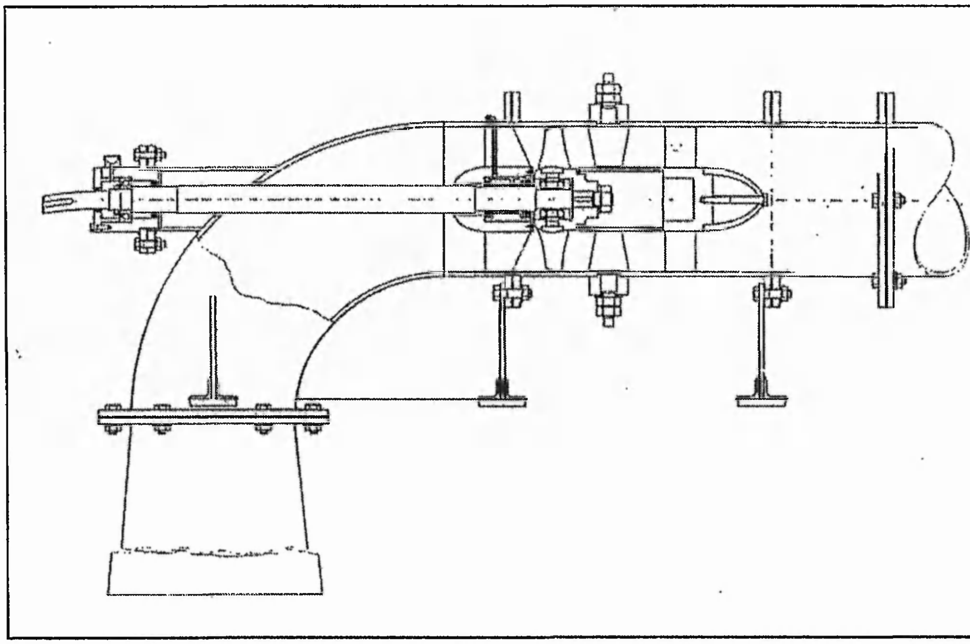


Figure 2.10 : Prototype axial flow turbine in Indonesia.

2.3.5 Low head propeller turbines from Bangalore - India

2.3.5.1 Propeller turbine with a helical bladed runner

The turbine was developed by the Department of Civil Engineering at the Indian Institute of Science in Bangalore, for a 5kW low head micro hydro scheme.

According to Rao,

> *"A simplified runner profile for a low head axial flow micro hydro turbine, which can be more economical than conventional machine runners with aerofoil shaped blades, has been developed and tested in India"* [Rao, 1986].

The eight runner blades of this turbine have a constant thickness of 3mm all along the blade and were machined out of a solid piece of metal. The turbine was designed to operate at 850rpm and 5m of head. The runner tip and hub diameters were 238mm and 100mm respectively. Tests of the turbine have shown good cavitation resistance with the overall efficiency of the turbine reaching 67% [Rao, 1986, Rao, 1988a & b].

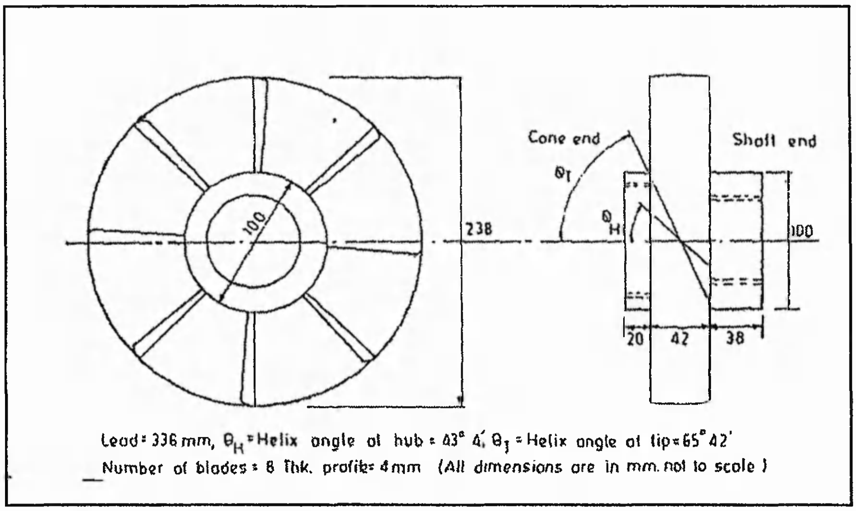


Figure 2.11 : The prototype turbine runner from Bangalore.

The propeller turbine from India is an example of a prototype developed in a university laboratory with no further work with field tests and improvements. The major drawbacks of this particular design can be summarized as :

- ☒ The turbine's low operating speed does not favour direct coupling to a generator.
- ☒ Though the runner blades are of constant thickness, machining was used to give the blades their helical form.
- ☒ The dimensions that have been used are not of standard sizes.
- ☒ Technology transfer has not been considered during the conception of the idea of the helical bladed runner. Despite the good performance of the turbine, ease of manufacturing has not been considered.

2.3.5.2 Ultra low head turbine for small canal drops

This is another design for low head power generation developed by the Turbomachines Laboratory at the Indian Institute of Science in Bangalore. It's power rating is much higher than those previously described, reaching the value of 40kW [Soundranayagan, 1988]. However, it demonstrates the use of propeller turbines for low head micro hydro schemes. The Göttingen 682 profile was chosen for the four runner blades. The turbine was designed for net head of 2.5m and a flow rate of $2m^3/s$ operating at 300rpm. The runner has a tip diameter of 0.85m with a hub to tip ratio of 0.45. Laboratory tests of the turbine have shown an efficiency of about 80% [Soundranayagan, 1988]. Though this particular design operates at a very high efficiency, it has the following disadvantages :

- ☒ The complexity of blade profiles that have been used does not allow the turbine to be fabricated.
- ☒ Special die cast tools and skills are required for accomplishing the accuracy of the blade profiles.
- ☒ The use of a belt drive or a gearbox would be inevitable as the turbine was designed to run at only 300rpm.

2.3.6 Innovative micro hydro plant - Papua, New Guinea

A prototype propeller turbine directly coupled to a generator-gearbox assembly, see *Figure 2.12*, has been developed by the Mechanical Engineering Department of the Papua New Guinea (PNG) University of Technology. The system is a 12V DC supply with the electrical power output reaching 200W. The turbine was designed for an operating speed of 200rpm and has no guide vanes. The eight runner blades have a constant thickness of 3mm and were made on a Computer Numerical Control (CNC) milling machine. The tip and hub diameters were 0.2m and 0.08m respectively. The turbine was designed to operate at a head of 1.5m and a flow rate of $0.06m^3/s$.

The theoretical power into the turbine is 883W. However, due to the simplicity in the design, the hydraulic losses are high and only 200W are produced [Ranatunga, 1991]. The disadvantages of this design can be summarized as follows :

- ☒ The turbine has poor performance, 22.7%, despite the accuracy of manufacturing.

- ☒ The use of a gearbox for driving the generator, would make transportation over rough terrain and maintenance difficult in developing countries.

- ☒ Fabrication of the runner blades would be difficult since the eight runner blades are arranged on a hub diameter of just 0.08m

- ☒ The use of a long turbine shaft.

- ☒ Reliability problems of the seal, which failed after only a couple of hours of operation, caused extensive damage to the generator.

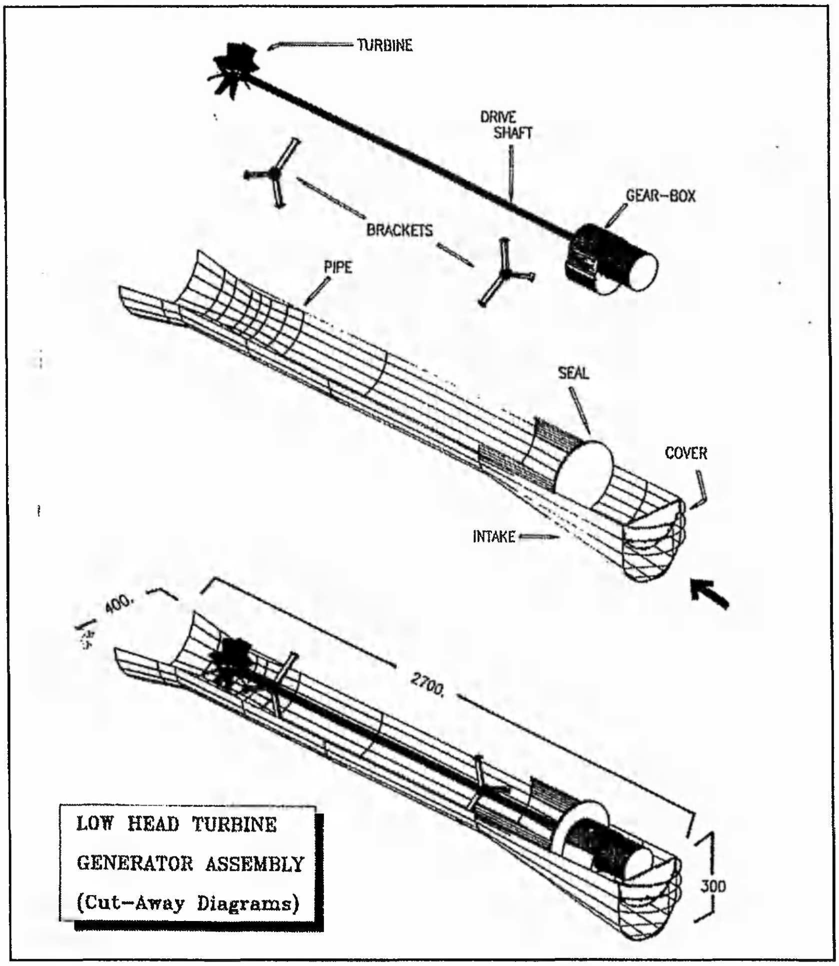


Figure 2.12 : Low head turbine generator assembly.

2.3.7 Propeller turbine for isolated micro hydro - Venezuela

This turbine was developed by Electrificación del Caroní (EDELCA) Micro hydro Department in Caracas for sites whose flow remains constant. The five runner blades and eleven guide vanes (fixed and movable respectively), were made of steel sheets bent and twisted. This is a 15kW machine for sites of 2.5m of head [Escalona, 1992]. The turbine was designed for a speed of 420rpm and a belt drive system is therefore used to drive the generator.

This micro hydro scheme was set up as a pilot project in Venezuella, but no further work has been published with regard to its progress and development. The major drawbacks related to the scheme and the turbine design can be summarized as :

- ☒ The scheme is uneconomical. It has an estimated cost of US$7163/kW installed while viable schemes cost on average of US$1500/kW installed.
- ☒ Low operating speeds not suitable for direct drive of a generator.
- ☒ Exact figures of the turbine efficiency are not available.
- ☒ The use of adjustable guide vanes adds to the complexity of the design. Moreover, manufacturing time and therefore costs are higher.

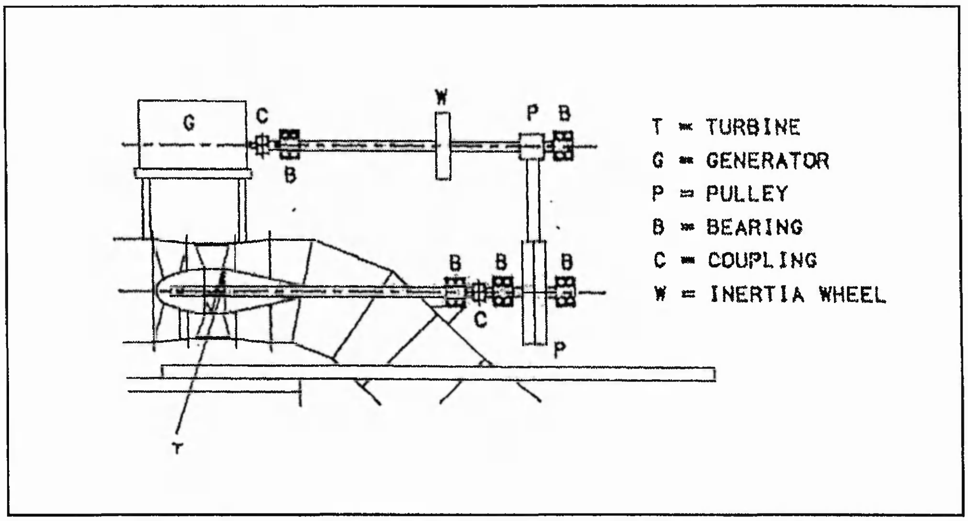


Figure 2.13 : Propeller turbine from Venezuela.

2.3.8 Low head turbine development - Peru

A propeller turbine has been developed by ITDG in Peru for low head sites [Viani, 1992 & 1993]. This prototype propeller turbine, which cost US$7000 to make, was designed for a net head of 4m and a flow rate of $0.49m^3/s$. The runner, which was made locally, was to designed to operate at 402rpm and has a tip diameter of 437mm with a hub to tip ratio of 0.5. It consists of four blades whose shape is based on the NACA 4418 and NACA 4406 profiles at the hub and middle blades sections respectively [Viani, 1990 and 1993]. The tip has a thin curved section.

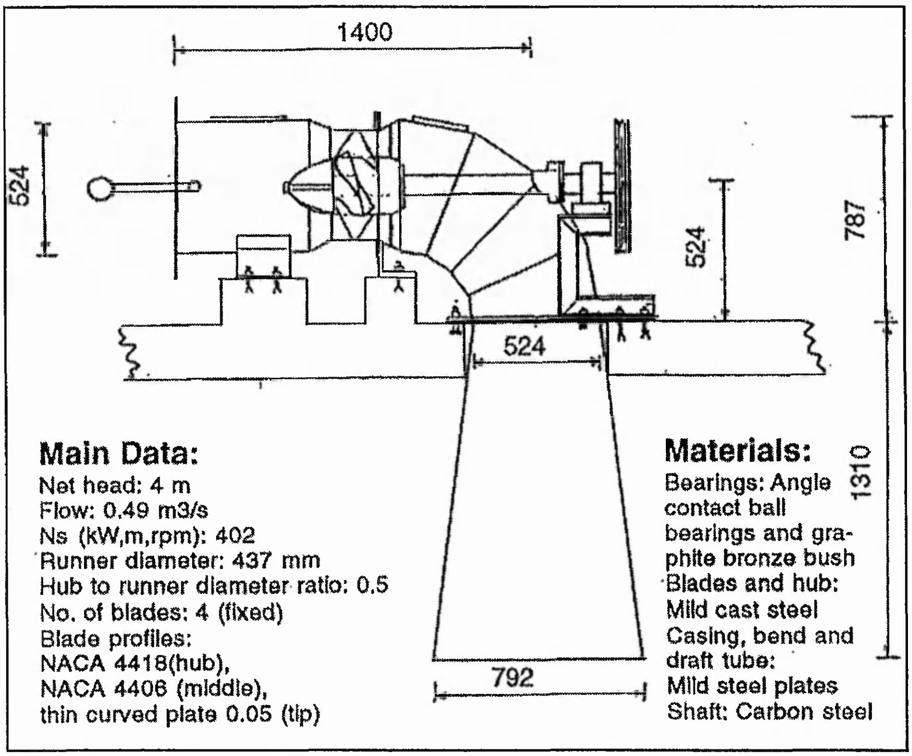


Figure 2.14 : Main dimensions of the propeller prototype.

Despite a peak turbine efficiency of 67%, the design of this propeller turbine has the following drawbacks :

☒ High prototype cost, US$2000 higher than a similar cross flow machine [Viani, 1992].

- ☒ The complexity of the blade profiles used prohibit fabrication and limit the manufacturing of such a turbine to highly skilled workshops that are usually found only in big cities in the developing countries.

- ☒ The use of a contact seal, which failed after 30 hour of operation, suggests that continuous operation of the turbine would be seriously affected when operating in silty environments.

2.3.9 An innovative low head hydropower unit - Ireland

The Irish company Polyturbine Design Ltd. has developed a propeller turbine which uses an all plastic water way guide vane assembly and reinforced plastic runner blades [Holmen, 1991]. The design flow rate for the 700mm runner, which is operating at 510rpm, is 2.6m^3/s for 2m of net head. The overall efficiency including generator losses is 70% where as for another unit operating at 3.2m of head and 69kW output the efficiency goes as high as 85%. One unit is used for the power requirements of the Mechanical Engineering Department of the University College Galway in Ireland.

Despite the high efficiencies obtained by these turbines, their design is inappropriate for use in developing countries for the following reasons :

- ☒ The cost for producing these units becomes very high for low heads. For example for a unit to operate at 3m of head the cost will be US$1785/kW installed whereas at 2m will the cost will be US$5571/kW installed.

- ☒ The use of moulded plastic limits their production to industrialized countries.

- ☒ Reliability in silty environments has yet to be proved.

- ☒ The design of such turbines has been developed for high power outputs and mass production in high volumes rather than small scale hydroelectric schemes suitable for local manufacture.

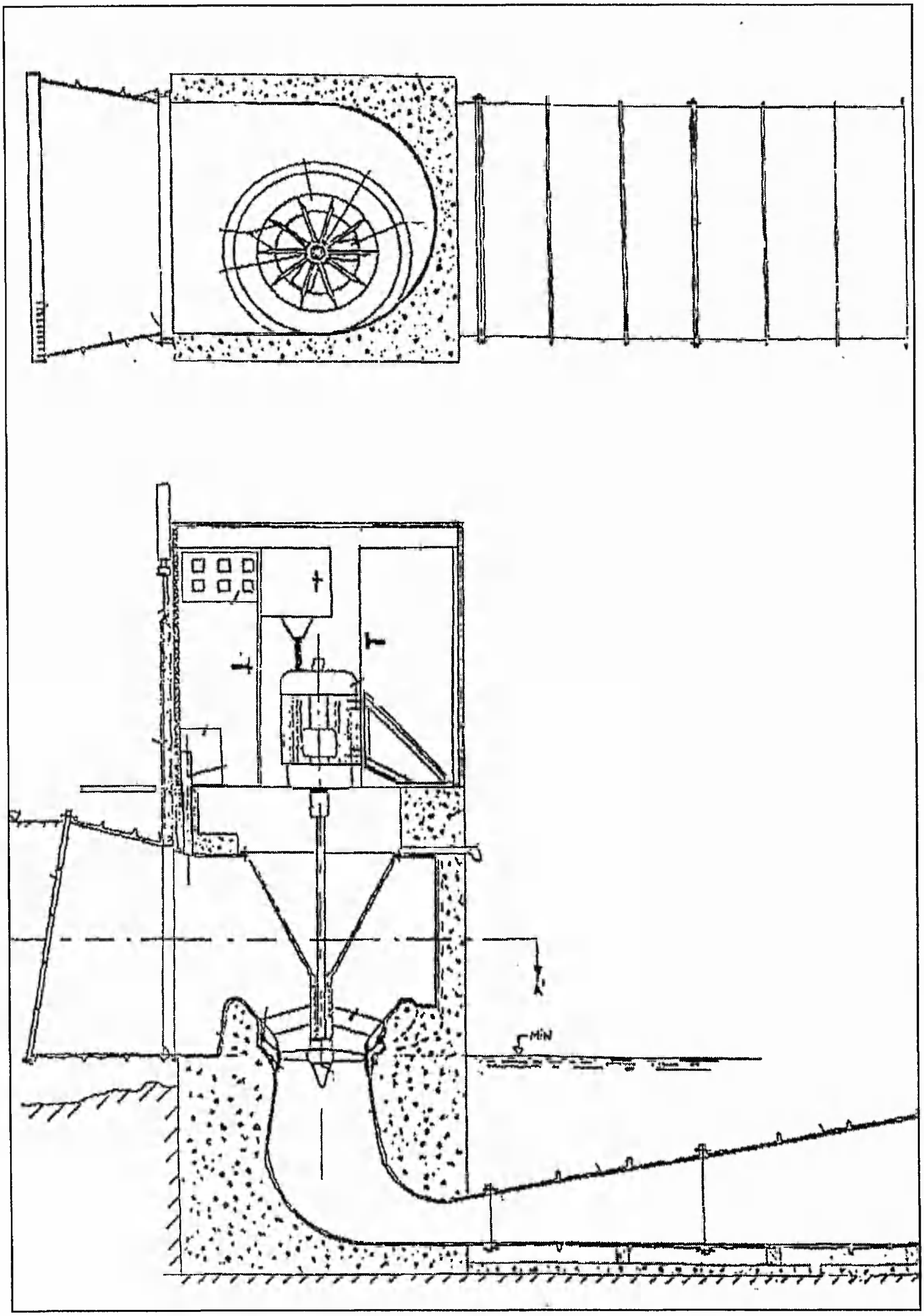


Figure 2.15 : The Polyturbine from Ireland.

2.3.10 Design of small water turbines for farms and small communities - USA

A prototype 5.5kW propeller turbine has been developed by the Massachusetts Institute of Technology [Durali, 1976]. The designer of this prototype has adopted a very different approach from what has been described so far. The runner blades, guide vanes and housing were all made of glass reinforced polyester resin (GRP). This was chosen in order to minimize difficulties associated with casting of steel [Ho, 1979] and to achieve better surface finish for blades and guide vanes.

The turbine was designed to operate at 10m of head, a flow rate of $0.049m^3/s$ and had runner tip diameter of 235mm. The estimated turbine-generator efficiency was 80%. The sixteen guide vanes and fifteen runner blades-rotating at 540rpm, were given a profile similar to that used in gas turbines. A hub to tip ratio of 0.8 was used in order to achieve minimum twist for the blades and facilitate manufacturing.

Manufacturing and performance evaluation of the first prototype suggest that there are major problems with the method that has been used for the hydraulic and mechanical design of the turbine as well as with the materials used [Ho, 1979]. The use of this turbine design for village hydroelectric schemes is inappropriate for the following reasons :

- ☒ Despite the accuracy of the design and manufacturing methods, the overall turbine-generator efficiency is 45%.
- ☒ The complexity of the flow surfaces, materials used and the high engineering standards. Workshops in developing countries would be unable to cope with the manufacturing processes and materials used.
- ☒ The costs involved for special tooling that would enable manufacturing in developing countries.
- ☒ Low operating speeds.
- ☒ Dimensions used not tailored to those of standard pipe work fittings.

☒ Excessive wear was observed during tests due to the presence of suspended matter in water suggests that the turbine is unsuitable for silty environments.

☒ Despite the claims of the designer of this unit [Durali, 1976] that it would be appropriate for remote communities, no attempt was made for technology transfer. On the contrary the turbine was developed with the prospect of being exported to these developing countries.

2.3.11 Propeller turbine for low head schemes - New Zealand

The turbine that was developed, built, installed and operated by the Department of Mechanical Engineering of the University of Canterbury in Christchurch, is a vertical shaft propeller turbine, as shown by *Figure 2.16* operating at a head of 2.8m and at $0.4m^3/s$. A speed of 612rpm was used for the design speed and the electrical output of the installed turbine was 3.7kW, (overall efficiency of generator and turbine around 34%) [Parker, 1993]. Mild steel was used to fabricate the turbine. It has no scroll casing and the eighteen guide vanes are straight and set at 30° to the radius of the machine. The eight runner blades were cut from flat sheets of metal, refer to *Figure 2.17*.

The draft tube is parallel, along the propeller section, and fitted with flow straightening vanes which also support the lower shaft bearing and a fixed conical streamlining below the rotating hub. Below this section, the draft tube itself is of a conical shape.

According to Parker,

> *"The upper bearing is supported clear of water by a tripod and comprises a self aligning ball thrust bearing. The lower bearing is a water lubricated rubber journal running on a stainless steel sleeve stub shaft with the lubricating water being supplied from the head water through the hollow shaft"* [Parker, 1993].

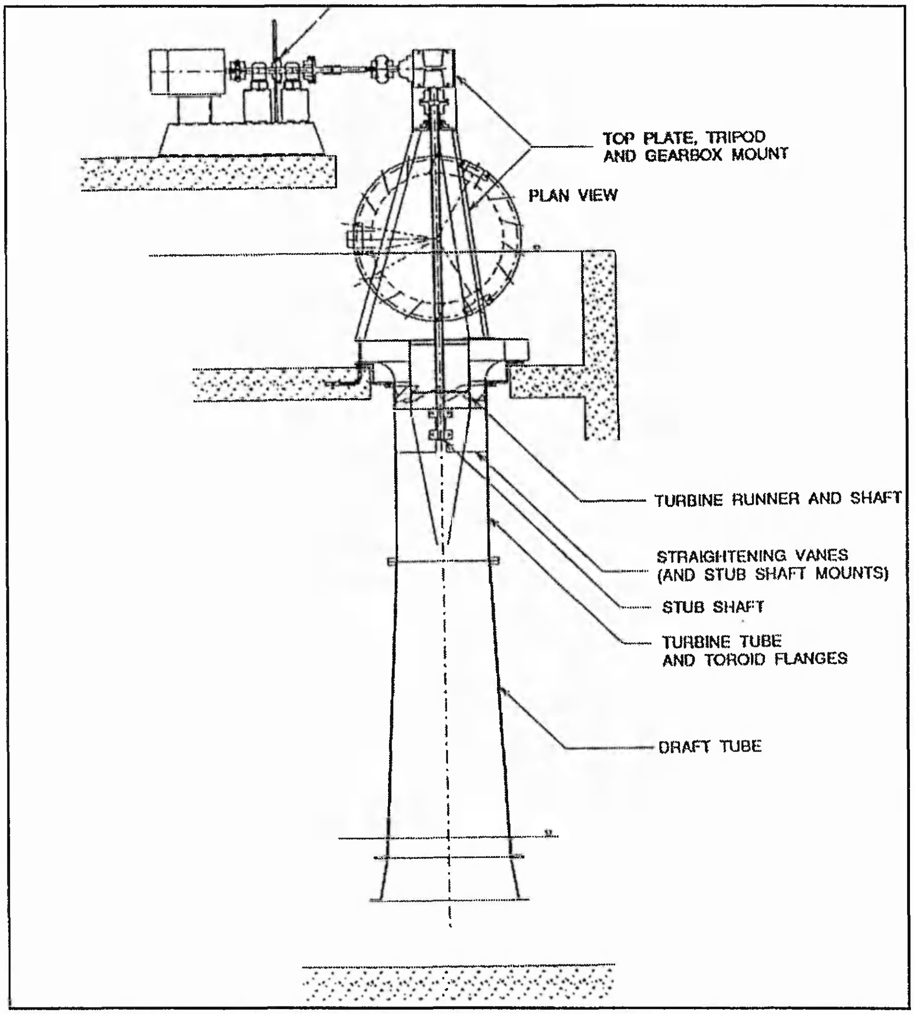


Figure 2.16 : Low head propeller turbine from New Zealand.

The unit was installed with a speed increasing gearbox that converts the 612rpm to 3000rpm. The hub and tip diameters are 270mm and 408mm respectively and the blades have a constant thickness of 3mm. Based on the same approach, a model turbine was developed and tested in a Hydraulics Laboratory of the University of Canterbury. The maximum efficiency of the model turbine was 57% [Parker, 1993 and 1996].

The design of this turbine has achieved a very simple approach with regard to blade design and fabrication and yet reliable and efficient to operate. However according to the literature available, further development work is required in order to increase the turbine power output without any alterations to the simplicity of the design.

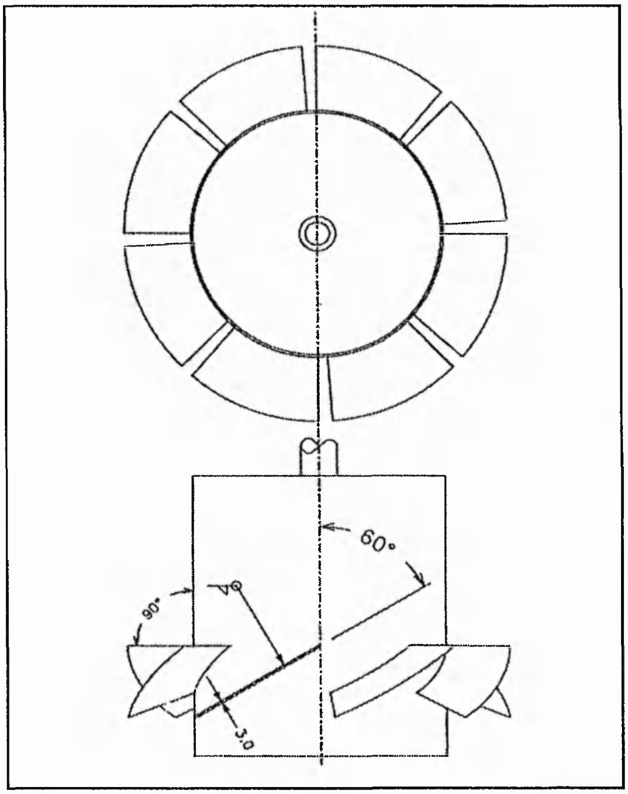


Figure 2.17 : Plan and elevation of the 8 bladed propeller from New Zealand.

Despite using blades made of constant thickness metal sheets, there are a number of disadvantages related to the design of this turbine which can be summarized as follows :

- ☒ Low operating speed does not allow direct coupling to a generator.
- ☒ Standard dimensions were not used during the design process.
- ☒ The prototype turbine is performing worse than the model one.
- ☒ Reliability problems with the water lubricated bearing.

2.3.12 Open flume axial flow turbine - Nottingham Trent University, UK

A prototype 1kW propeller turbine has been designed and constructed by a student of the Mechanical Engineering Department of Nottingham Trent University [Heitz, 1993].

The prototype had five guide vanes and four runner blades which were fabricated using constant thickness stainless steel plates bent and twisted to the desired angles. The turbine was designed to operate at 2100rpm, 2.9m of head and a flow rate of $0.06m^3/s$. A 0.13m turbine runner diameter was used with the hub to tip ratio being 0.4. Tests performed by Mali [1994] indicate that the turbine design is very inefficient and has a maximum efficiency of 20%. A similar turbine was designed and installed at a low head site in London[2] using the same design approach. The turbine was designed to operate at a speed of 650rpm. A belt drive system was used to drive a 4 pole induction generator using a belt drive. Tests carried out by Williams [1995b] show an overall efficiency of 24%.

The turbine designs described in this section were designed and manufactured for use in developing countries. However, field tests suggest that there is room for improvement in the following areas :

- ❑ The hydraulic design of the runner blades. Heitz [1993] has designed the blades assuming that that the lift coefficient would remain constant across the whole blade. As a result, the blades were developed with long and short chord lengths at the hub tip sections respectively, a design approach usually used only for fan design, [Bohl, 1991].

- ❑ The layout of the turbine. Long shafts are required with open flume layouts which often experience vibrations. In addition, incorrect design of the intake tank can cause air to be drawn into the turbine [Mali, 1994].

- ❑ Use of speeds that would allow direct drive of induction generators.

- ❑ Use of standard sizes to allow pipe work fittings to be incorporated into the design and therefore assist fabrication and reduce costs.

---

[2] Described in Appendix B.

2.4. Concluding remarks

Propeller turbines are well suited for low head micro hydroelectric schemes. Their main advantages are :

- ☑ Compact designs due to high specific speeds which enable ease of transportation [Xuein, 1989].
- ☑ Direct coupling to generator sets [Demetriades, 1995a].

A directly coupled *propeller turbine-generator unit* offers several advantages over belt drive systems and gearboxes [Smith, 1992a]. These are :

- ☑ Improved system efficiency.
- ☑ Increased turbine and generator bearing lifetime.
- ☑ Reduced costs.
- ☑ Lower maintenance requirements and simplicity of installation.

The need for low head power generation equipment has led over the last few years to the development of a number of novel devices and water turbines-particularly *propeller turbines*. However, they are inappropriate for use in developing countries because of one or more of the following [Green, 1993b; Waltham, 1995 & 1996 and Woollacott, 1996] :

- ☒ Importing the machines in developing countries is prohibitably expensive.
- ☒ Unsuitable for local manufacture due to high engineering standards and complicated designs. Some of the designs also rely on imported materials.
- ☒ High manufacturing costs.

- Local communities cannot cope with the difficulties of maintenance and repair.

- Reliability problems due to operating conditions, e.g. high silt content in water.

A summary of the design features of the turbines described in section 2.3 is given by *Table 2.1*. The development of the prototype *propeltric unit*, described in Chapter 3, is based upon the knowledge gained by the review of existing low head propeller turbine designs presented in this chapter as well as the experiences from field trips and visits[3]. *Figure 2.18* shows the parameters that are considered for the development of an appropriate propeller turbine suitable for direct coupling to an induction generator based on the work presented by Demetriades [1995b & 1996].

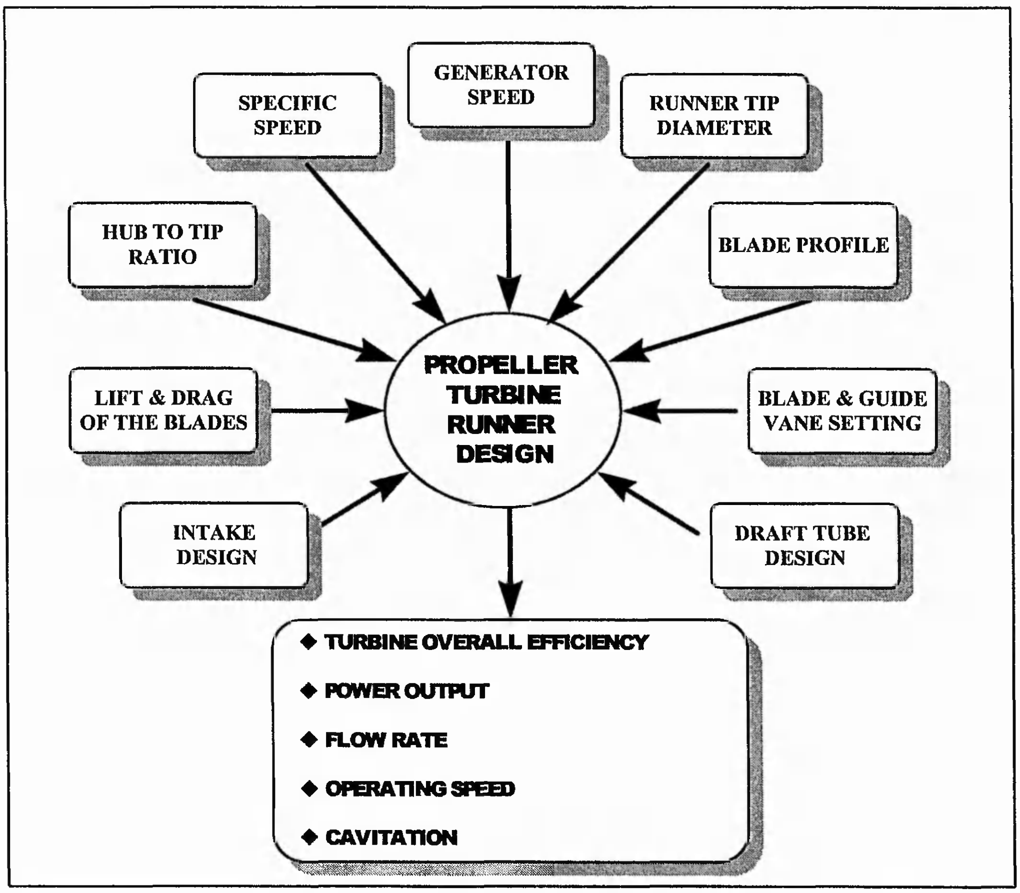


Figure 2.18 : Propeltric unit design parameters.

[3] Details of the visits can be found in Appendix B.

TURBINE ORIGIN	DESIGN SPEED	BLADE PROFILE	RUNNER MANUFACTURE	RELIABILITY OF DESIGN	DIMENSIONS	TURBINE EFFICIENCY	POWER 1-5 kW	COST
China	Direct Drive	Aerofoil	Casting	Poor	Not Standard	30 to 40 %	Below	Low
Vietnam	Direct Drive	Aerofoil	Casting	Good	Not Standard	≈ 50 %	✓	High
Indonesia	Low	Göttinghen	Casting	N/A	Not Standard	N/A	✓	N/A
India (5 kW)	Low	Constant thickness	Milling	N/A	Not Standard	67 %	✓	N/A
India (40 kW)	Low	Göttinghen	Casting	N/A	Not Standard	80 %	Above	High
Papua New Guinea	Low	Constant thickness	Milling	Poor	Not Standard	less 30 %	Below	N/A
Venezuela	Low	Constant thickness	Fabrication	N/A	Not Standard	50 % (estimated)	Above	High
Peru	Low	NACA	Milling	Poor	Not Standard	67 %	✓	High
Ireland	Low	Aerofoil	Plastic moulding	Good	Not Standard	85 %	Above	High
USA	Low	Aerofoil	Plastic moulding	N/A	Not Standard	45 %	Above	High
New Zealand	Low	Sheet metal	Fabrication	N/A	Not Standard	57 %	✓	N/A
UK	High	Sheet metal	Fabrication	Good	Not Standard	24 %	✓	Low

Table 2.1 : Summary of the design features of low head propeller turbines.

Chapter 3

Design / Fabrication

of the Propeltric Unit

"For every problem there is one solution which is simple, neat, and wrong".

Edmund Burke

3.1 Chapter brief

The chapter presents the design methodology used for the development of the prototype propeller turbine whose key features were introduced in Chapter 1. The method described here is a compromise between *optimum efficiency* and what is *easy to design and make*, at *low cost* for *reliable continuous operation* in rural areas of developing countries.

The method used is based on the design theory of axial flow fans, pumps and turbines. One and two dimensional flow theories and models have been used as these are generally simple to use with the minimum data input and computational analysis. Detailed computation of water turbine flow fields using the equations of fluid motion are possible, but significant commitment of engineering and computational time is required. The dimensional accuracy of such mathematical models - sometimes down to a fraction of a degree - can only be achieved by using advanced manufacturing methods usually associated with Computer Aided Design (CAD) and Manufacturing (CAM). As already described in Chapter 1, the success of village hydroelectric schemes depends to a great extent on local manufacture which has a limited level of dimensional accuracy. The use of very accurate mathematical models, even if these are simple enough to be used by field workers, would not be of any benefit for developing this prototype.

During the course of the chapter, reference is made to the innovative propeller turbine designs described in Chapter 2, section 2.3. The major differences between the proposed method with conventional design approaches are identified as well as the advantages and limitations of the proposed method. Problems related to fabrication are explained and suitable solutions are given. The chapter concludes by introducing the development of a non contact seal that would be used to overcome the reliability problems contact seals ecounter due to the silt content in water.

3.2 The gross and net head across a reaction turbine

The site *gross head* is defined by the height between the water level at the intake from the stream or forebay and the water level at the tailrace as it is shown by *Figure 3.1*. However, the turbine cannot fully utilize this *gross head* as part of it is lost due to *friction losses* in the penstock leading to the turbine intake [Nechleba, 1957 and Daugherty, 1985]. The *net head* is therefore given by

$$H_N = H_G - \xi_P = \left[\frac{P_B}{\rho g} + \frac{V_B^2}{2g} + Z_B\right] \qquad \{3.1\}$$

The definition of *net head* across the turbine described here is used by micro hydro field engineers for the design of small scale village hydroelectric schemes [Harvey, 1993]. The definition of *net head* is different to the one defined in other literature [Massey, 1989 and Sayers 1991] as it includes the kinetic energy losses at the exit of the draft tube [Warnick, 1984 and Douglas, 1995]. Detailed analysis of the losses in the turbine unit are given in Chapter 4.

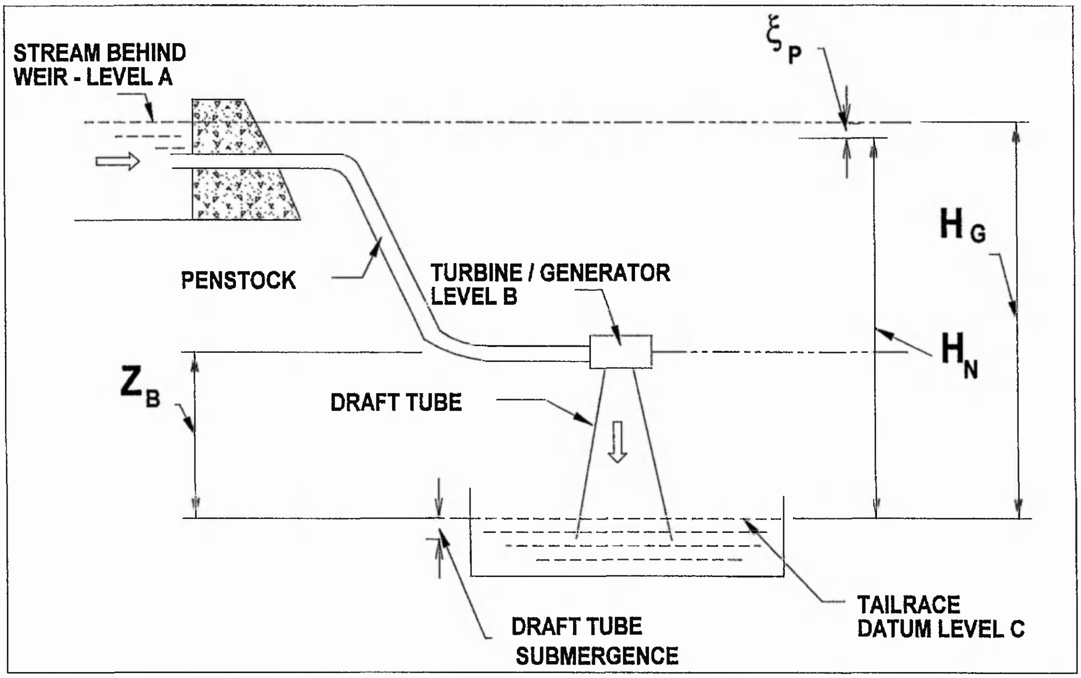


Figure 3.1 : Net head available for a reaction water turbine.

3.3 Propeltric unit layout

The direct coupling of the turbine runner to the generator shaft has a number of advantages with regard to the layout of the proposed unit which can be summarized as follows :

- ❑ No use of speed increasing mechanisms for driving a generator.
- ❑ Compact design due to the high operating speeds.
- ❑ The use of a similar layout to the one used for a Peltric unit.

In the process for deciding the layout for the prototype propeltric unit, there are a number of parameters that need to be considered, which are directly related to the economics and viability of such a unit. These parameters are :

- ❑ Simplicity of blade profile shapes to enable designers and developers overcome the problems related to the complexity of runner blade design and manufacturing.
- ❑ The use of materials and utilization of the skills that are locally available in developing countries.
- ❑ The use of dimensions that would allow standard pipes and fittings to be incorporated into the design.
- ❑ The amount of civil works required.
- ❑ The silty operating conditions and the poor infrastructure for maintenance available in rural areas, suggest that a careful selection of bearings and seals is required for reliability and low cost maintenance on site.
- ❑ Ease of transportation and on-site installation as propeltric units may be installed in remote communities, that are not accessible by road.

In addition to the above, for the prototype design, space limitations of the Fluids Laboratory in the Department of Mechanical Engineering of Nottingham Trent University needed to be taken into account.

3.3.1 Open flume or tube type ?

The *open flume* and the *tube type* layouts, which are shown in *Figure 3.2* and *Figure 3.3* respectively, can be used for the development of the prototype propeltric unit. *Table 3.1* presents a summary of the advantages and disadvantages of the two layouts.

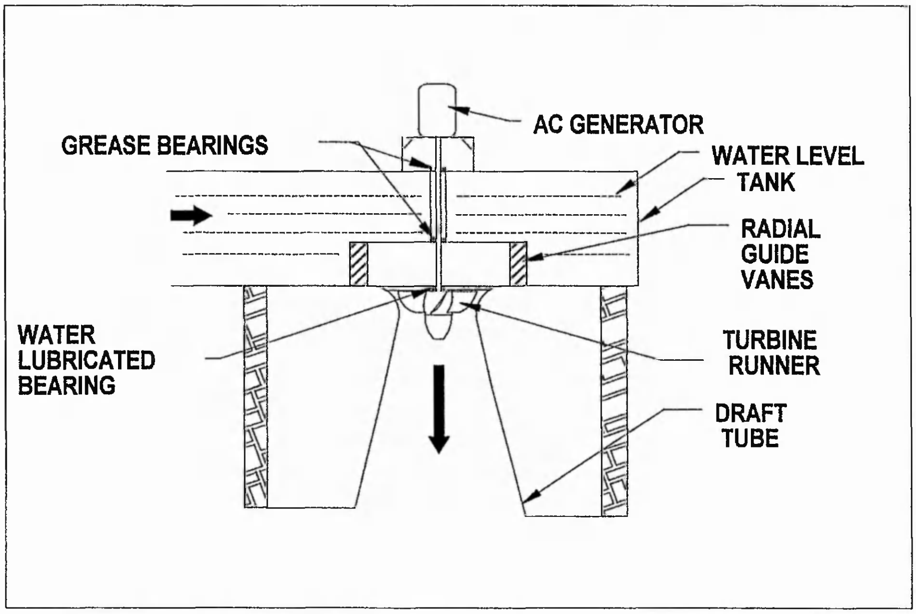


Figure 3.2 : Open flume.

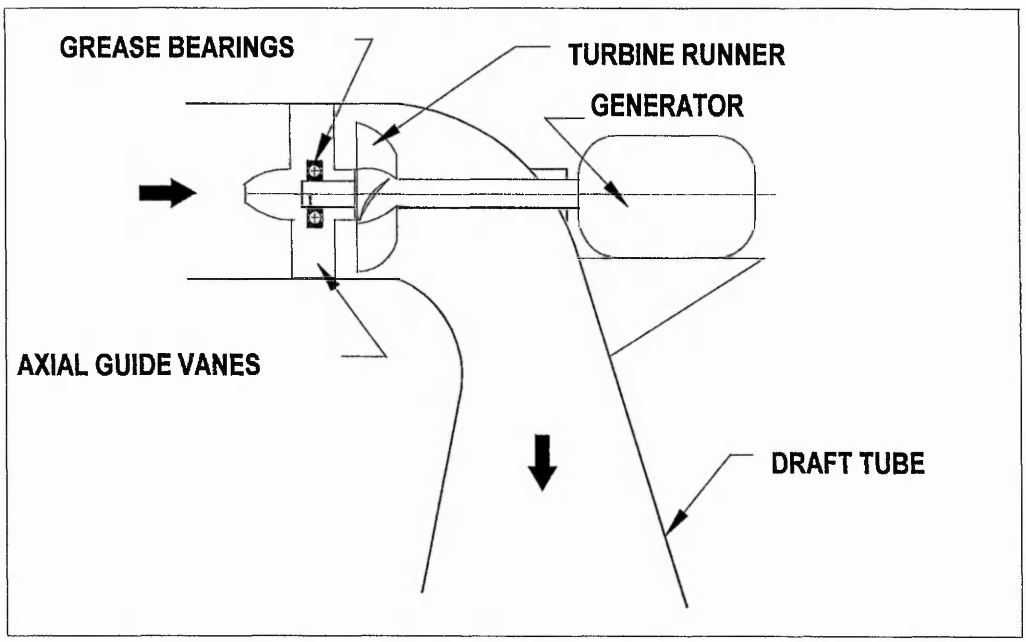


Figure 3.3 : Tube layout.

The *open flume type,* consists of a big tank or a chamber at the end of a canal with the turbine installed at the bottom of the tank. Unless a very expensive waterproof generator is used, the generator must be installed out of reach of water. This results in using very long shafts which require careful alignment to avoid vibrations. The *open flume type* does not require a seal between the generator and the turbine. Though a very short length of penstock is required, significant amount of civil works is necessary to provide support for the tank and construction of the canal for diverting the flow from a small river.

The *tube turbine* layout consists of a small section of a pipe in which the turbine sits. A short penstock is used and since both turbine and generator come as a compact unit, pre - fabrication and installation on site are easier in comparison with the *open flume type*, especially for units of small size and high specific speeds. The major drawback of this arrangement is the seal requirement.

OPEN FLUME	**TUBE TYPE**
<u>Advantages</u>	<u>Advantages</u>
☑ No seal is required.	☑ Short shaft lengths can be used
☑ Generator situated well away from water.	☑ Direct coupling of runner onto the generator shaft.
☑ Easy access to the generator.	☑ Ease of transportation when pre - fabricated in small sections.
<u>Disadvantages</u>	☑ Alignments are done more accurately in the workshop.
☒ A tank is required.	☑ Minimum civil works required.
☒ Civil works required for tank support.	<u>Disadvantages</u>
☒ Bearing alignment may need to be done on site.	☒ Careful design to avoid flow disturbance by the shaft.
☒ Long shafts may result in excessive vibrations.	☒ Seal requirement.
☒ Head fluctuation in the tank affects performance.	☒ Danger of generator failure if the seal does not perform effectively.
☒ Air might be drawn into the turbine if the tank capacity is too small.	

Table 3.1 : Advantages and disadvantages of open flume and tube type turbines.

The tube type layout was chosen mainly because it allows the direct coupling of the turbine runner onto the generator's shaft, makes use of the generator's bearings and requires less civil works. Important features of the proposed layout are described in section 3.3.3.

3.3.2 Configurations of tube type units

Following the decision for the propeltric unit to be based on the tube type design layout, three fundamental questions need to be addressed :

- ❑ Where to extract the power from, upstream or downstream ?
- ❑ What type of bearings to be used, greased or water lubricated ?
- ❑ Whether to use an overhung or central runner ?

For all of the following three choices there are eight possible configurations which are shown by *Figures 3.4 & 3.5*. The advantages and disadvantages of these possible layouts are given by *Table 3.2*. All information that is presented here, has been collected from field workers, local manufacturers and project managers[1].

3.3.2.1 Downstream or upstream power extraction ?

Propeller turbines can be designed so that the power can be extracted either *upstream* or *downstream* the runner.

Downstream power extraction has the advantage that the local pressure is much lower than on the *upstream side* and sealing is therefore much easier. With reference to Chapter 2, Soundranayagan [1988] and Viani [1990] have adopted the *downstream power extraction* in their designs.

Extracting the power *upstream* the turbine runner enables direct coupling to the generator shaft. However, a reliable sealing method is essential to ensure no water escapes through the seal and thus ensure failure free operation of the generator.

[1] Refer to Appendix B for information on field tips and site visits and Appendix D for useful contacts and addresses.

3.3.2.2 Are water lubricated bearings worth considering ?

The use of water lubricated bearings is attractive because of the simplicity in the way the turbine is constructed as well as the considerable savings in initial costs and the maintenance free requirements. For these reasons, water lubricated bearings have been used in micro hydro power generation units by many designers [Heitz, 1993; Holmen, 1991; Parker, 1993; Rao, 1987 and Viani, 1990]. Water lubricated bearings are made of a variety of materials, including : *vulcanized rubber*; *various plastics* (e.g. nylon or the modern graphite - impregnated polymers); or *metal* (especially bronze). The main disadvantages of water lubricated bearings are :

☒ Potentially short life.

☒ Relatively low efficiency due to high friction losses.

☒ High risk of damage from silt.

Susanto [1981] and Soundranayagam [1988] believe that it is actually more cost effective to install proper grease lubricated bearings and to use effective and reliable sealing methods for preventing generator failure and frequent maintenance requirements.

So far, reliability problems with water lubricated bearings [Mali, 1994 and Williams, 1995] suggest that further research is required to investigate the effects of vibrations and silty/abrasive environments on their reliability.

3.3.2.3 Is the use of an overhung runner worth considering ?

The turbine runner can either be mounted between two bearings, usually referred to as *centrally mounted runner* or can be *overhung* on the end of the shaft. A comparison between all the power extraction options is presented in *Table 3.2*.

Centrally mounted runners require very precise alignment between bearings but provide a very rigid support for the runner assembly. Grease bearings in water tight enclosures are commonly used for this arrangement. The process of mounting the bearings and the costs of the water tight enclosures are the major drawbacks of *centrally mounted runners* [Soundranayagam, 1988; Susanto, 1988 and Viani, 1992].

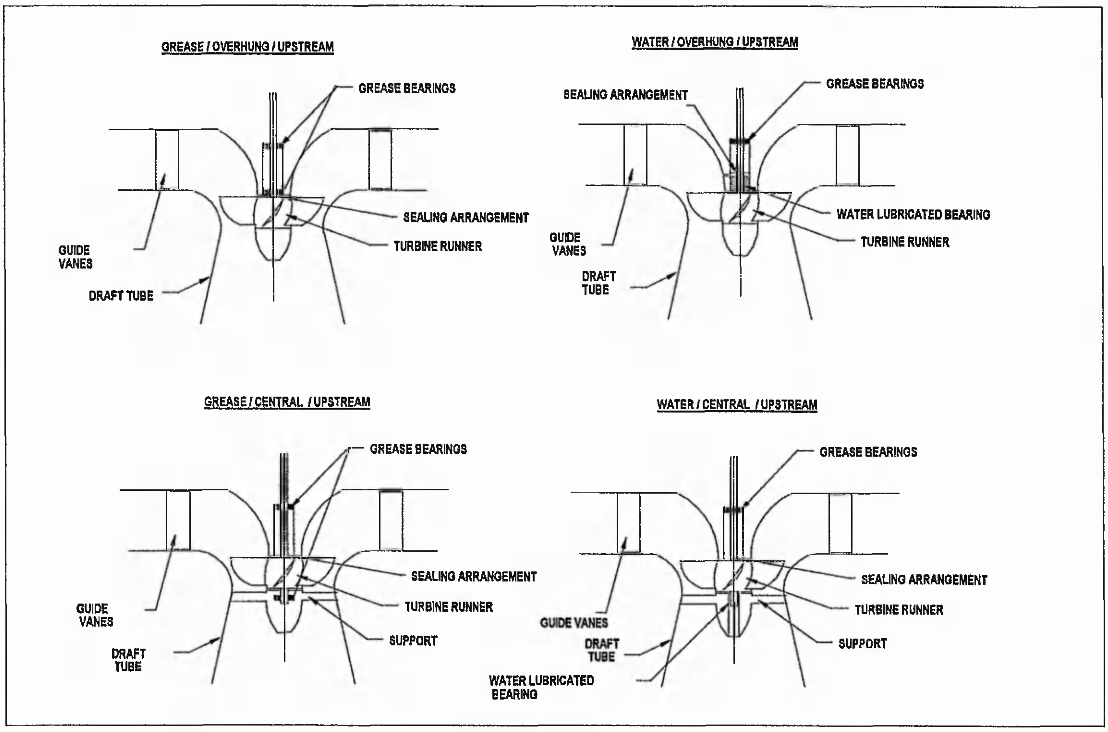


Figure 3.4 : Extracting the power upstream of the runner.

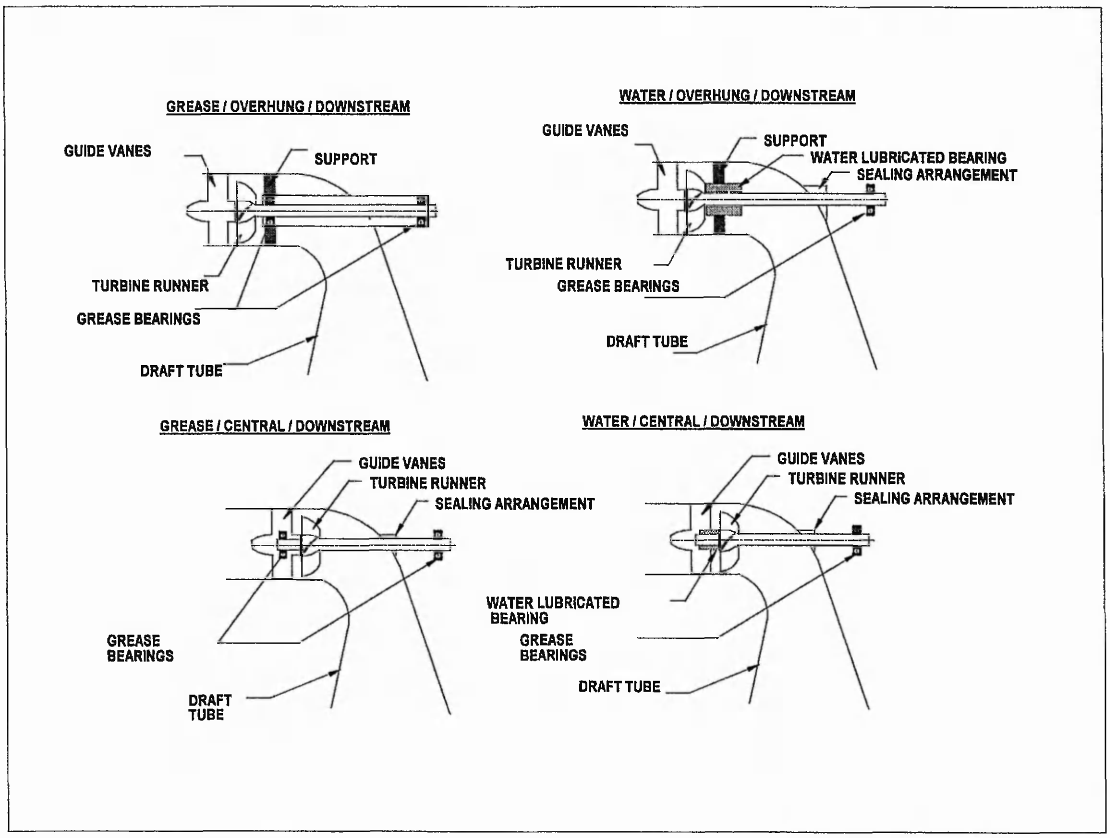


Figure 3.5 : Extracting the power downstream of the runner.

UPSTREAM POWER EXTRACTION	DOWNSTREAM POWER EXTRACTION
Grease lubricated bearings with overhung runner	**Grease lubricated bearings with overhung runner**
☑ Relatively easy to make. ☑ Good efficiency. ☑ Long life. ☑ Direct coupling to a generator shaft will ensure no extra bearings to be used. ☑ Easy to access and remove the runner.	☑ Good efficiency. ☑ Long life if materials are carefully chosen. ☒ Difficult to make. ☒ Reliability of bearings depends on the effectiveness of the seal. ☒ Difficulties in accessing the runner.
Grease lubricated bearings with central runner	**Grease lubricated bearings with central runner**
☑ Good efficiency. ☑ Long life. ☒ Difficult to make. ☒ Access to the runner is difficult.	☑ Common design for large turbines. ☑ Long life and good efficiency. ☒ Fairly easy to make. ☒ Difficulty in accessing the runner.
Water lubricated bearings with overhung runner	**Water lubricated bearings with overhung runner**
☑ Relatively common design for small machines, up to 5 kW. ☑ Easy to make. ☑ Easy access to the runner. ☒ Reliability problems in silty environments.	☒ Not used in any design described in Chapter 2. ☒ Very difficult to access the runner. ☒ Fairly difficult to make.
Water lubricated bearings with central runner	**Water lubricated bearings with central runner**
☒ Difficulties accessing the runner. ☒ Reliability problems of the bearing in silty environments.	☑ A common arrangement for small machines up to 5 kW. ☑ Easy to make.

Table 3.2 : Comparison of upstream and downstream power extraction configurations.

The major advantage of the *overhung* design is that the runner can be removed (e.g. for repairs) without the need for dismantling the entire shaft assembly. *Overhang runner arrangements* have been used by Holmen [1991], Rao [1988], Ranatunga [1991] and for the Vietnamese family units described by Green [1993a]. With the generator mounted in a vertical position, the generator bearings can be used to take the axial and radial loads from the turbine runner and therefore, no extra bearings are required as is the case when the generator is mounted horizontally. This arrangement is used by the Peltric system [Smith, 1992a] and has so far proved to be very reliable, cost effective and easy to make.

3.3.3 The proposed layout

The guide vanes may be arranged *axially* or *radially*. A simple approach, which is most feasible with the *radial* arrangement, is to incorporate the guide vanes into the concrete base of the power house or use an inlet spiral casing, as shown by *Figure 3.6*. The radial guide vanes are mounted onto a ring which is bolted into the centre of the casing. The use of such installation ensures minimum civil works requirements.

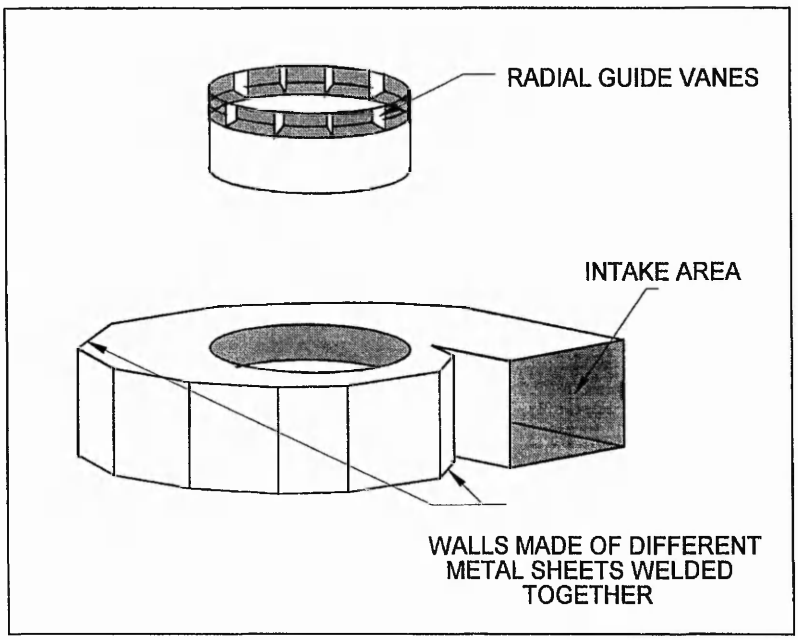


Figure 3.6 : The spiral casing.

The *tube type* layout with a spiral casing and *radial guide vane* arrangement is used for the development of the prototype unit[2] as shown by *Figure 3.7*. Power is extracted *upstream* of the *overhung* runner. The major advantage of the above configuration is the direct coupling of the propeller runner onto the generator's shaft, which requires a small extension (extension of generator shafts is widely done in Nepal for example by Katmandu Metal Industries[3]). The generator's bearings are used to take the axial and radial runner loads[4]. Less components, such as bearings and bearing housings are used, less workshop time is spent for mounting the generator - runner assembly and therefore the overall cost of the unit is reduced.

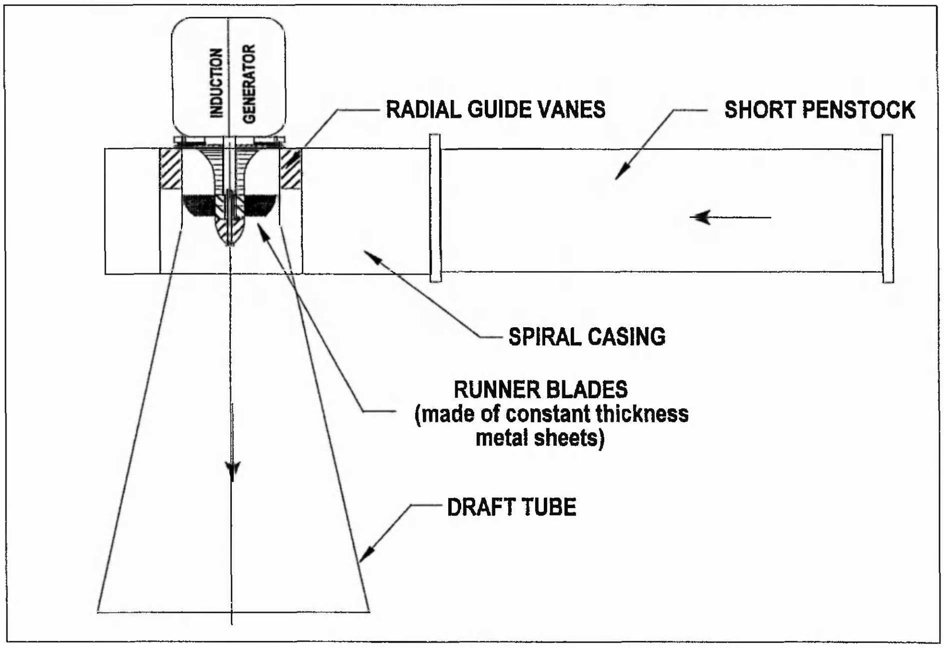


Figure 3.7 : Schematic of the proposed design layout for the prototype propeltric unit.

While the turbine was fabricated, every effort was made to use basic equipment and tools that are available in workshops in developing countries. The use of a lathe was limited to the turning of the turbine hub, runner blade tip diameter and the shaft extension. The unit is made in different sections that would allow ease of transportation and installation over rough terrain in rural areas. The assembly of the unit takes no more than two hours for one person using only an 10M spanner.

[2] Detailed design drawings are available in Appendix G.

[3] Katmandu Metal Industries P.V.T. ltd., Cha3-812 Nagal Quadon, Chhetrapati, Katmandu 3, Nepal.

[4] Refer to Appendix F for sample calculations.

3.4 Hydraulic design of the turbine runner

During the course of this section, the turbine runner design is presented and explained based upon the conclusions from the work presented in Chapter 2.

3.4.1 Turbine runner dimensions

The method described here is summarized by *Figure 3.8*. All calculations were performed with the aid of a spreadsheet, a printout of which is shown in section 3.4.3.

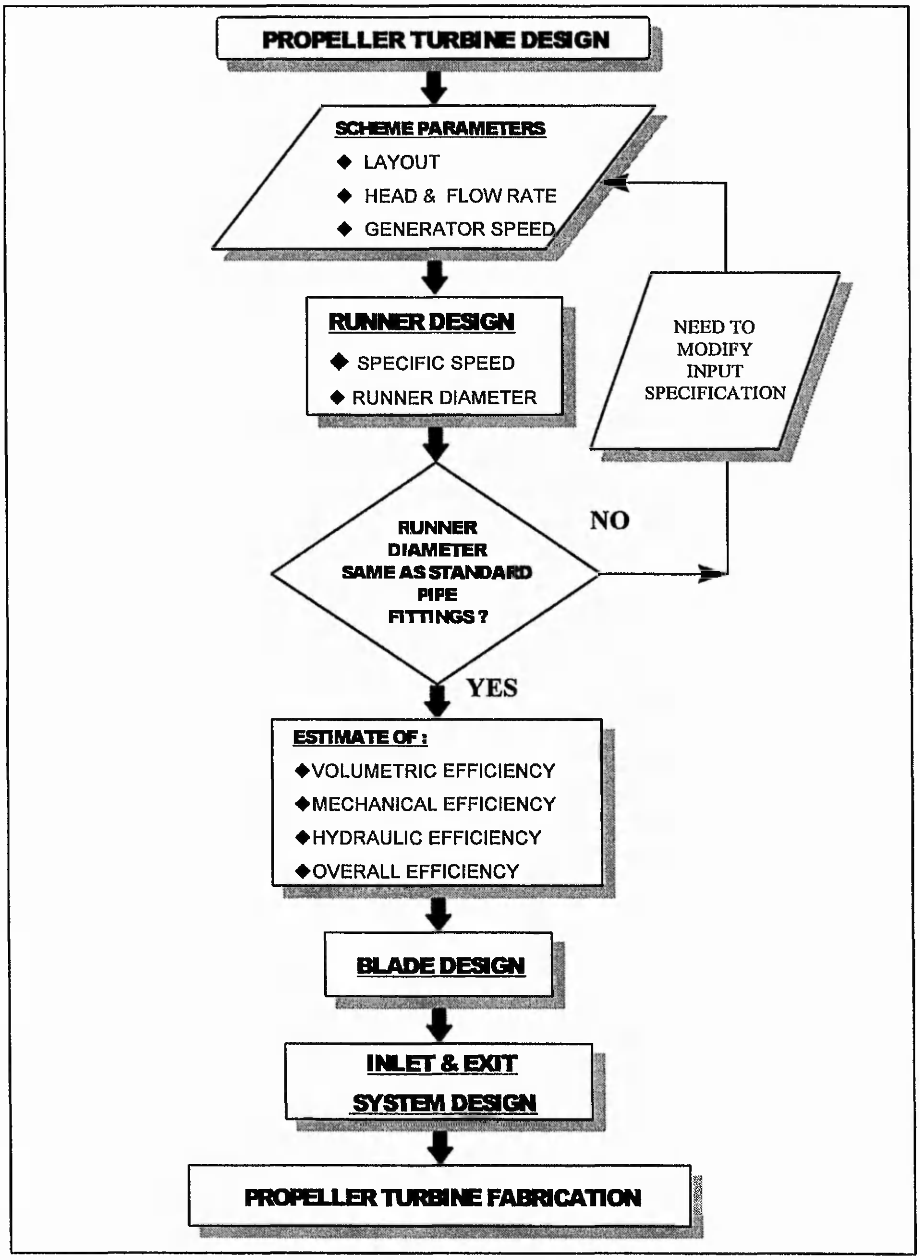


Figure 3.8 : Design methodology for the runner dimensions.

Axial flow water turbine runners are designed for specific speeds in the range of 90 to 270 $min.^{-1}$ as shown by *Figure 3.9*. This range is based on data and information available in literature for the design of propeller and Kaplan turbines by Bohl [1991] and Schweiger [1992].

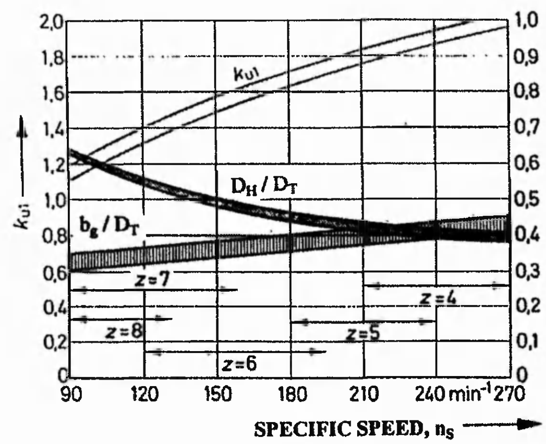


Figure 3.9 : Propeller turbine dimensions, [adopted from Bohl, 1991].

The use of *Figure 3.9* is very important as it enables the selection of the *number of runner blades*, the *hub to tip ratio* and the *tip velocity coefficient* using n_S. The physical dimensions of the runner are directly related to the specific speed of the turbine.

The design speed of the turbine is determined by the type of the generator used and the operating speed which it was designed for. For this particular prototype development, the characteristics of a 4 pole, 240V induction generator are used. According to Smith [1994b] a 4 pole induction generator can generate at speeds up to 15% of the generating speed, i.e. the generating speed range is from 1550rpm up to about 1770rpm. The range of generating speeds allows the designer to have some flexibility with regard to the turbine diameter and the use of standard dimensions. With an operating speed of 1560rpm, *net head* of 2m and a *flow rate* of 0.072m^3/s, the *dimensional specific speed* of the turbine is given by

$$n_s = \frac{N\sqrt{Q}}{H_N^{\ 3/4}} = 248.89\ min^{-1} \qquad \{3.2\}$$

With reference to *Figure 3.9*, four runner blades would be used for this design with a *hub to tip ratio* of 0.4. The tip peripheral velocity of the runner is related to the theoretical spouting velocity [Warnick, 1984] $\sqrt{2gH_N}$ and is given by

$$u_T = k_{u1}\sqrt{2\,g\,H_N} = 12.216\ m/s \qquad \{3.3\}$$

The value of k_{u1} was determined by plotting both the upper and lower limits of k_{u1} against the specific speed data from *Figure 3.9* as shown by *Figure 3.10*. With the aid of curve fitting, the value of k_{u1} is given by the following function :

$$k_{u1} = -5.03714\,\mathrm{E}^{-6}\,n_s^2 + 0.00647024\,n_s + 0.651786 = 1.95 \qquad \{3.4\}$$

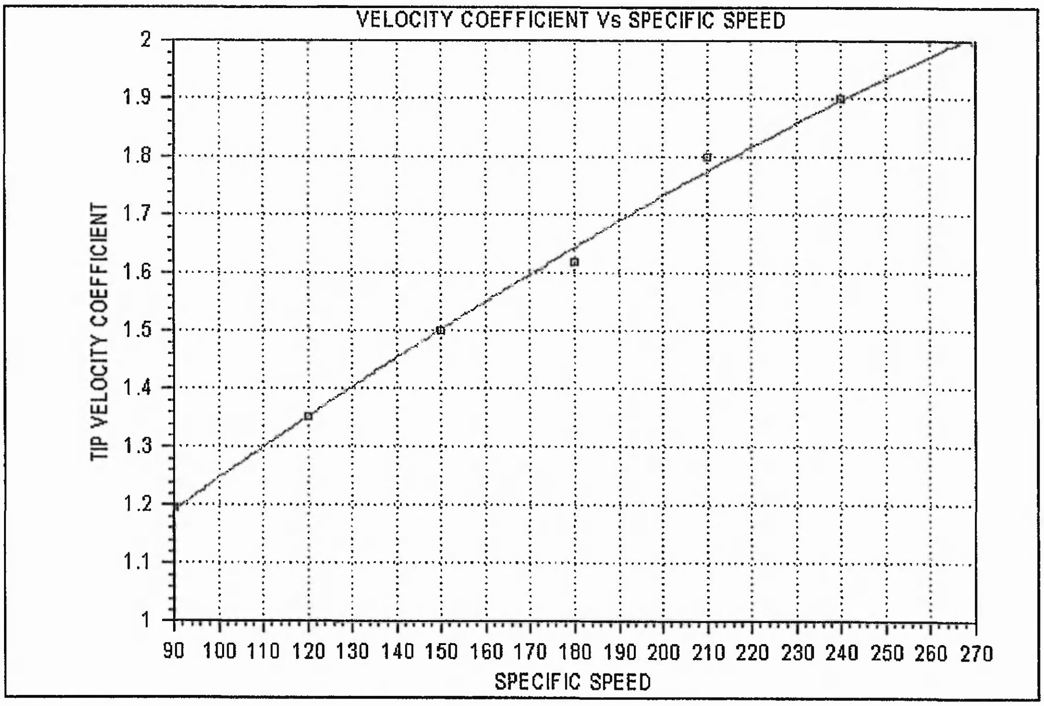


Figure 3.10 : The function evaluation curve for calculating k_{u1}.

The peripheral velocity of the runner may also be expressed as

$$u = \frac{\pi D N}{60} = \omega r \qquad \{3.5\}$$

Combining equations {3.3} and {3.5}, the tip diameter of the turbine runner becomes

$$D_T = \frac{84.6\,k_{u1}\sqrt{H_N}}{N} = \frac{84.6\ \times 1.95\ \times \sqrt{2}}{1560} = 0.15\ m \qquad \{3.6\}$$

A pipe size of 6" internal diameter[5] would be used to fabricate the runner enclosure. The calculation for D_T is an iterative process whose value would have to be approximated to the nearest pipe size. Part of the *flow rate* and/or *net head* available would therefore be sacrificed. However, the advantages of using the standard pipe sizes would result in reduction of manufacturing and material costs.

The effect that high operating speeds have on the overall turbine runner dimensions can be clearly demonstrated by repeating the same calculation using a 6 pole induction generator. The values of n_s and D_T become 166min^{-1} and 0.187m respectively, i.e. 25% increase in diameter, which result in probably 50% increase in material.

The use of such high *specific* and *turbine speeds* has been attempted only by Heitz [1993], where the design speed of the turbine was 2100rpm. The rest of the designs described in section 2.3 of Chapter 2 have followed the more traditional way of using relatively low specific speeds ranging from 125 to 190min^{-1} and driving the generator with belt drive systems without considering the use of standard dimensions.

3.4.2 Estimating the turbine efficiency

During the process of energy conversion, losses occur in the various parts of water turbines. How good a water turbine is in converting the available hydraulic energy into mechanical energy is indicated by the *turbine overall efficiency* given by

$$\eta_o = \frac{\text{Shaft Power output from the turbine}}{\text{Fluid input power through the machine}} = \frac{T\,\omega}{\rho\, g\, H_N\, Q} \qquad \{3.7\}$$

During the course of this research project, three very important efficiency terms which the *overall efficiency* of the turbine can be divided into will be considered. These are :

- ❐ *Volumetric efficiency.*
- ❐ *Mechanical efficiency.*
- ❐ *Hydraulic efficiency.*

5 Imperial units for pipe sizing are still widely used in developing countries especially those of South East Asia.

Due to the lack of information available in literature and the reluctance of turbine manufacturers to provide details of their design methods, values of these efficiencies are crude estimates and are used for the preliminary design of the prototype turbine. The method used for estimating these losses is based on empirical formulae and assumptions which have been used over the years mainly for estimating the efficiencies of machines from model tests.

At this point, it is important to state that unless a machine is tested, the exact value of these losses cannot be determined. A more detailed analysis of these efficiencies and the parameters that are related to are given in Chapter 4.

3.4.2.1 Volumetric efficiency considerations

The *volumetric efficiency* refers to the amount of water flow that is not utilized by the turbine runner due to *leakage losses. Leakage losses* occur due to the gap between the turbine blade tip section and the turbine casing wall as well as through seals. Since not all the water is utilized during the energy transfer process, the *volumetric efficiency* of a turbine is defined as :

$$\eta_V = \frac{\text{Total flow} - \text{Leakage flow}}{\text{Total flow}} = \frac{Q - Q_L}{Q} \qquad \{3.8\}$$

A very useful empirical formula has been derived by Kovalev [1963] and Mosonyi [1957] based on the same method that Stepanof [1957] has introduced for leakage losses through small gaps given by

$$\eta_V = \frac{100 \ - \ (\text{percentage leakage loss})}{100} \qquad \{3.9\}$$

where the *percentage leakage losses* are given by

$$\upsilon_L = 12.5\frac{\sqrt{H_N}}{Q}\, D_T \, s \, 100 \qquad \{3.10\}$$

The effect of the gap between the turbine runner tip section and the turbine casing wall on the *volumetric efficiency* is very important for small turbines and can be demonstrated by the example shown in *Table 3.3*.

Turbine	H_N	Q	D_T	s (mm)	η_V (%)
Propeltric	2	0.072	0.15	1	96.32
unit	2	0.072	0.15	1.5	94.48
Kaplan turbine	2	5	1.2	1	99.58
Bohl [1991]	2	5	1.2	1.5	99.36

Table 3.3 : Volumetric efficiency comparison between small and large axial flow turbines due to the gap between runner blades and turbine casing.

For large machines with power output ranging from a few to hundreds of MW, *leakage losses* are very small and efficiencies can be as high as 99.9%. As it is clearly shown by *Table 3.3*, the value of volumetric efficiency of large machines does not deteriorate, as it is the case with the small size propeller turbines, even when these are manufactured by the same standards in developing countries. For the design of the propeltric unit it was assumed that *leakage losses* would occur only through the turbine tip section and there would be no losses through the non contact seal, whose performance is evaluated in Chapter 4. The design *volumetric efficiency* for the prototype propeller turbine is 94.5%.

3.4.2.2 Mechanical efficiency

A certain amount of shaft power is used to overcome the various *mechanical losses* in a hydraulic turbine, which are associated with *disc friction losses*, *bearing losses*, and *friction losses* due to the presence of seals. The *mechanical efficiency* for a water turbine is given by

$$\eta_M = \frac{T\,\omega}{T\,\omega + \sum mechanical\ losses} \qquad \{3.11\}$$

Disc friction losses are experienced when a disc rotates in a hydraulic medium, [Sadek, 1960 and Salami, 1970], and are related to the *Reynolds number* and the surface area of the disc. *Disc friction losses* are very important for the analysis of Francis turbines and centrifugal pumps. As it is clearly stated by Mosonyi [1957] and Raabe [1985], *disk friction losses* are not considered for the design of propeller turbines.

The *bearing losses*[6] for the prototype propeltric unit are 3.26W. *Mechanical losses* would also include the power consumed by the non contact seal, described in section 3.8. The *mechanical efficiency* of the prototype turbine is estimated to be equal to 98%.

3.4.2.3 Hydraulic efficiency

The value of *hydraulic efficiency* of a water turbine is very important as it represents the effectiveness with which the energy is transferred from the fluid to the runner and is equal to

$$\eta_H = \frac{H_N - \sum\{\text{hydraulic head loss in the turbine}\}}{H_N} = \frac{H_E}{H_N} \qquad \{3.12\}$$

Hydraulic head losses occur in various parts of a water turbine. These include losses in the *spiral casing*, *guide vanes*, *runner* and the *draft tube*. The understanding and prediction of the flow behaviour in a turbine unit is crucial for cost and performance optimisation. Turbine manufacturers spend a large amount of time on improving the hydraulic efficiency of their designs. The use of complicated mathematical models, three dimensional fluid analysis and flow visualisation methods are among the methods usually employed.

The nature and the design of micro hydro turbines requires a different approach. Though the efficiency of the turbine is important, costs, ease of manufacturing and reliability are important too. Sometimes turbine efficiency is sacrificed to a certain degree in order to provide a solution that would be affordable by the local communities in developing countries. Unless a turbine is tested, the *hydraulic efficiency* can only be estimated. Since turbine manufacturers rarely give away details of their designs, methods and techniques available in literature can be used for a preliminary design. In many design textbooks assumptions are made for the *hydraulic efficiency* based on data that have collected from turbine and pumps manufacturers. For example, Lazarkiewicz [1965] suggests to use a *hydraulic efficiency* of 83% for an axial flow pump design, but later on explains that the *hydraulic efficiency* of a pump can be in the range of 65% to 96% depending on the specific speed of the pump, the size and the quality of manufacture.

[6] Calculations available in Appendix F.

Test results of the propeller turbine designs described in section 2.3 of Chapter 2 indicate that the design *hydraulic efficiencies* used for designing these turbines were higher than what the actual test results have shown. Heitz [1993] has used a hydraulic efficiency of 90% based on the method described by Bohl [1992]. However, the test results presented by Mali [1994] and Williams [1995] have shown a very poor performance of the turbine with the maximum efficiency just over 24%. Durali [1976] has used a method originally developed for axial flow gas turbines by Craig [1971] with the same effects. Rao [1986] on the other hand, has assumed a value of 60% for the *overall efficiency* of the propeller turbine and then predicted a value of 67% for the *hydraulic efficiency*. Soundranayagam [1988] and Viani [1990] have assumed a value of 80% and 75% respectively.

The examples described above suggest that a more accurate and reliable way has to be found for the preliminary design of the propeller turbine. For the purposes of this research project, it is initially assumed that *hydraulic losses* would occur in the runner and draft tube sections only, while the losses in the spiral casing and the guide vanes are considered to be negligible. The analysis is based on the work of Hutton [1954], Mosonyi [1957] and Nechleba [1957]. The losses in the runner are assumed to be 2/3 of those of the runner and draft tube combined. This is based on the aerodynamic method described by Hutton [1954] and Raabe [1983]. The draft tube losses, which are defined in section 3.7 of this Chapter, are given by

$$\xi_{dt} = \frac{\left[1 - \eta_{dt}\right] V_{m2}^2 - V_{m3}^2}{2g} = 0.1656\ m \qquad \{3.13\}$$

It is important to state here that the method would be valid only if there is no exit whirl velocity from the turbine runner as explained in the following section. This approach is very simple and can be easily used by field workers in developing countries who have a basic knowledge of fluid machines and hydraulic theory.

The *turbine hydraulic loses* are therefore estimated to be 0.497m which results to a *hydraulic efficiency* value of 75.2%. The *turbine overall efficiency* may now be expressed as :

$$\eta_o = \eta_H \times \eta_V \times \eta_M = 0.696 \qquad \{3.14\}$$

3.4.3 Turbine velocity diagrams

In order to minimize the velocity head loss from the draft tube exit, the design of the turbine is based on a zero whirl component of the fluid at the runner exit. The Euler equation for a propeller turbine becomes

$$gH_E = u\left[V_1 \cos\alpha_1\right] = uV_{u1} = 14.754\ m^2 / s^2 \qquad \{3.15\}$$

Figure 3.11 shows part of the spreadsheet[7]. used for the design calculations of the propeltric unit.

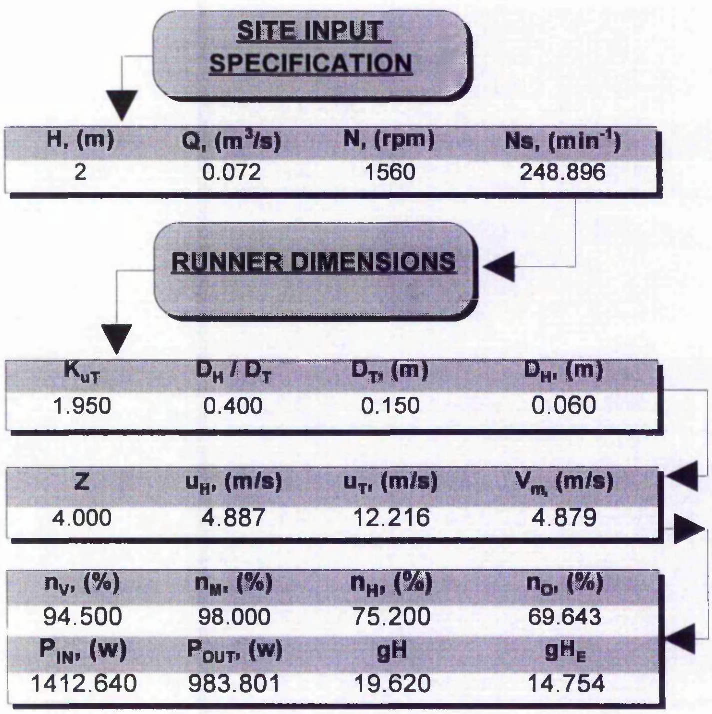


Figure 3.11 : Design calculations for turbine runner dimensions and turbine efficiencies.

For an axial flow machine, the velocity of the runner blades is given by

$$u = u_1 = u_2 = \omega\, r \qquad \{3.16\}$$

The *meriodional* or *axial flow velocity*, which is parallel to the turbine shaft, is constant at the inlet and outlet sections of the runner and is given by :

[7] Details are found in Appendix F.

$$V_e = V_{a1} = V_{a2} = \frac{Q}{\frac{\pi}{4}\left[D_T^2 - D_H^2\right]} = 4.879\ m\ /\ s \qquad \{3.17\}$$

Having determined the velocity components described above, the velocity diagrams for the prototype propeller turbine can be drawn as shown in *Figure 3.12*. The design calculations for the *tip*, *hub* and *mid sections* of the turbine blades are presented by *Figure 3.13*, *Figure 3.14* and *Figure 3.15* respectively.

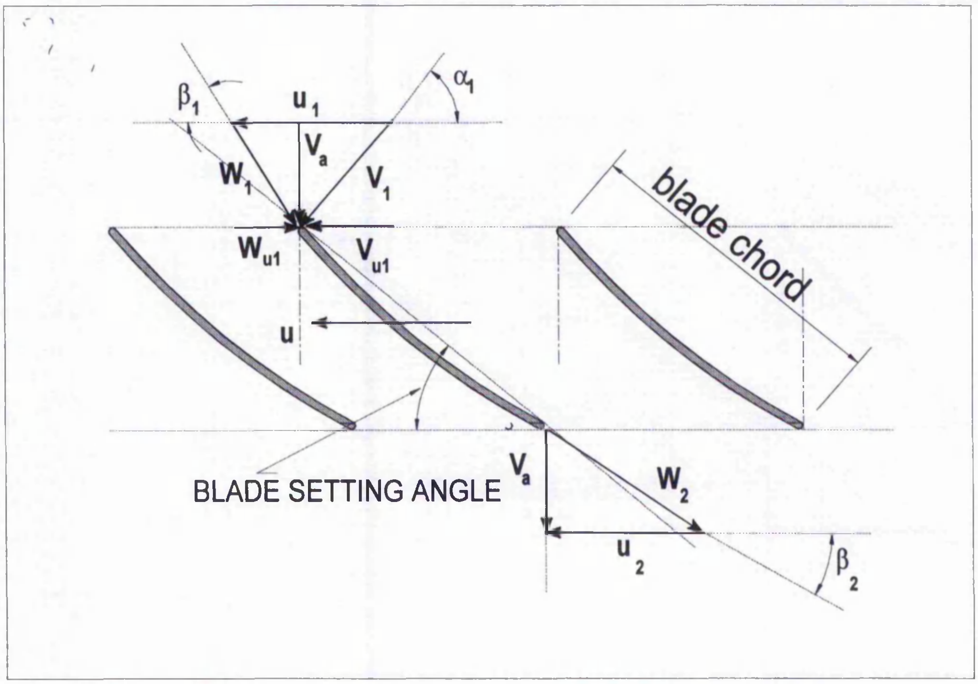


Figure 3.12 : Definitions used for velocity diagrams in the turbine cascade at inlet and exit blade sections.

D_T, (m)	V_1, (m/s)	α_1, (deg.)	W_{u1}, (m/s)
0.150	5.026	76.095	11.008
u_T, (m/s)	V_2, (m/s)	α_2, (deg.)	W_{u2}, (m/s)
12.216	4.879	68.230	12.216
V_{uT1}, (m/s)	W_1, (m/s)	β_1, (deg.)	ΔW_u, (m/s)
1.208	12.041	23.902	1.208
V_{uT2}, (m/s)	W_2, (m/s)	β_2, (deg.)	$\beta_1 - \beta_2$, (deg.)
0.000	11.200	21.770	2.132
V_m, (m/s)	W_{oo}, (m/s)	β_{oo}, (deg.)	t_T, (m)
4.879	12.595	22.789	0.117

Figure 3.13 : Velocity calculations for tip section.

D_H, (m)	V_1, (m/s)	α_1, (deg.)	W_{u1}, (m/s)
0.060	5.737	58.248	1.867
u_H, (m/s)	V_2, (m/s)	α_2, (deg.)	W_{u2}, (m/s)
4.887	4.879	45.047	4.887
V_{uH1}, (m/s)	W_1, (m/s)	β_1, (deg.)	ΔW_u, (m/s)
3.019	5.224	69.055	3.019
V_{uH2}, (m/s)	W_2, (m/s)	β_2, (deg.)	$\beta_1-\beta_2$, (deg.)
0.000	6.905	44.953	24.102
V_{in}, (m/s)	W_{oo}, (m/s)	β_{oo}, (deg.)	t_H, (m)
4.879	5.933	55.309	0.047

Figure 3.14 : Velocity calculations for hub section.

D_m, (m)	V_1, (m/s)	α_1, (deg.)	W_{u1}, (m/s)
0.105	5.175	70.524	6.826
u_m, (m/s)	V_2, (m/s)	α_2, (deg.)	W_{u2}, (m/s)
8.552	4.879	60.296	8.552
V_{um1}, (m/s)	W_1, (m/s)	β_1, (deg.)	ΔW_u, (m/s)
1.725	8.391	35.553	1.725
V_{um2}, (m/s)	W_2, (m/s)	β_2, (deg.)	$\beta_1-\beta_2$, (deg.)
0.000	9.845	29.704	5.848
V_m, (m/s)	W_{oo}, (m/s)	β_{oo}, (deg.)	t_m (m)
4.879	9.106	32.395	0.082

Figure 3.15 : Velocity calculations for mid section.

The relative and absolute velocity flow angles are later on used for the design of the runner blades, section 3.4.4. Looking at these calculations, it is obvious that the flow is experiencing more deflection towards the hub section in comparison to the deflection experienced by the middle and tip blade sections. The reason for such an increased amount of twist between the hub and tip sections of the blade is because of the very high operating speed of the turbine, a design parameter that is determined by the generator speed and the value of the hub to tip ratio which was determined from *Figure 3.9* .This observation is of great value because the designer can decide at this stage about the amount of twist the blade would have to be given for a particular design. As it is described later on in this Chapter, section 3.4.5, the amount of twist is an important parameter for blade fabrication using metal sheets of constant thickness. Solutions to this problem are briefly discussed in section 3.4.5 whereas complete design solutions are presented in Chapter 4.

3.4.4 Blade design

Conventional axial flow turbine runners have aerofoil shape blades using well known profiles such as *NACA*, *Y Clark* and *Göttingen*. The characteristics of these profiles, such as their *lift* and *drag* coefficients, can easily be obtained from textbooks [Abbott, 1959; Houghton, 1982 and Riegels, 1961]. However, due to the geometrical complexity of these shapes, specific milling operations and skills are required for their production that are not commonly available in small workshops in developing countries. And even if the skills and machinery are available, turbine runner blades made of aerofoil shaped profiles would cost a lot more compared with fabricated blades made of flat sheets of metal, bent and twisted to the desired angles.

The design approach described in this section is adopting the same principles used in aerofoil theory for a *cascade of blades* made of constant thickness sheets of metal. Experiments conducted on both blades made of constant thickness sheets of metal and aerofoil sections, described by Eckert [1944], showed that the same efficiency has been attained by both shapes, provided that the geometry is preserved (the results from these tests are valid when the condition $Re \geq 3.25 \times 10^5$ is met, i.e. turbulent flow regime). The value for Re for the tip and hub section of the runner blades is 16×10^5 and 3.47×10^5 respectively and is determined using the following equation :

$$R_e = \frac{W_\infty \, c}{\nu} \qquad \{3.18\}$$

Figure 3.16 shows the steps followed for the design of the runner blades. The value of the *lift coefficient* can be as low as 0.2 at the *tip section* of the blade whereas at the *hub section* it might well be equal to 1. In order to establish the most convenient blade shape for fabrication, calculations were performed for *solidity ratio* values ranging between 0.6 and 1.2 following the suggestions made by Sigloch [1993].

The *blade spacing* is given by

$$t = \frac{\pi D}{Z} \qquad \{3.19\}$$

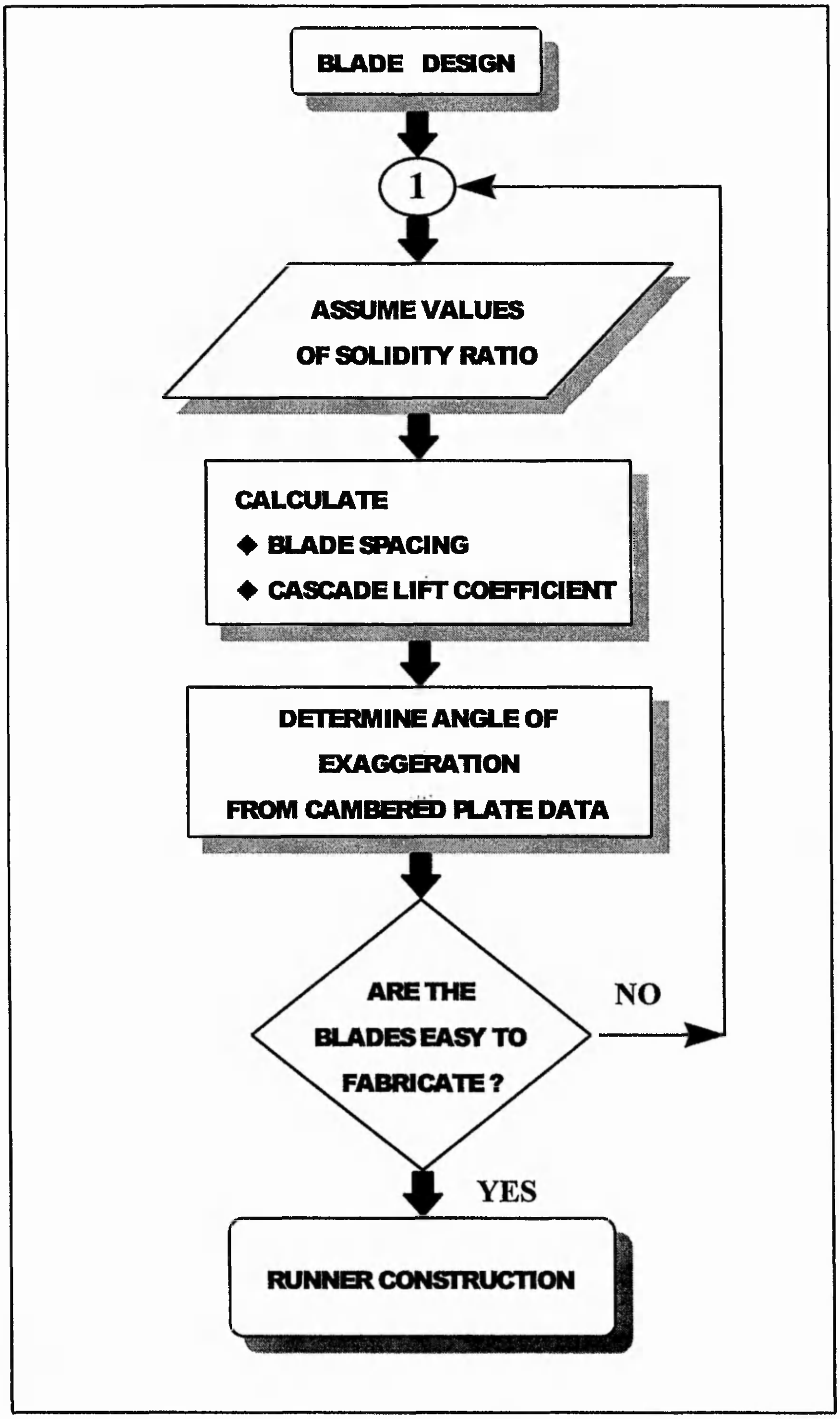


Figure 3.16 : Blade design flow chart.

3.4.4.1 Lift coefficient for the blade cascade

For the design of the turbine blades, *Figure 3.17* is used in order to establish a relationship between the *cascade lift coefficient*, *blade chord length* and *blade spacing*.

Details of the method described here are available in the published work of Shepherd [1965] and Turton [1984]. With reference to *Figure 3.17*, the *lift force* acts in a direction perpendicular to W_∞ and is given by :

$$L = \frac{1}{2} \rho \, C_L \, W_\infty^2 \text{ [blade area]} \tag{3.20}$$

The *drag force* acts in the same direction as W_∞ and is given by :

$$D = \frac{1}{2} \rho \, C_D \, W_\infty^2 \text{ [blade area]} \tag{3.21}$$

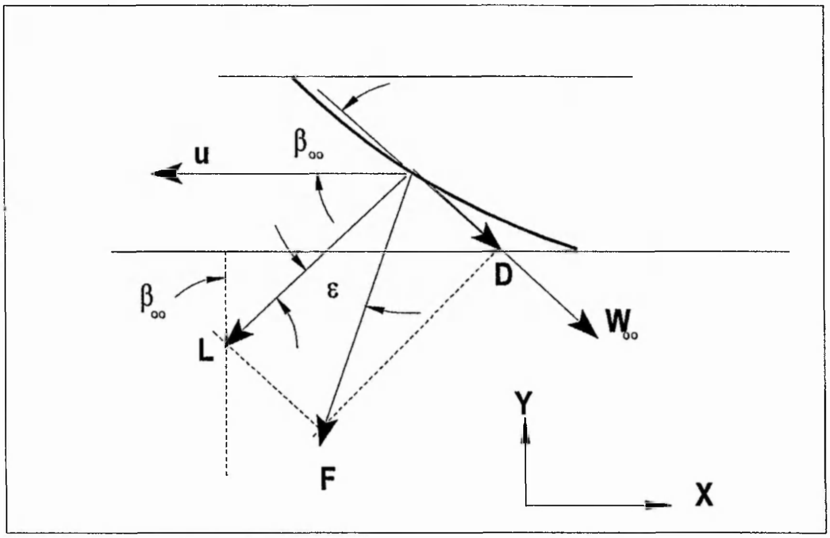


Figure 3.17 : Force analysis of a blade in a cascade.

The *blade area* is found by multiplying the *chord length* with the *unit length* of the blade [Csanady 1964]. Both *lift* and *drag* forces consist of *axial* and *tangential* components. The resultant force in the tangential direction is given by

$$F_X = L_X - D_X = L\sin\beta_\infty - D\cos\beta_\infty \tag{3.22}$$

From the velocity diagrams, given that $W_\infty = \dfrac{V_a}{\sin\beta_\infty}$, equation {3.22} becomes

$$F_X = \frac{1}{2} \rho \, \frac{V_a^2}{\sin^2 \beta_\infty} \text{[Area of blade]} \, C_L \left[\sin\beta_\infty - \frac{C_D}{C_L} \times \cos\beta_\infty \right] \tag{3.23}$$

The primary purpose of an aerofoil is to produce *lift* when placed in fluid stream. However, it is well known that a body in a fluid stream will experience *drag* as well. A measure however of how good an aerofoil is performing is the *lift* to *drag ratio*. The higher the ratio, the better the performance of the aerofoil. Information published by Wallis [1961] for flat plates and cambered constant thickness plates, presented by *Table 3.4*, suggest that for a small *angle of attack*[8] (otherwise referred to as *angle of incidence* or *approach*), i.e. up to about 8°, the value of the *drag coefficient* is very small.

Profile type	Flat	Flat	4% Camber	4% Camber
Angle of attack	4°	8°	4°	8°
C_L	0.3	0.76	0.78	1.0
C_L / C_D	12	8	38	22
C_D	0.025	0.095	0.0205	0.0445

Table 3.4 : Data on flat and cambered plates, [adopted from Wallis, 1961].

It is therefore evident that the value of C_D is very small and for the purposes of this design it will be assumed to have a very small value. Equation {3.23} becomes

$$F_X = \frac{1}{2}\,\rho\,\frac{V_a^2}{\sin^2\beta_\infty}\,c\,\left[C_L\cos\beta_\infty\right] \qquad \{3.24\}$$

Since the main purpose of a *cascade* is to deflect the flow though the turbine, there is always a change in momentum across the cascade and a force associated with it. Applying the momentum equation in the direction of the relative velocity,

$$F = \overset{o}{m}\,[\Delta W_u] = \overset{o}{m}\,\left[W_{u1} - W_{u2}\right] \qquad \{3.25\}$$

The unit mass flow rate through the turbine is given by $\overset{o}{m} = \rho\ V_a\ (area)$ where $area = t \times unit\ depth\ of\ blade$.

[8] Small angles of attack are used during the design of hydraulic machines so that the blade drag coefficient to be small and to avoid running the blades near stall conditions.

As $W_{u1} = \dfrac{V_a}{\tan \beta_1}$ and $W_{u2} = \dfrac{V_a}{\tan \beta_2}$, substituting in equation {3.25}

$$F_x = \rho\, V_a\, t \left[\frac{V_a}{\tan \beta_1} - \frac{V_a}{\tan \beta_2}\right] = \rho\, V_a^2\, t \left[\cot \beta_1 - \cot \beta_2\right] \qquad \{3.26\}$$

Combining {3.24} and {3.26} the *cascade lift coefficient* becomes

$$C_L = 2\, \frac{t}{c} \left[\cot \beta_1 - \cot \beta_2\right] \sin \beta_\infty \qquad \{3.27\}$$

But, $\cot \beta_1 = \dfrac{W_{u1}}{V_a}$, $\cot \beta_2 = \dfrac{W_{u2}}{V_a}$ and $\sin \beta_\infty = \dfrac{V_a}{W_\infty}$. Therefore equation {3.27} becomes :

$$C_L = 2\, \frac{t}{c}\, \frac{\Delta W_u}{W_\infty} \qquad \{3.28\}$$

Equation {3.28} assumes frictionless flow. The value of the difference between the tangential components of the relative velocity is given by

$$\Delta W_u = \left[W_{u1} - W_{u2}\right] \qquad \{3.29\}$$

3.4.4.2 Blade geometry

The profile used for fabrication simplicity is the circular arc. The blade is treated as a skeleton line which is known as the *camber line* of the blade and has a radius given by

$$R = \frac{c}{2 \sin\left(\frac{\theta}{2}\right)} \qquad \{3.30\}$$

Being a circular arc, the *maximum camber* of the blade is located at 50% of *chord length* and is equal to

$$f \approx \frac{c^2}{8\,R} \qquad \{3.31\}$$

The geometry of a circular arc blade profile and the associated parameters are shown by *Figure 3.18.*

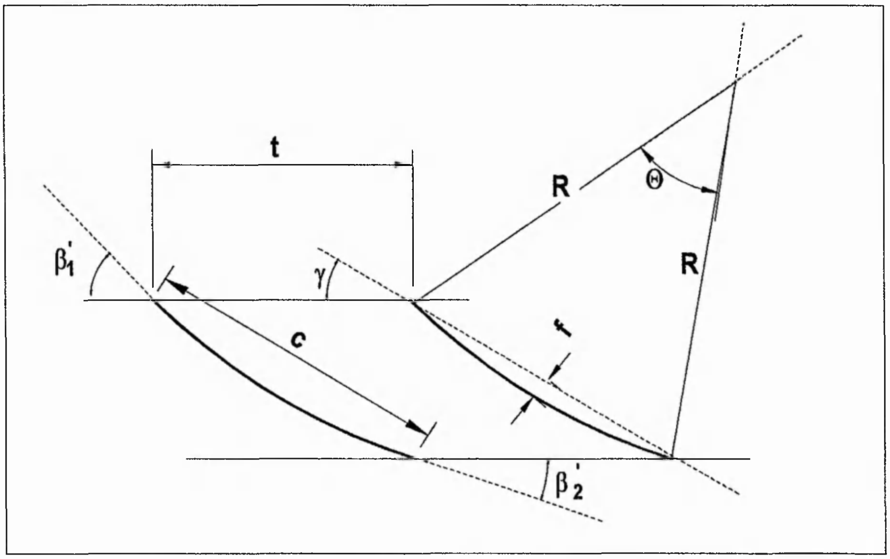


Figure 3.18 : Blade geometry of circular arc profile.

The circular arc is first constructed so that the inlet and outlet flow velocities are tangential to the arc, demonstrated by *Figure 3.12.* In this case, θ can be determined directly from the inlet and outlet flow angles as follows

$$\theta = \beta_1 - \beta_2 \qquad \{3.32\}$$

However, research by Wienig [1935] suggests a correction to be applied by the so called *angle of exaggeration* or the *Wienig factor* shown by *Figure 3.19.*

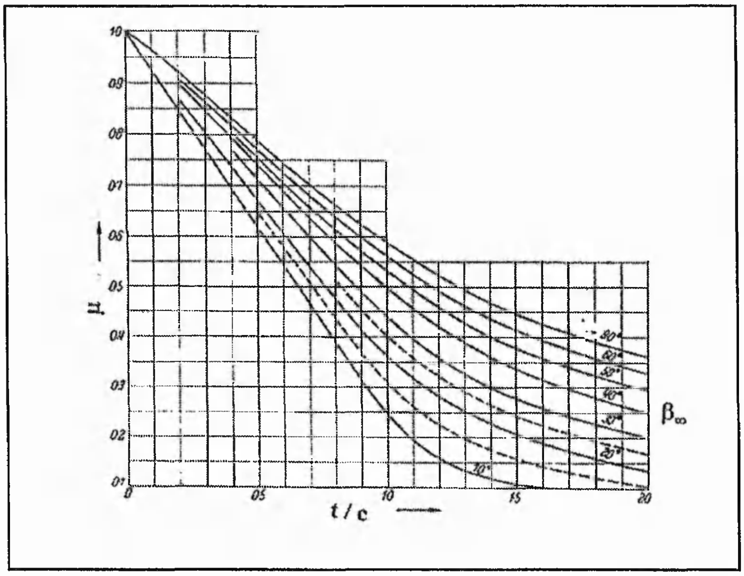


Figure 3.19 : Wienig factor for circular arcs [adopted from Eck, 1973].

The *blade camber angle* and the *flow deviation angle* are not the same and the velocity vectors at inlet and outlet sections are no longer tangential to the skeleton line of the profile. The *blade setting angle*, which is taken in the same direction as β_∞, and the *solidity ratio* are related to the values of the *angle of exaggeration* as follows

$$\mu = \frac{\text{flow deviation}}{\text{blade camber}} = \frac{\beta_1 - \beta_2}{\beta_1' - \beta_2'} \qquad \{3.33\}$$

Due to the fact the blades are of finite thickness, the influence of the profile usually needs to be taken into account according to the work presented by Bohl [1991] and Pfleiderer [1986]. The thickness of the blade, which was designed using a 3mm sheet metal plate[9], would have an effect on the cross section of the flow channel leading to a change in the flow around the blade and therefore the flow deviation angle, which in the case of water turbines would be increased. The blade setting angle would therefore be altered to correct for this effect as shown below

$$\beta_\infty^* = \beta_\infty + \Delta\beta_\infty \qquad \{3.34\}$$

The correction value of $\Delta\beta_\infty$ can be found from the following expression

$$\Delta\beta_\infty = \Delta\beta_{\infty 1}\left[\frac{c}{t}\right]^2 \qquad \{3.35\}$$

Details given by Bohl [1991] suggest the use of $\Delta\beta_{\infty 1} = 0.3^\circ$. The correction blade setting angle for the maximum solidity ratio that has been considered is

$$\Delta\beta_\infty = 0.3\left[\frac{c}{t}\right]^2 = 0.3\,[1.2]^2 \approx 0.43^\circ \qquad \{3.36\}$$

The correction angle calculation is an iterative process and is repeated until the corrected values do not change. However, the accuracy of a fraction of a degree can not be of any benefit since the fabrication method used is not as accurate. For the purposes of this research project, the iteration method was not used.

Table 3.5 and *Table 3.6* show the geometrical description of the runner blades for the hub and tip sections respectively.

[9] Calculations available in Appendix F.

t/c	0.800	0.900	1.000	1.200	1.300	1.400	1.500	1.600
c, (m)	0.059	0.052	0.047	0.039	0.036	0.034	0.031	0.029
C_L	0.812	0.913	1.015	1.218	1.319	1.420	1.522	1.623
μ	0.640	0.580	0.550	0.480	0.450	0.420	0.380	0.360
θ, $(\beta'_2-\beta'_1)$, (deg.)	37.530	41.413	43.672	50.041	53.377	57.189	63.209	66.721
R	0.091	0.074	0.063	0.046	0.040	0.035	0.030	0.027
f	0.005	0.005	0.004	0.004	0.004	0.004	0.004	0.004
f/c	0.080	0.088	0.093	0.106	0.112	0.120	0.131	0.137

Table 3.5 : Geometrical definition of the propeller turbine for hub section.

t/c	0.800	0.900	1.000	1.200	1.300	1.400	1.500	1.600
c, (m)	0.147	0.131	0.117	0.098	0.090	0.084	0.078	0.073
C_L	0.153	0.172	0.191	0.229	0.249	0.268	0.287	0.306
μ	0.500	0.450	0.400	0.325	0.280	0.255	0.225	0.215
θ, $(\beta'_2-\beta'_1)$, (deg.)	4.252	4.724	5.314	6.541	7.592	8.336	9.448	9.887
R	1.979	1.584	1.267	0.858	0.682	0.577	0.475	0.426
f	0.001	0.001	0.001	0.001	0.001	0.002	0.002	0.002
f/c	0.009	0.010	0.012	0.014	0.017	0.018	0.021	0.022

Table 3.6 : Geometrical definition of the propeller turbine for tip section.

Figure 3.20 shows details for the *blades setting angles* and *blade camber* at four different sections of the runner blades obtained by the design calculations.

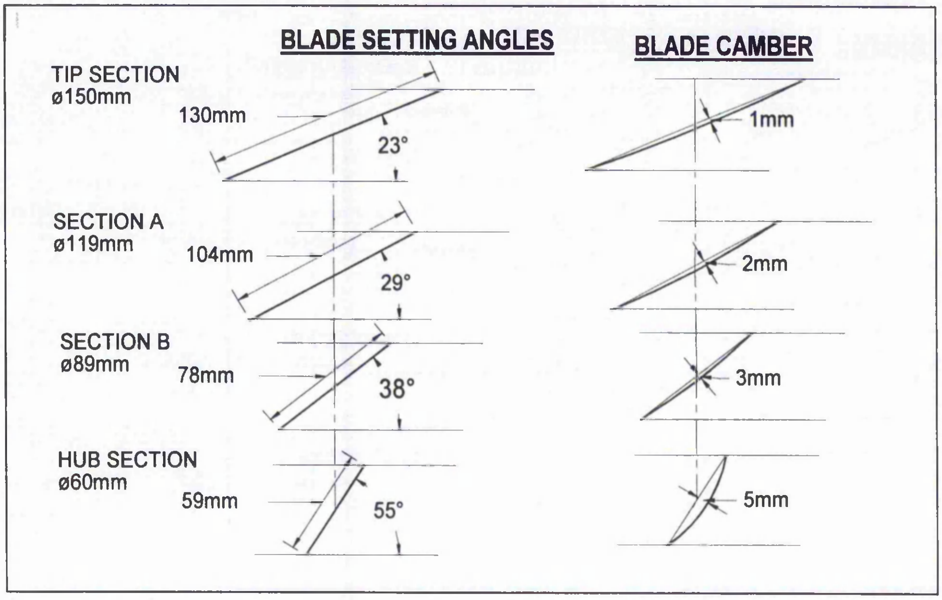
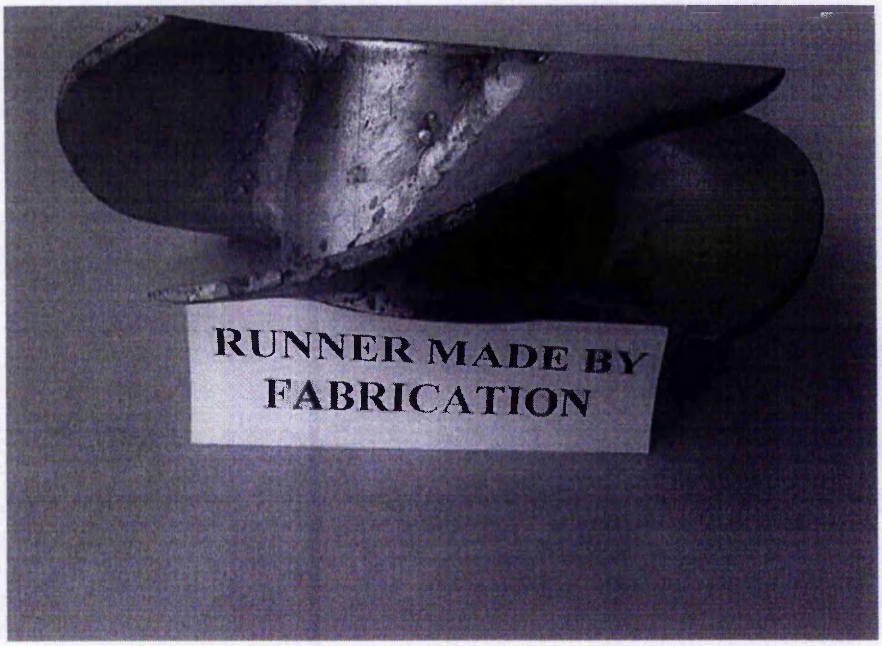


Figure 3.20 : Details of blade setting angles and blade camber for the prototype runner.

3.4.5 Fabrication of the runner blades

The fabricated prototype runner is shown by *Picture 3.1*. The value of the *chord length* is of great importance and has carefully been selected during the development of the prototype runner blades, see shaded areas in *Table 3.5* and *Table 3.6*.



Picture 3.1 : The fabricated prototype runner.

Both the hub and tip sections of the blade have the same height in the y-axis. For all sections of each blade a compromise with regard to chord length was made. A long chord length at the hub section was used to assist the welding of the blade onto the runner hub whereas the rest of the blade dimensions were selected in such a way so that the shape of the blade to be easily cut out of a metal plate. The tip chord length was then determined and the blade was cut from a metal sheet.

In order to identify the difficulties related to fabrication, it was decided to fabricate the runner blades using basic tooling and equipment that are commonly found in workshops of developing countries. Each 3mm constant thickness blade was attached onto the runner hub by arc welding at an angle of 55°, a value determined by the design calculations. With the aid of a vice and a wrench, each blade was given a slow gradual twist at the tip section. No camber was given to the blades.

3.4.5.1 Assessment of the fabrication method

As it is clearly seen from the design calculations, the blades have a high twist between the hub and tip sections. This is a result of the high operating speed used and the relatively small value of the hub to tip ratio. Bending and twisting flat plates to form the desired blade shape has proved to be a challenge. As it can be seen from *Picture 3.1*, both the trailing and leading edges of the blade resemble an S-shape blade, a geometry which is very different to what the original design is, as shown by *Figure 3.21*.

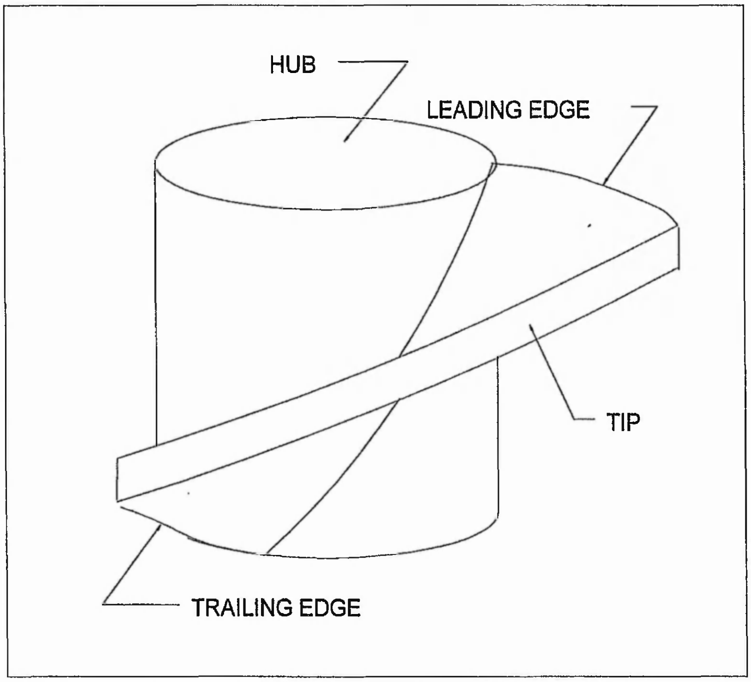


Figure 3.21 : The prototype turbine runner.

The major disadvantages related to fabrication are :

- The geometrical accuracy of the blade can only be achieved if there is very little twist between the hub and tip sections.

- Blades can only be fabricated when no camber exists. Though the blade thickness allows bending of the blades to form the design camber, twisting of the blade tip section is difficult due to added strength of the metal sheet.

- Arc welding causes thermal distortion and therefore geometrical inaccuracies.

3.4.5.2 Alternatives to fabrication

The runner hub and blades can be made as one piece on a milling machine for best possible accuracy, but this would mean that manufacturing of the propeltric unit would be more difficult to achieve in developing countries and at higher costs. Alternatively, a device (like a manually operated press) can be used to form the runner blades. This method however, would require special tooling which would be cost effective only when is used for mass production of runner blades.

The use of casting is an attractive method for producing turbine runners. However, the complexity of the runner geometry would require careful design of casting molds unless each blade is made separately (as it is the case with the Pelton Wheel buckets) and then are attached onto the runner hub. The attachment of the runner blades onto the runner hub would require careful consideration and a significant amount of workshop time would be spent for doing so. Moreover, due to the wide diversity of skills, standards and materials (especially in the South East Asia region), the reliability and quality of the finished runner would inevitably vary.

3.4.5.3 Runner blades made on a milling machine

A second runner was made using a milling machine, to determine what effect the inaccuracy due to fabrication had on the turbine overall performance. The runner blade geometry was generated on Unigraphics CAD system and then each blade was machined on a CNC milling machine. *Figure 3.22* shows the geometrical generation of the blade pattern and the details of the cutter paths required to machine the blade out of a solid metal block. Each blade was machined separately due the limited rotational movements imposed by the milling machine table.

In order to avoid geometrical distortion due to the welding the blades onto the runner hub, retaining slots were machined onto the cylindrical surface of the hub. As a result that the hub section of the blade was not a true helix, machining of these retaining slots, which were 5mm in depth, was difficult. Moreover, slots had to be machined much wider than the blade thickness.

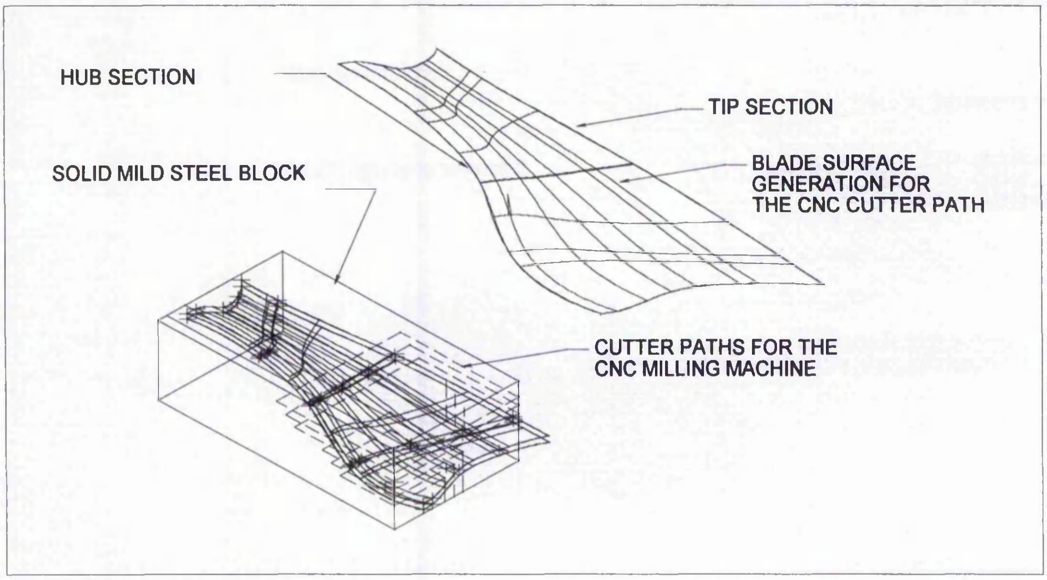


Figure 3.22 : Blade pattern made on a CNC system.

Picture 3.2 shows the runner with the machined blades welded in position. The fabricated runner would be referred to as *runner 1*, while the runner with machined blades would be referred to as *runner 2*. Both runners were tested during the course of this research project for comparison and the conclusions with regard to the effects of geometrical accuracy on the turbine overall efficiency are found in Chapter 4.

Picture 3.2 : Runner made using a CNC milling machine.

3.5 Inlet system design

The inlet system of the turbine unit consists of a *spiral casing* and *radial guide vanes*. The advantages of such arrangement are as follows :

- ❑ The turbine runner can be directly coupled to the generator shaft, as in the case of Peltric systems.

- ❑ Minimum civil works are required and ease of installation on site as the casing can provide the means for supporting the unit and the draft tube.

3.5.1 Spiral casing design

Information on spiral casing design available in literature refers mainly to large scale hydroelectric plants and the use of spiral casings is very uncommon with low head, low cost micro hydro schemes. Nechleba [1957] suggests that instead of using a spiral casing, the turbine can be placed into a concrete pit with its centre line offset for small *net heads* (maximum of 10m), regardless whether it is a Francis or Kaplan turbine. Kovalev [1963] on the other hand, recommends that the open flume layout is used for low heads. However, field trials of open flume layouts in Sri Lanka and the UK[10], have shown that their use is often accompanied by problems related to air drawn into the turbine and long vibrating shafts. The use of the spiral casing on the other hand, would allow more reliable operating conditions as well as the direct coupling of the turbine runner onto the shaft of the induction generator. The performance of the casing is examined in Chapter 4 where conclusions are drawn with regard to its performance during testing.

The spiral casing is an important component of a hydraulic turbine with its main purpose being to guide the incoming flow uniformly along the circumference of the turbine runner. The spiral casing is characterized by the *nose angle*. This is the angle between the principal radius of the casing and that defining the location of the nose. For the purposes of this research project, it was decided to use a 20° nose angle for the design of the spiral casing as shown by *Figure 3.23*. The decision for using 20° for the nose angle was based on the recommendations found in literature.

[10] Details of field work in Sri Lanka and the UK are available in Appendix B.

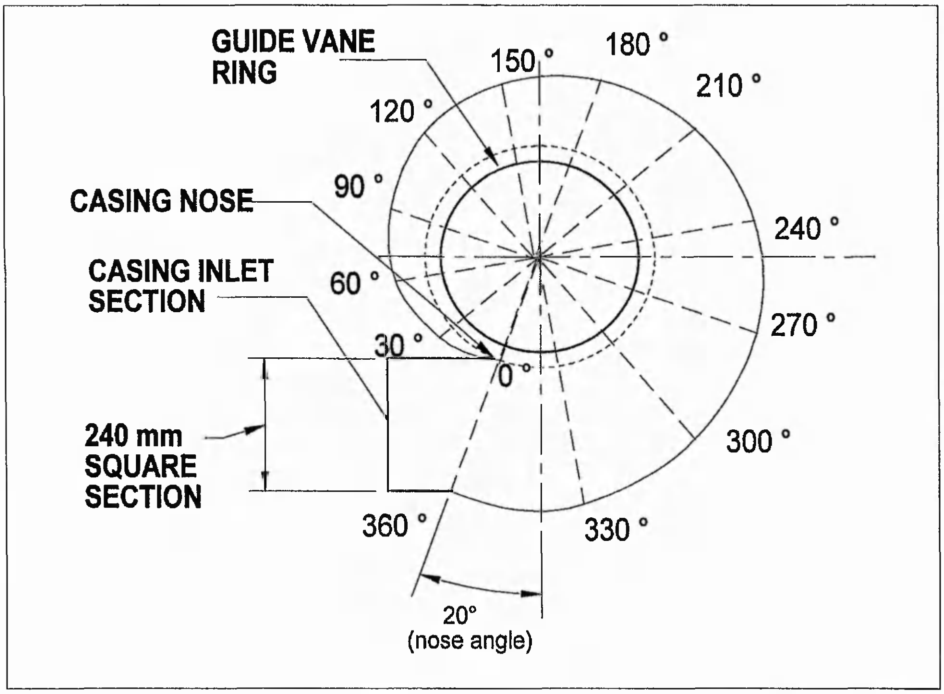


Figure 3.23 : Spiral casing layout.

The design of the spiral casing can be done by using one the following two methods :

- Using *constant angular momentum.*
- Using *constant average velocity.*

The first method implies that the angular momentum of a fluid particle in the casing is constant, i.e. $V_u\, r = constant$. This method is widely used for the design of large machines [Audisio, 1991 and Krivchenko, 1993]. The cross section of these designs is in most cases of an elliptical shape, but a circular section is also used for designing and manufacturing simplicity. Bohl [1992] and Mosonyi [1957] favor the trapezoidal section. The method of constant angular momentum is used by many turbine manufacturers who employ three dimensional analysis of the flow in the casing.

The present design is based on the second method where the average fluid velocity in the spiral casing is assumed to be constant at any position. As Nechleba [1957] suggests, this method is much easier to use as far as simplicity is concerned and yet reliable enough for correct flow distribution. The overall dimensions of the casing would be bigger compared with the dimensions that would have been obtained using the method of constant angular momentum.

The velocity at the inlet of the casing is calculated using equation :

$$V_{ce} = k_{ce}\sqrt{2gH_N} = 1.25\,m/s \qquad \{3.37\}$$

The value of k_{ce} is an empirical value and there seems to be a wide range of values for k_{ce}. While Bohl [1992] suggests a value of 0.15 - 0.25 for k_{ce}, information published by Nechleba [1957] indicate that for a 2m head k_{ce} should be equal to 0.26. The value of the k_{ce} was taken to be 0.2 following recommendations given by Mosonyi [1957]. Since the average velocity is constant at any section of the casing and equal to the entry velocity to the casing, with reference to *Figure 3.24*, the flow rate though the casing is given by

$$q = \frac{\lambda}{360^\circ}Q \qquad \{3.38\}$$

The flow distribution in the spiral casing is shown by *Figure 3.24* and is based on the work presented by Nechleba [1957].

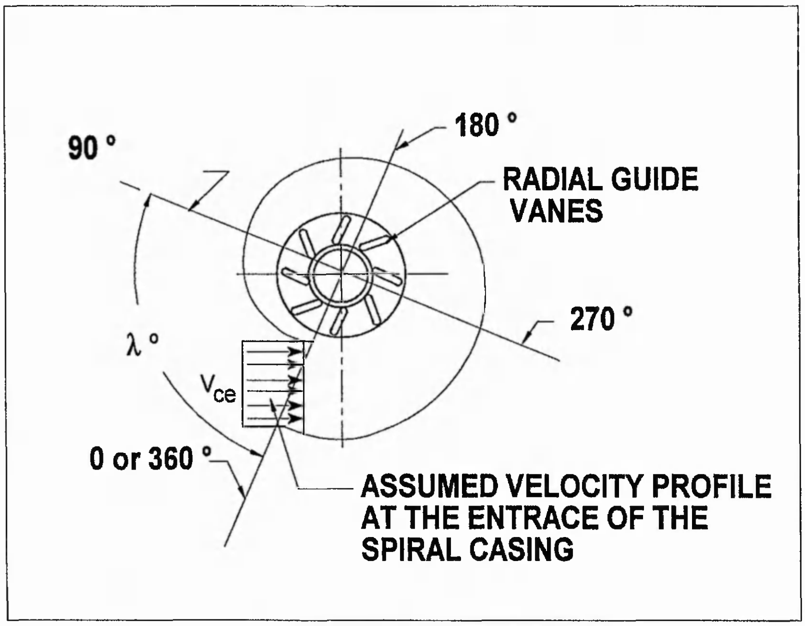


Figure 3.24 : Flow estimation.

The height of the casing is kept constant and equal to the height of the intake section for manufacturing and design simplicity. *Table 3.4* shows the dimensions of the spiral casing. Details of these dimensions are shown by *Figure 3.25*.

λ (°)	q (m^3/s)	cross sectional area (m^2)	x-dimension (m)
0	0	0	0
30	0.006	0.00483	0.020
60	0.012	0.00967	0.040
90	0.018	0.01450	0.060
120	0.024	0.01934	0.080
150	0.030	0.02417	0.100
180	0.036	0.02901	0.120
210	0.042	0.03384	0.140
240	0.048	0.03868	0.0160
270	0.054	0.04351	0.180
300	0.060	0.04835	0.200
330	0.066	0.05318	0.220
360	0.072	0.05802	0.240

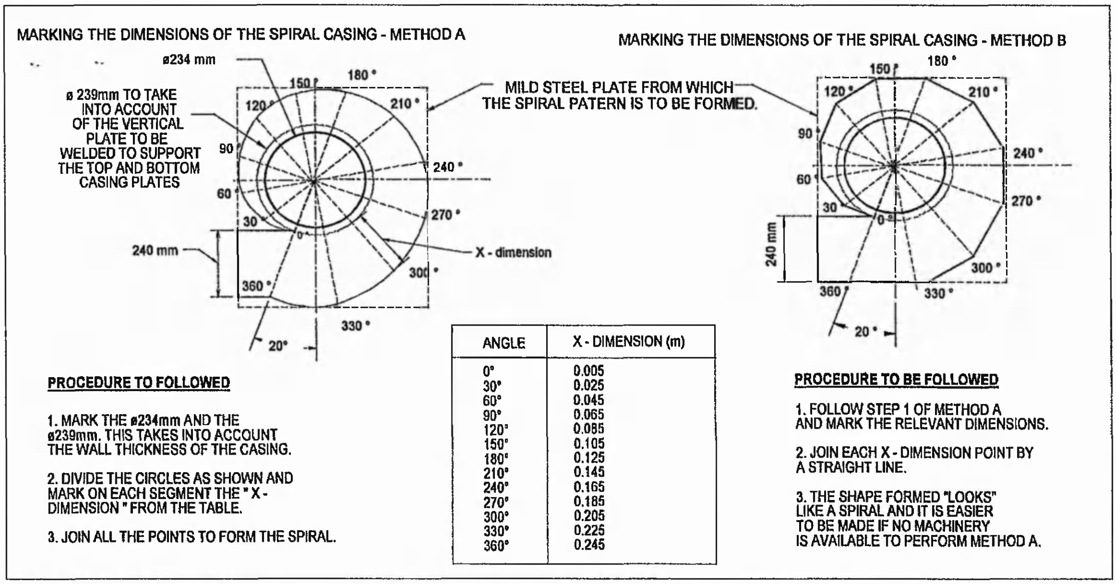
Table 3.4 : Spiral casing design calculations.

During the design process, it was assumed that the velocity profile at the intake of the casing, shown by *Figure 3.24*, would be maintained at all sections. The casing and guide vane design was therefore based on this assumption.

Losses in the spiral casing were at this stage not considered. However, the design of the spiral casing is evaluated in Chapter 4 using the experimental results. Main emphasis is placed upon the velocity distribution in the casing and a revised design approach is proposed and explained using the law of constant angular momentum.

3.5.2 Fabrication of the spiral casing

Figure 3.25 shows the dimensional details for manufacturing the spiral casing. The shape of the casing was modified[11] as shown by both *Figure 3.25* and *Figure 3.26* to assist fabrication. The casing was fabricated using steel plates 5mm in thickness. Thermal stresses due to welding caused geometrical distortion which was corrected with the aid of a hydraulic press.



ANGLE	X - DIMENSION (m)
0°	0.005
30°	0.025
60°	0.045
90°	0.065
120°	0.085
150°	0.105
180°	0.125
210°	0.145
240°	0.165
270°	0.185
300°	0.205
330°	0.225
360°	0.245

Figure 3.25 : Spiral casing development.

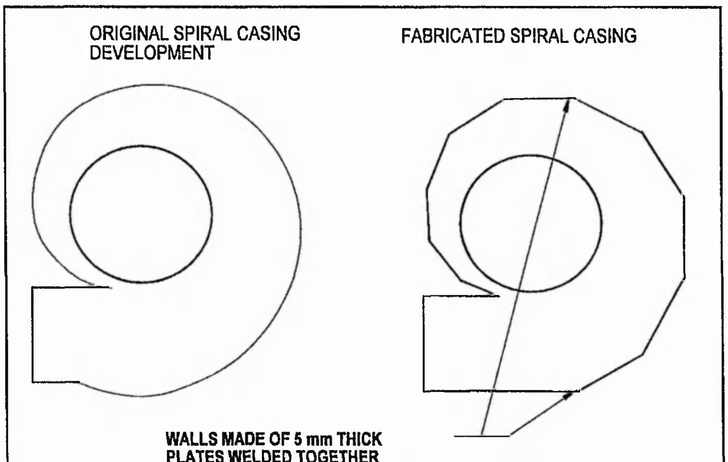


Figure 3.26 : Proposed casing layout.

[11] While designing the casing, the difficulties imposed by the shape of the spiral side walls were assessed and was therefore decided to alter the shape of the spiral casing to assist fabrication and reduce costs.

3.5.3 Guide vane design

The purpose of the *guide vanes* is to supply the turbine runner blades with the design flow rate at the correct angle. Though a number of designers have over the years suggested to avoid using even number of guide vanes [Nechleba, 1957], Bohl [1991] and Sigloch [1993] adopt exactly the opposite approach and recommend the number of guide vanes being a multiple of four, shown by *Table 3.5*. These data are based on collection of turbine designs that the above authors have incorporated in their textbooks. The advantage of using even numbers is that it would be much easier for local workshops in developing countries to divide a circle into even sections using their existing experience and/or equipment.

D_T	up to 0.25	0.25 to 0.5	0.5 to 0.7	0.7 to 1.2	1.2 to 2.2	2.2 to 4
Z_g	8	12	14	16	20	24

Table 3.5 : Selection of guide vane number, [adopted from Sigloch, 1993].

The length of the guide vane is recommended to be 0.15 to 0.2 of D_T. The eight fixed guide vanes, 30mm long, were cut from a flat sheet metal plate[12] and were not given any shape due to their small dimensions. The leading edge was filed to a radius of 0.2 of the maximum camber of the runner blades and the trailing edge thickness was trimmed to about 4% to 6% of the runner blade length.

3.5.4 Setting of the guide vanes

The *absolute velocity* at any section of the *guide vane ring* consists of two components and is given by the following equation

$$V_g = \sqrt{V_{ug}^2 + V_{ag}^2} \qquad \{3.39\}$$

The *guide vane setting angle*, which is the angle between the component of *absolute velocity* and the *axial velocity*, is equal to

$$\tan\alpha_g = \frac{V_{ag}}{V_{ug}} \qquad \{3.40\}$$

[12] Details are found in Appendix G.

The *absolute velocity* leaving the guide vanes, shown by *Figure 3.27,* was designed to approach the runner blades with the same direction as the one calculated for the *flow absolute velocity*. Guide vanes are normally set to match the absolute velocity angles at the mid section of the runner blades. For the purposes of this research project, the tip section angle is used where the biggest surface area is available. The *peripheral* component of the fluid velocity leaving the guide vanes was calculated using the constant angular momentum theorem between the runner tip section and the exit of the guide vanes as follows :

$$V_{ugi} = \frac{V_{u1T}\, D_T}{D_{gi}} = \frac{1.205 \times 0.15}{0.16} = 1.1297\, m / s \qquad \{3.41\}$$

The *axial velocity* at the exit of the guide vane is found using equation {3.40}. The height of the guide vanes is therefore given by :

$$b_g = \frac{Q}{\pi\, D V_a} = 0.035m \qquad \{3.42\}$$

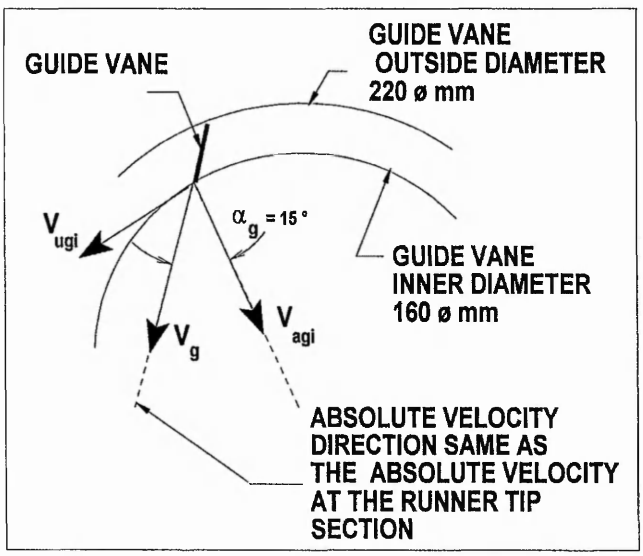


Figure 3.27 : Guide vane setting.

The effects of guide vane setting are discussed in Chapter 4 based on the test results.

3.6 Draft tube design

The draft tube is a significant part of a reaction turbine and has a twofold purpose in a reaction turbine :

- ☐ To utilize the vertical distance between the turbine exit and tailwater level, termed *static head.*

- ☐ To regain, by its expanding shape, most of the kinetic energy inherent in the discharge leaving the turbine. The regainable head is referred to as the *dynamic draft head.*

The effects of draft tube inlet velocity and pressure recovery are very important on the overall system performance [Moses, 1982]. Overall turbine efficiency and turbine losses are related to the recovery of kinetic energy in the draft tube. Today, reaction turbine manufacturers consider the design of the draft tube as important as any other part of a hydro electric scheme and state of the art technology and Computational Fluid Dynamics (CFD) are used to improve efficiencies and velocity distributions.

With reference to *Figure 3.28*, using the tailwater level as a datum, Bernoulli's theorem is applied to an element leaving the runner and draft tube respectively,

$$\frac{P_2}{\rho g} + \frac{V_2^2}{2g} + Z_s = \frac{P_3}{\rho g} + \frac{V_3^2}{2g} + \xi_{dt} \qquad \{3.43\}$$

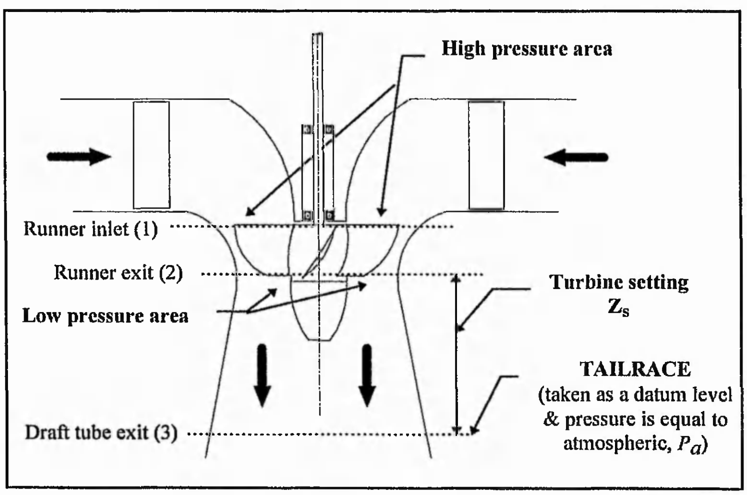


Figure 3.28 : Turbine setting.

The losses in the draft tube arise mainly from the frictional and eddy losses. The conversion of kinetic energy into potential involves more losses than vice versa. The difference between potential energies at the runner exit and at the tailwater level respectively is also utilized by the turbine. As the velocity of water at the turbine exit is V_{a2}, it is obvious that due to the diffuser effect in the draft tube, the head is regained and it is termed as the *dynamic draft head*,

$$Dynamic\ head = \frac{V_{a2}^2}{2g} - \frac{V_{a3}^2}{2g} - \xi_{dt} \qquad \{3.44\}$$

From the above equation, since the energy used for the conversion in the diffuser is given by $\frac{V_{a2}^2 - V_{a3}^2}{2g}$, the diffuser efficiency may be defined as follows, as given by Mosonyi [1957] and Hutton [1959]

$$\eta_{diffuser} = \frac{V_{a2}^2 - V_{a3}^2 - 2g\xi_{dt}}{V_{a2}^2 - V_{a3}^2} \qquad \{3.45\}$$

However, the efficiency of the draft tube can not equal that of the diffuser, since in the case of the draft tube, in addition to ξ_{dt}, the velocity head $V_{a3}^2 / 2g$ remaining at the bottom of the draft tube is also to be considered as a loss. The efficiency of the draft tube should be correctly computed by regarding the entering kinetic head $V_{a2}^2 / 2g$ as the energy converted or utilized in the draft tube. Thus,

$$Dynamic\ head = \eta_{dt} \frac{V_{a2}^2}{2g} \qquad \{3.46\}$$

Combining equations {3.44} and {3.46}, the draft tube efficiency yields

$$\eta_{dt} = \frac{V_{a2}^2 - V_{a3}^2 - 2g\xi_{dt}}{V_{a2}^2} \qquad \{3.47\}$$

During the analysis described so far, it was assumed that the entrance velocity in the draft tube is equal to the exit velocity from the turbine runner and that there will be no exit whirl at the runner. The efficiency of a straight draft tube is given by *Figure 3.29* and the losses can therefore be estimated using equation {3.47}.

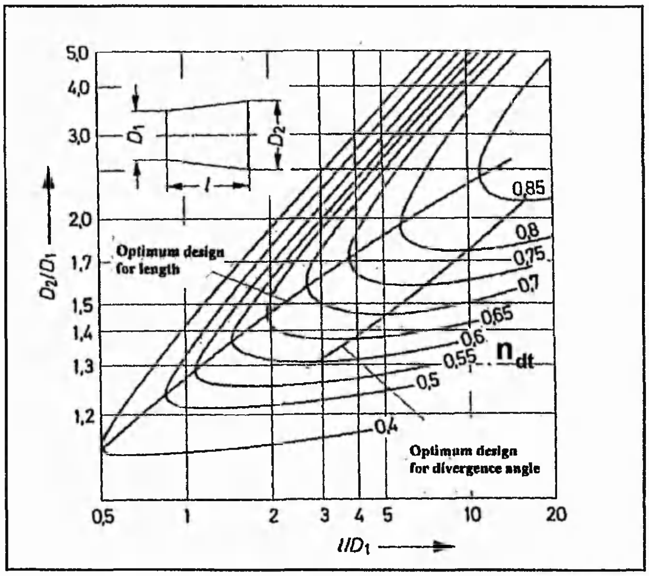


Figure 3.29 : Draft tube efficiency, [adopted from Bohl, 1991].

The *straight conical draft tube* is used for this particular design mainly due to the simplicity of manufacture. This particular shape, shown by *Figure 3.30*, is characterized by its *length*, *divergence angle* and the *ratio* of the *areas at inlet and exit.* All of these parameters require attention as cavitation and flow separation may result if they are not selected carefully.

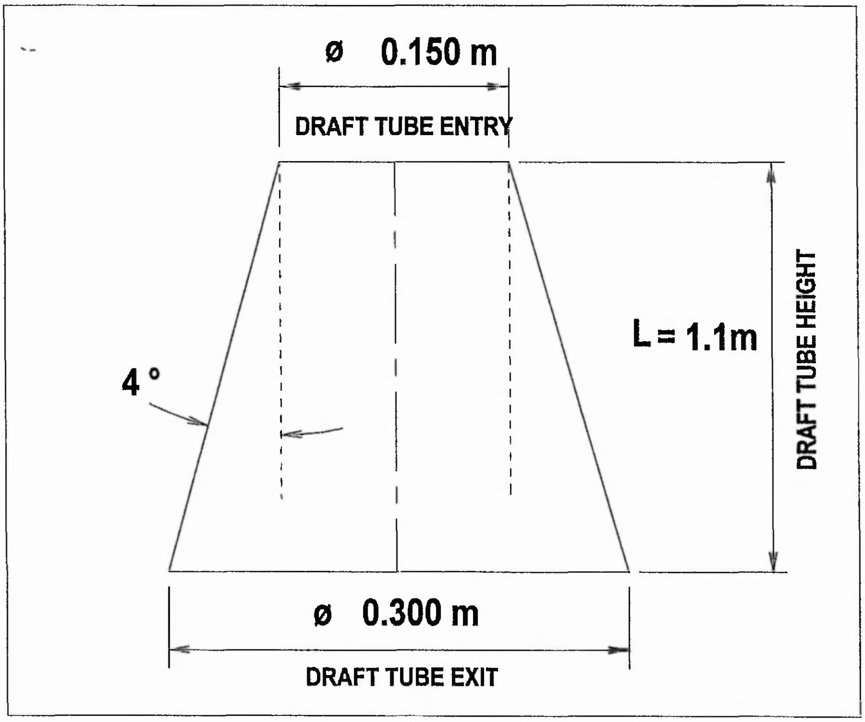


Figure 3.30 : Draft tube geometry.

First of all, the *permissible height* at which the turbine is set above the tailrace is determined as explained in section 3.7 of this Chapter. Flow separation occurs when the draft tube divergence angle is too large. Ideally, the draft tube angle, at which both friction and separation losses are minimum, is between 8° and 10°. As indicated by experiments [Mosonyi, 1957 and Mosses, 1982] losses increase sharply at angles wider than 10° to 12°. The draft tube angle is given by :

$$tan\frac{\vartheta}{2}=\frac{D_3-D_2}{2L} \tag{3.48}$$

Using *Figure 3.29*, for a draft tube efficiency around the value of 82%, a diameter ratio of two, the length of the draft tube is about 1.1m. For low head schemes, the length of the draft tube has to be kept as short as possible to avoid excessive civil works and therefore extra costs. Using equation {3.48} draft tube convergence angle is calculated to be 8°, which is within the acceptable range.

3.7 Cavitation and turbine setting

Hydraulic turbines having water as a working fluid are susceptible to a phenomenon which limits their performance under certain conditions. This phenomenon is known as *cavitation*, and is defined as

> *"the local vaporization of water due to the operating conditions which becomes visually apparent as vapour bubbles in the water and is usually accompanied by noise, vibration, and drop in turbine efficiency*" [Arndt, 1981a].

The characteristic types of cavitation in turbines [Krivchenko, 1994] are as follows :

- *Profile*, originating when blades are exposed to cavity flow in the region of lower pressure, usually round the area of the trailing edges of blades.
- *Slotted*, when water flows at a great pressure drop through clearances, for instance between the runner blades and the turbine case.
- *Local*, caused by water flowing past irregularities, e.g. bolt heads.

The effects of cavitating flow on a hydraulic turbine can be very serious :

- With a sufficiently developed cavitation hydraulic losses increase, resulting in lower efficiency and decreased discharge.
- In the presence of cavitation a characterizing noise is heard, and a higher than normal vibration appears that can cause structural damage.
- When a hydraulic turbine operates under cavitating conditions, erosion and pitting of surfaces, especially of the runner blades, is rapid at places where cavities collapse.

Cavitation is prevented when the *local pressure* at any point of the flow passages is kept higher than the *vapour pressure of the water*, i.e. $P > P_V$. In practice real pressures within the turbine are not known. The *Thoma cavitation constant* is therefore used to determine the *permissible setting* for a turbine to prevent cavitation. The relationship between turbine setting and the Thoma cavitation coefficient is given by

$$H_s \leq (H_b - H_v) - \sigma_{Th} H_N \qquad \{3.49\}$$

The value of σ_{Th} is a function of the type of turbine used, i.e., the *specific speed* of the turbine [Arndt, 1991; Pearsal, 1974 and Turton, 1984]. Since there is no data available for σ_{Th} of propeller turbines, the empirical data presented by Bohl [1994] for Kaplan turbine designs would be used. With reference to *Figure 3.31*, which was adopted from Bohl [1994], the value of σ_{Th} is 1.28 for a specific speed of 248 min.$^{-1}$.

Using equation {3.49}, the *maximum permissible setting*[13] for the prototype propeller turbine is about 7.3m. In order to check whether the above condition is true, with reference to *Figure 3.28*, the Bernoulli equation is applied between the exit section of the turbine runner (assumed to be the same as the entry to the draft tube) and exit from the draft tube at tailrace level as follows :

$$\frac{P_2}{\rho g} + \frac{V_{a2}^2}{2g} + Z_2 = \frac{P_3}{\rho g} + \frac{V_{a3}^2}{2g} + Z_3 + \xi_{dt} \qquad \{3.50\}$$

[13] For the purposes of this research project the water vapour pressure is taken to be 2340 N/m^2 @ 20°C.

For a draft tube efficiency of 80%, the pressure at the runner exit is 1.04 bar. The 7.3m turbine setting limit suggests that the runner blades of the turbine are not in any danger of cavitation since the statement $P_2 > P_v$ is true. However, care was taken while the turbine was fabricated to avoid sharp edges and surface irregularities in the spiral casing, the runner blades and the draft tube which could result into local pressure drops and therefore favour local cavitation [Arndt, 1981b & 1979].

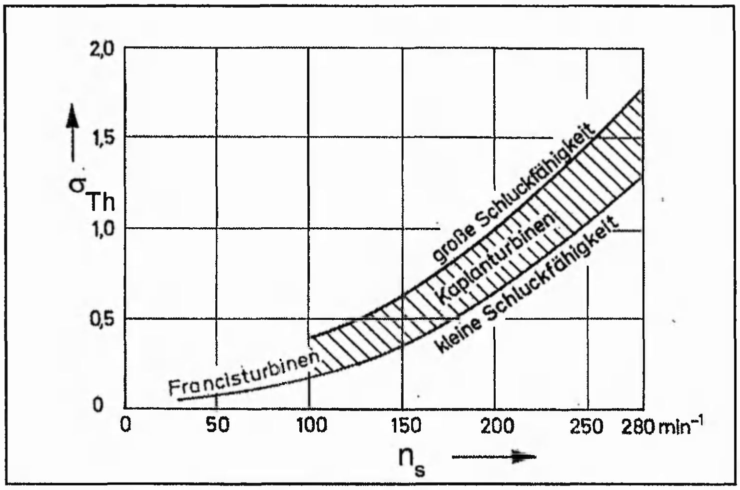


Figure 3.31 : Thoma cavitation coefficient, [adopted from Bohl, 1994].

3.8 Material selection

The essential requirements for a successful micro hydro turbine are *performance* and *life*. The *performance* of the turbine is determined by the method used for the hydraulic design, testing and development work while *life* is defined as the total number of hours or days before one or more parts of the turbine must be replaced to maintain an acceptable performance. *Life* depends on the resistance of the materials used for the various parts of the turbine to *abrasive* and *cavitation wear*.

3.8.1 Material selection for abrasive wear prevention

Abrasion is understood to be the removal of material due to microscopic surface tearing by solid particles, that are transported in water, on contact with the surface material. The extent of material removal depends on the velocity and force with which these particles act on a particular surface. The wear rate is proportional to velocity cubed.

Despite using silt settling facilities, abrasive silt content in water (particles of which are usually of 50 to 300μm in size) is high due to deforestation and frequent storms. Reports from field workers in Sri Lanka and Nepal indicate excessive erosion of Pelton wheel buckets which require frequent replacement, sometimes every couple of years. In most cases, the bronze buckets erode around the area which is hit by the water jet, i.e. the high velocity areas. In the Northern part of India it has been reported that most of the turbines suffer serious damage due to abrasive wear and plants are often shut down for repairs [Swaroop, 1993]. In view of this, efforts are being made especially by turbine manufacturers, to find ways of avoiding or reducing abrasion damage.

Abrasion damage in Kaplan turbines may occur at the suction surfaces of the blades, more towards the tip and trailing edges. Reduction of flow velocities through the turbine and reshaping of the runner blades while the turbine is designed can prevent blade erosion. However, there are cases that the design requirements do not allow any of these measures to be taken and therefore limit the designers to the use of material that would slow down or even prevent abrasive erosion. For example, Sulzer Wyss, the Swiss turbine manufacturer, has recently developed a ceramic known as the SXH48, which can be sprayed with the aid of a high velocity flame onto the turbine blades. Though it is claimed that SXH48 provides good protection against small sand grains, details of the extend of its success are not available [Swaroop, 1993]. Polymer overlays on turbine blades have also been used [Armentrout, 1993]. Though very successful in slowing the phenomenon of cavitation and abrasive erosion, their use would require specific skills not easily found in small workshops in Third World countries. Moreover, these methods are suitable for mass produced components and would therefore be very uneconomical for one off use.

The design of this prototype turbine is governed by simplicity of manufacture and low costs. When considering materials for designing a pump or a turbine, the erosion properties of any material must be evaluated. However, this can be a difficult task since many of these properties and characteristics of both the fluid flowing through the machine and the materials under consideration are not known in sufficient detail. According to Karassik [1976] most materials used in pump construction have been corrosion-tested under static conditions in a wide variety of liquids, but very few have been tested in a high velocity abrasive environment.

The effective abrasion resistance of any metal will depend on its position on the Mohs or Knoop hardness scale, shown by *Figure 3.32*. Cast iron and bronze should be avoided when the silt content is high. Stainless steel on the other hand, though commonly used in large scale hydroelectric schemes, would result in additional material costs. Moreover, welding of stainless steel requires skills that would not be available in small workshops in developing countries.

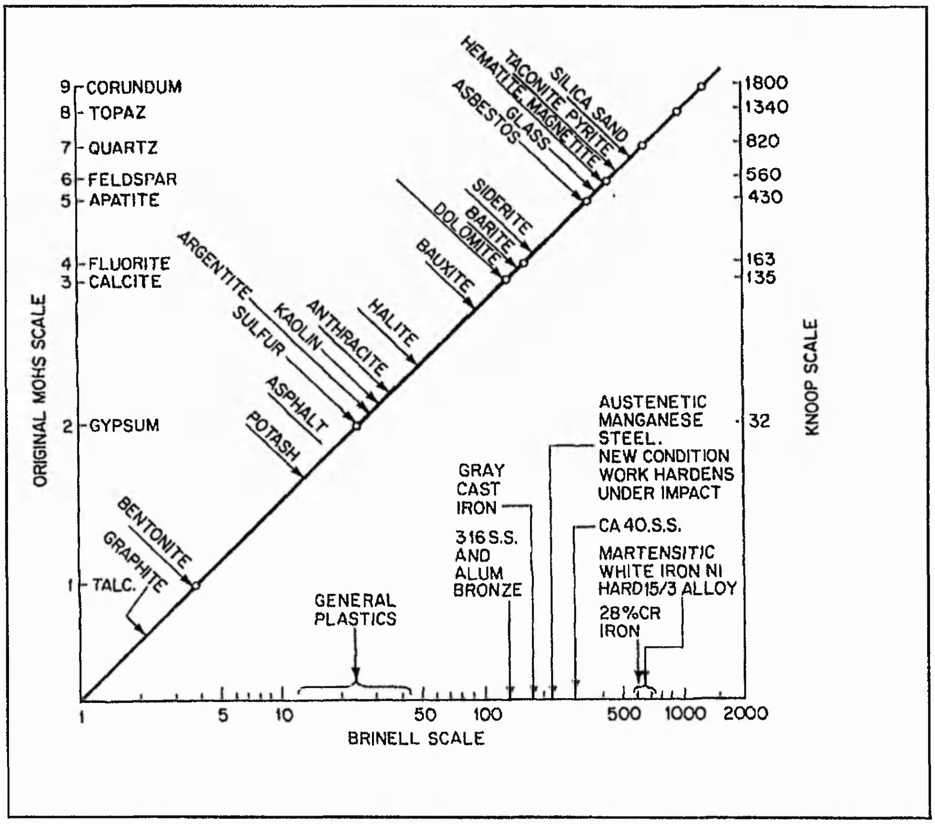


Figure 3.32 : Approximate comparison of hardness values of various common ore minerals and metals, [adopted from Karassik, 1976].

The use of mild steel is the most appropriate option for the following reasons :

- ☑ It is widely available and used by small workshops.
- ☑ It is a low cost material compared to aluminum.
- ☑ It has better abrasion resistance than bronze.

3.8.2 Selection of materials for cavitation prevention

Though it has been examined and proved that the turbine would not be subjected to cavitation, section 3.7, cavitation may still occur locally due to surface irregularities. The resistance of materials to cavitation depends both on the chemical composition and the method of machining and heat treatment used, and in particular, on the smoothness of the surface. The behavior of metals subjected to cavitation is similar to that of metals subjected to corrosion [Lazarkiewicz, 1965]. Any notches, nicks, scratches, flaws or sharp corners on the surface of metals attacked by cavitation accelerate the beginning of pitting [Arndt, 1979]. *Table 3.4* shows some experimental results with regard to the weight loss of a variety of metals subjected to cavitation.

Metal	Weight loss after 2 hours (mg)
Rolled stellite (High cost and difficult to machine)	0.6
Welded aluminum bronze (83% Cu, 10.3% Al, 5.8% Fe)	3.2
Cast aluminum bronze (83.1% Cu, 12.4%Al, 4.1% Fe)	5.8
Welded stainless steel (two layers, 17%Cr, 7% Ni)	6.0
Cast stainless steel (12% Cr)	13.0
Welded mild steel	**97.0**
Aluminum	124.0
Brass	156.0
Cast iron	224.0

Table 3.4 : Weight loss in materials, [adopted from Warnick, 1984].

With reference to *Table 3.4*, it is evident that due to the high operating speeds required for direct coupling to induction generators, it would be best if cast iron and brass are not used. In addition to the advantages of using mild steel described in section 3.8.1, mild steel is easy to work with. Careful welding during fabrication would ensure good surface finishes and would therefore minimize the risk of local cavitation.

3.9 Contact or non contact seals ?

The layout of the propeltric unit requires a seal to be fitted where the turbine shaft exits the spiral casing. Conventional sealing devices for use in the sealing of rotating shafts can be split into two main categories, *contact* and *non contact seals*.

Contact seals are the most common and are classified as follows :

- ❑ *Compression packings* which use some form of packing material that is compressed round the shaft by a gland or a spigot. These seals are very reliable and give enough warning, as they usually start to leak, before failure, and are widely used in developing counties.

- ❑ *Radial face mechanical seals* consist of a contracting radial face which is maintained by a mechanical force, usually a spring. This type of seal is widely used in pumps, but their reliability is badly affected by the silt and sand content in water.

- ❑ *Rotary shaft lip seals* use a form of a rubber lip which presses against the shaft. Whilst stationary, sealing is achieved by a spring where as whilst rotating sealing is achieved hydrodynamically.

Experience has shown that life expectancy of contact seals is reduced by the eroding action of silt and sand in water [Viani, 1990]. Failure of the seal would result in excessive running costs with the scheme being shut down for repairs over prolonged periods of time. In addition, water escaping from the inlet casing would give rise to leakage losses and causes drop of power. It can also result in generator failure, especially when direct drive between the generator and turbine shaft is used.

The use of non contact seals has the following major attractions :

- ☑ Little or no maintenance requirements.

- ☑ Increased reliability in abrasive environments.

- ☑ No requirement for additional components.

The obvious disadvantage of non contact seals is that they only have a sealing effect when there is a rotating motion of the shaft.

3.9.1 Design considerations

The fluid flow associated with a disc rotating close to a stationary wall has been of vital importance to designers of rotating machinery. The idea of *non contact seals* originates from the gas turbine industry [Wood, 1964] and is generally described as a *centrifugal seal* for application to rotating shafts or *slinger seal* [Reshotko, 1968 and Thew, 1969]. The sealing effect is purely due to the shear forces acting on the liquid, which is enclosed by one rotating and one stationary disc [Mellor, 1968; Bayley, 1969; Köchler, 1971 and Sambo, 1983].

For the purposes of this research project, the method developed by Reshotko [1968] is adopted alongside experimental information from the development of a prototype *non contact seal* which has been designed and tested at Warwick University [Rees, 1995]. The general arrangement for the *non contact seal* is shown by *Figure 3.33*. The housing of an ordinary slinger seal is not part of this arrangement.

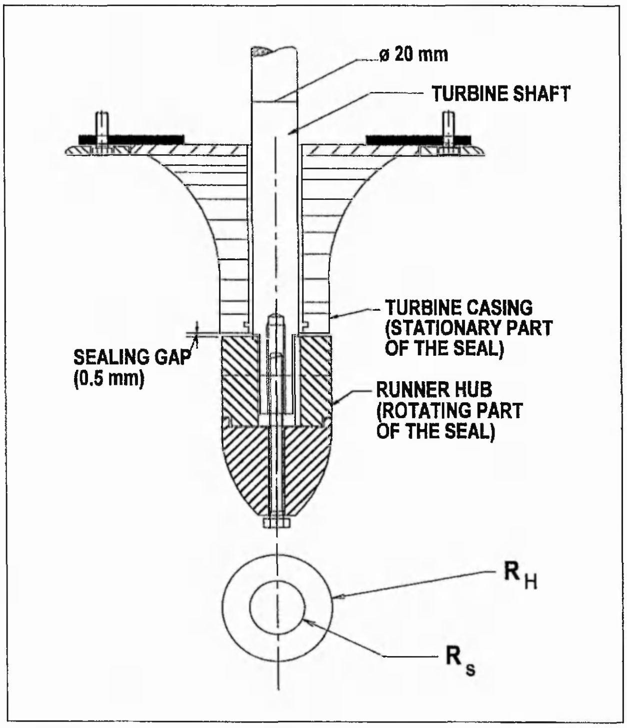


Figure 3.33 : Dimensions of the non contact seal.

The parameters that have been considered for the design of the *non contact seal* are :

- ❑ The gap between the stationary and rotating parts of the seal.
- ❑ The rotational speed of the turbine.
- ❑ The disc radius.
- ❑ The surface finish of the discs.

3.9.2 Design of the seal

As there are no changes in the temperature and viscosity of water flowing through the turbine, their effects have been ignored. The surface roughness height for the two discs, which were produced on a lathe, is 1.6 μm. Although the disc is not completely enclosed in a casing, it is assumed that the presence of the turbine blades, which are attached to the rotating disc (runner hub) form conditions similar to those described in literature [Daily, 1960 and Reshotko, 1968].

Daily [1960] conducted a comprehensive theoretical and experimental study of the fluid mechanics associated with a plane, smooth disc rotating in an enclosed casing for a rotational speed of $10^3 \le Re \le 10^7$. For the design of the non contact seal the rotational Reynolds number based on the axial spacing between the rotating and stationary disc is used which was derived by Mellor [1968] and Reshotko [1968]

$$Re_g = \frac{\omega s^2}{\nu} \qquad \{3.51\}$$

According to the work of Reshotko [1968], the pressure difference capability of the disc seal depends on k^2, where k is defined as the effective ratio of the angular velocity of the fluid to the angular velocity of the rotating disc. This is given by

$$\Delta P = \frac{\rho\ \omega^2 K^2}{2}\left\{r_H^2 - r_S^2\right\} \qquad \{3.52\}$$

The value of k, shown in *Table 3.4* , varies with Re_g for laminar flow.

$Re_g = \dfrac{\omega s^2}{\nu}$	k
$Re_g < 3$	0.540
$3 < Re_g < 10$	0.510
$10 < Re_g < 80$	0.333
$80 < Re_g < 1000$	0.248
$Re_g > 1000$	0.313

Table 3.4 : Empirical values of k used in the design process [adopted from Reshotko, 1968 and Rees, 1995].

The pumping effect (theoretical results) between the two discs is given by *Figure 3.34*.

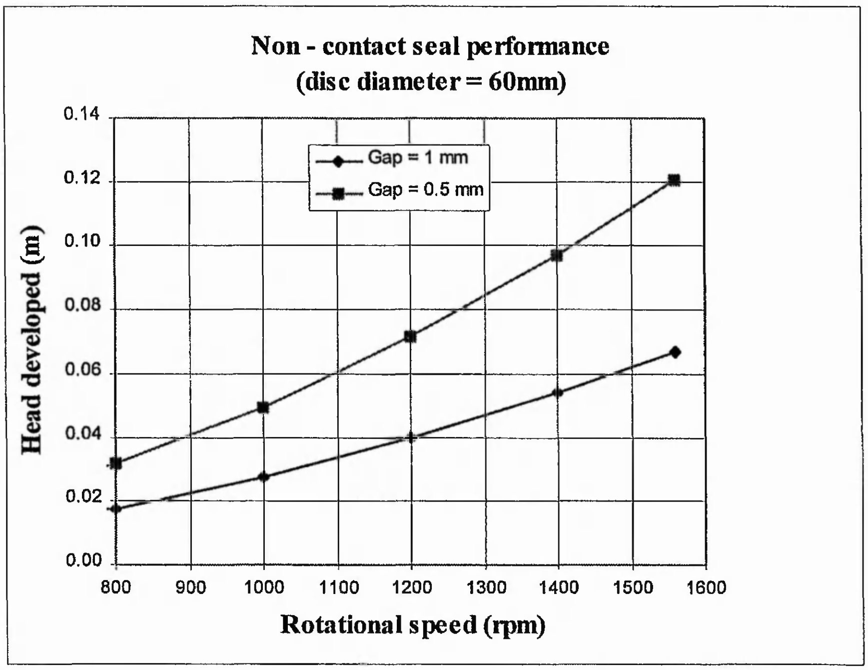


Figure 3.34 : Theoretical Performance of the non contact seal.

As it is clearly seen, the development of a head difference across the gap not only depends on the speed of the rotating shaft but on the gap between the discs as well. The gap is limited to 0.5mm due to the limitations imposed by the fabrication capabilities of local workshops in developing countries.

3.9.3 Power requirement for the non contact seal

The power requirement of the seal depends on the retarding torque exerted by the engaged fluid on the rotating disc. Reshotko [1968] gives the following equation for the torque required to drive the disc

$$T = \frac{\pi\ \mu\ \omega}{2\ s} G\, r_H^4 \left\{ 1 - \left(\frac{r_S}{r_H} \right)^4 \right\} \qquad \{3.53\}$$

where G is defined as the azimuthal velocity function, which depends on the rotational Reynolds number based on radial position, and is given by

$$G = 0.542\, Re_g^{1/2} \qquad \{3.54\}$$

The power requirements of the non contact seal are shown by *Figure 3.35.*

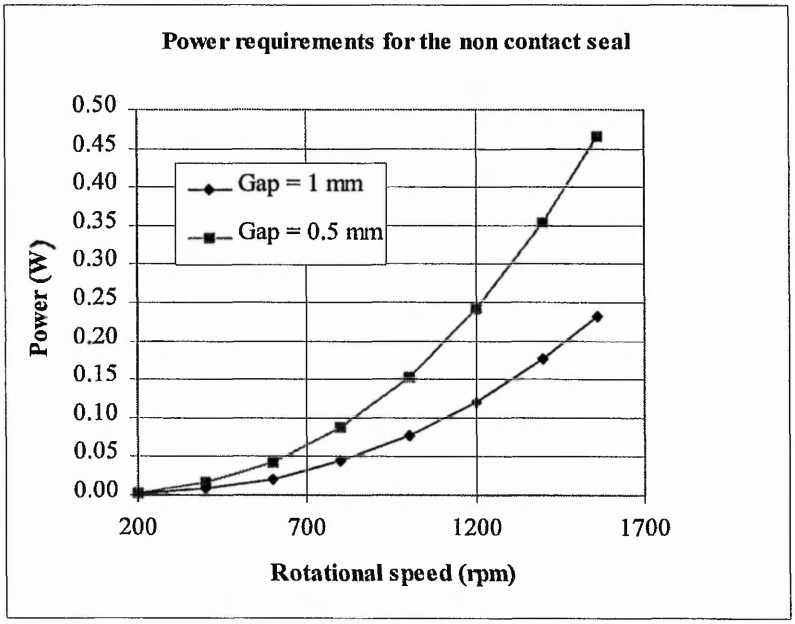


Figure 3.35 : Estimated power requirement for the non contact seal.

3.9.4 Concluding remarks on the non contact seal design

The work presented here is an investigation of *non contact seal* developments mainly for micro hydro schemes. The production of an accurate and complete model for the sealing system is possible but would require a test rig specially set up for testing non conatct seals.

The effectiveness of this *non contact seal* was investigated by observing whether there has been any water leakage during testing. These observations and conclusions with regard to the use of *non contact seals* for micro hydro schemes are found in Chapter 4.

As it is clearly seen from both *Figure 3.34* and *Figure 3.35*, the performance of the non contact seal depends not only on the operating speed of the turbine but on the gap between the sealing area as well as the diameter of the discs concerned. Rees [1995] has investigated three different disc sizes for the experimental investigation of the non contact seal with gap sizes down to 0.1mm and speeds going up to 1500rpm. The results concluded that discs with diameters of 300mm and 225mm were providing effective sealing whereas the 150mm diameter disc was not.

The seal can also be made with vanes on the rotating disc which will act in the same way as those of a centrifugal pump. This design has been investigated also by Rees [1995] who concluded that there is an increase in the pumping effect. However, such a design will require more power and there will be difficulties with its fabrication in developing countries.

3.10 Comparison with conventional design approaches - Conclusions

The design approach described in this chapter is in many ways different to those adopted for the design of innovative low head propeller turbines discussed in Chapter 2. Though there is a limited amount of information in literature regarding the method used for certain propeller turbine designs, in the authors opinion, the methodology presented in this chapter has the following advantages :

- ❑ Calculation of *leakage and mechanical losses* which have not been considered in any of the designs described in Chapter 2. With small propeller turbines, the gap between the runner blades and the casing has a significant impact on the *volumetric efficiency*, section 3.3.

- Estimation of *hydraulic losses* and therefore the value of the *hydraulic efficiency* rather than making assumptions based on information available in hydraulic turbine text books or steam/gas turbine performance prediction methods. A more detailed analysis of hydraulic losses is presented in Chapter 4.

- The use of operating speeds suitable for direct coupling of the turbine runner onto the generator shaft whose bearings can be used to take the axial and radial loads from the turbine runner. This arrangement has been tried with great success with Peltric sets, [Smith, 1992a].

- The use of a spiral casing and radial guide vanes that allows the generator to be as close as possible to the turbine runner and therefore assist direct coupling of the runner onto the generator shaft.

- The use of turbine overall dimensions tailored to standard sizes of pipe work fittings to facilitate fabrication. Discussion of the use of standard dimensions is expanded in Chapter 4, where a new propeller turbine design is introduced.

- The use of constant thickness sheets of metal for the runner blades instead of the conventional aerofoil sections which are difficult to make. The design of blades is discussed further in Chapter 4 where blade design parameters are varied to reduce the twist from hub to tip and therefore assist fabrication.

- The design of a non contact seal that would be an advantage to contact seals in silty conditions.

A simple design of a low head propeller turbine has been proved. The important feature of this prototype is the use of simple geometry circular arc blades made of constant thickness sheets of metal, as opposed to conventional design approaches where aerofoil profiles are used. The design also utilizes as much local skills and materials as possible while providing a low cost, reliable, robust and easily maintained system for the needs of remote communities.

Chapter 4

Experimental Work

and Design Evaluation

"What we anticipate seldom occurs; what we least expect generally happens"

Benjamin Disraeli

4.1 Chapter brief

This chapter introduces the setting up of a *power test rig* to obtain the performance characteristics of the prototype propeller turbine. The current set up complies with the guidelines set by the British Standards Institution (BSI), the International System Organization and the International Electromechanical Commission (IEC). Turbine tests were conducted for *constant head non cavitating conditions*. Test results are presented in tabulated and graphical form.

The aim of the work presented in this chapter is the review and analysis of the turbine component losses and the mechanisms associated with them. Conclusions are drawn with regard to the effect of velocity distribution in the spiral casing, shock losses, boundary layer and pressure distribution effects on the runner blade performance and velocity head losses at the draft tube exit. A review of the design methodology is presented and explained. The chapter concludes with a cost analysis for the prototype propeller turbine and a cost/kW comparison with a Pelton turbine manufactured by a Nepali workshop.

4.2 Test rig set up

The present set up of the *power test rig* allows the performance characteristics to be obtained under *non cavitating* conditions. Part of the test rig, which is not a closed type, was already in place in the Fluids Laboratory located in the Department of Mechanical & Manufacturing Engineering of Nottingham Trent University. The layout and dimensions are shown by *Figure 4.1* and *Figure 4.2* respectively. Water is pumped from a large tank[1] located under the floor level of the Fluids Laboratory. A 10 inch variable pitch axial flow pump is used to control the flow to the turbine. Fine adjustments to the pump's speed with a rheostat allow direct control of the flow rate and head while testing.

[1] The water in the tank had a constant temperature of 20° C during testing. The density and kinematic viscosity of water are taken at this temperature for the work presented in this chapter.

The current test rig set up complies with the requirements set by International Standards for the testing of hydraulic machinery [IEC, 1965 and 1973; BSEN 60041, 1994; BS5316, 1977 and ISO5167, 1980]. A data logging system is used for collecting test data and monitoring the characteristics of the turbine in real time.

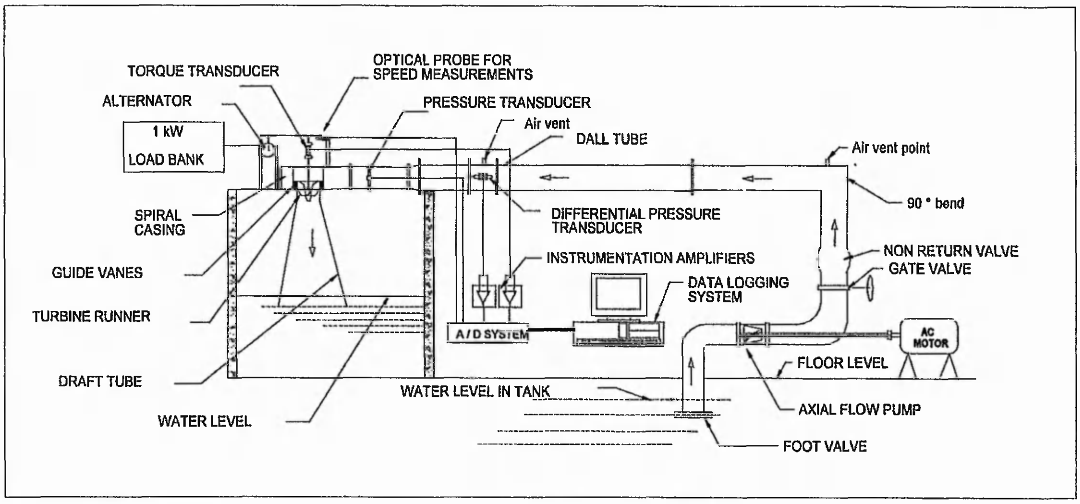


Figure 4.1 : Test rig layout.

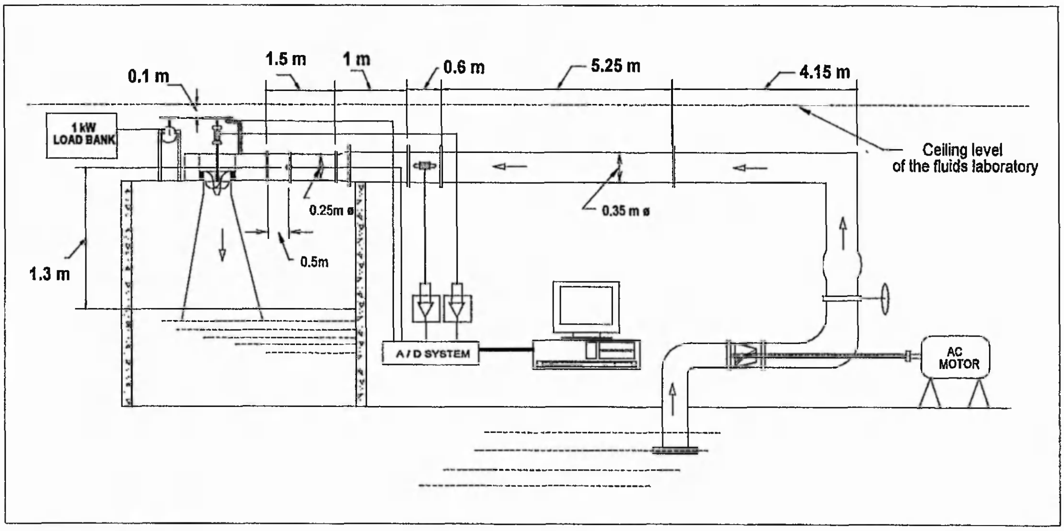


Figure 4.2 : Test rig dimensions.

Further improvements, such as the introduction of a flow conditioning device and pulsation dampers, which are recommended by BS1042 [1992], were not possible due to the limitations imposed by the dimensions of the Fluids Laboratory.

4.2.1 Head and flow measurements

A *Dall Tube,* which is a pressure differential device [BS 7405, 1991] working on the same principle as a venturi type meter [Miner, 1956], was used for flow rate measurements. The Dall Tube was installed at 9.4m downstream the 90° bend and 2.5m upstream the turbine intake, i.e. 31 and 10 pipe diameters respectively[2]. The pressure across the Dall tube was measured with a pressure differential transducer[3,4].

A pressure transducer was used to measure the net head across the turbine. The transducer was mounted onto a pressure tapping ring as it is recommended by IEC [1991] and BS EN 60041 [1994]. The pressure tapping ring was positioned at 2 pipe diameters upstream the spiral casing intake, [BS 5316 : Part2, 1977].

4.2.2 Power output measurement and loading

Water turbine characteristics are obtained by tests performed over a wide range of *operating speeds* and *flow rates* under constant *head* conditions. The load on the turbine shaft can be either in *mechanical* or *electrical form.*

The use of a *mechanical load,* such as a *mechanical brake,* was considered to be impractical due to limitations imposed by the space available in the Fluids Laboratory. *Synchronous* or *induction generators* cannot be used for testing either, due to their operating speed range.

The use of a *vehicle alternator* for loading the turbine was therefore considered ideal as it allows testing to be carried out over a wide operating speed range. A belt drive was used (speed increasing 2:1 ratio) to avoid operating the alternator at low speeds.

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[2] According to ISO 5167, [1980], for a device with a diameter ratio of 0.7, the Dall Tube should be at a minimum of 28 pipe diameters from the 90° bend and 7 pipe diameters before the turbine entrance.

[3] Model PDCR 120/2WL, 175 mbar range(differential pressure), made by DRUCK-UK, mounted horizontally at the center line level of the Dall tube.

[4] Details of the calibration of the flow measuring system are available in Appendix H.

The current supplied to the field windings of the vehicle alternator was adjusted with a power supply. The adjustment of the field current across the alternator's field windings was used for fine adjustments to the turbine's operating speed. The three phase output from the alternator was fed through an AC/DC rectifier into a resistive 1kW load bank (DC light bulbs were used rated at 25V). The torque available on the turbine shaft was measured by a torque transducer[5] which was mounted between the turbine and the speed increasing pulley, see *Figure 4.1*. The turbine shaft speed was measured by an optical tachoprobe[6].

4.2.3 Test rig limitations

The current test rig set up has the following limitations :

- ❑ Measurement of flow velocity components, using for example pitot static tubes, are not possible due to limited access to the test rig.
- ❑ Static pressure measurements were only taken at the draft tube entry.
- ❑ The 25V DC light bulbs are sensitive to sudden changes in the generated voltage and the 1kW load bank limits testing to 3m of head.
- ❑ The test rig is not a closed type. Water is therefore required to be pumped through the system for a period of 20 to 30 minutes, before any flow/head measurements are made, in order to flush the air out of the system.
- ❑ No cavitation tests can be performed.
- ❑ The uncertainty of the overall efficiency[7] is ±3.22%.

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[5] Torque range 0-24 Nm, made by Westland Aerostructures Ltd., Isle of Wight, UK.

[6] Speed range 100-6000 rpm.

[7] Details of error analysis are available in Appendix H.

4.3 Turbine performance tests

The aim of this section is to present and discuss the test results. In addition, wherever necessary, reference is made to other sections of this chapter, where the evaluation of the turbine performance and analysis of component losses is made.

Tests were performed for a speed range of 200 rpm up to 2000 rpm and a head range of 2m to 3m, depending on the capability of the 1kW load bank to accommodate the power generated. The head across the turbine was kept constant over the above speed range i.e. *performance testing under constant head conditions*. The tailwater level was maintained constant so that the turbine inlet was set at 1.3m above the tailwater. Unit values were used in order to assist plotting the test results as a single curve. Values of N and H_N are normalized wherever necessary using the affinity laws. Test results are presented in the following manner :

- ❑ Selected test results of N versus Q, P_{OUT} and η_O.

- ❑ Unit values of $\dfrac{N}{\sqrt{H_N}}$ versus $\dfrac{P_{OUT}}{H_N^{3/2}}$, $\dfrac{T}{H_N}$, $\dfrac{Q}{\sqrt{H_N}}$ and η_O.

Two runners were tested with the following features :

- ❑ *Runner 1*, fitted with fabricated blades, tested for a surface finish[8] of 10μm Ra roughness. This surface finish is normally obtained when fabrication is used.

  Runner 2, fitted with blades made on a milling machine, tested for two different surface finishes, 10μm and 1.4μm Ra roughness. (rough and smooth surface finish respectively).

[8] The surface relative average (Ra) roughness of the turbine blades was determined using the Electro-Formed surface Roughness Comparison Standard Tactile Testing.

4.3.1 Performance characteristics of runner 1

Table 4.1 shows the performance characteristics for *runner 1* tested at 2, 2.5, 3 and 4m. The efficiency curves[9] for *runner 1* are shown by *Figure 4.3*.

H_N	Q	N	P_{out}	$\eta_{O(bep)}$ (%)
2	0.047	899	374	40.6
2.5	0.052	1050	551	43.2
3	0.055	1116	708	43.7
4	0.058	1113	1051	46.2

Table 4.1 : Best efficiency characteristics for runner 1 (normalized values).

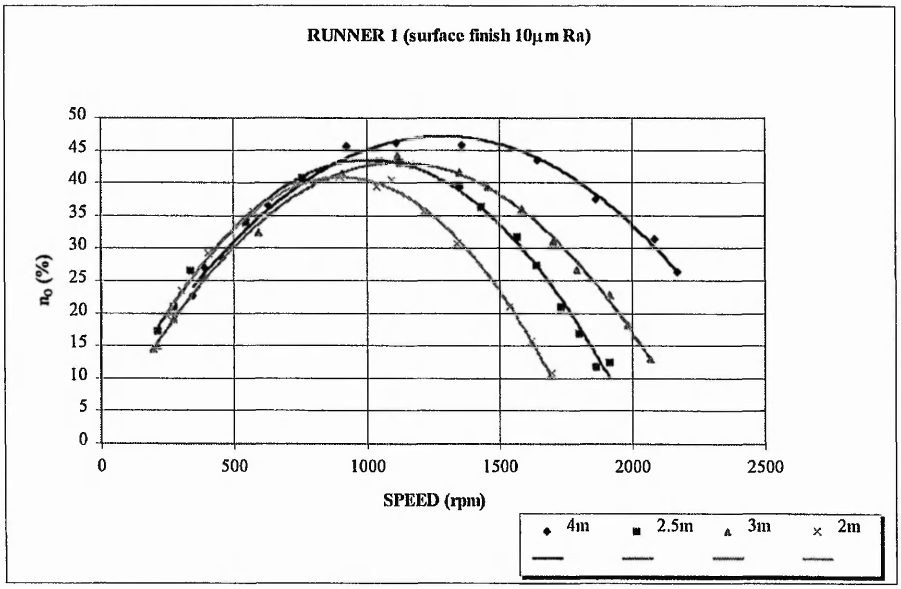


Figure 4.3 : Efficiency test results for runner 1.

The turbine was designed for a best efficiency point (bep) at 1560rpm. However, the test results presented in *Table 4.1*, show a drop in speed of 33%, or 600rpm, at the bep.

[9] Curves were drawn with Microsoft Excel, using a polynomial trend line fitting.

Moreover, the efficiency of the fabricated runner was reduced by 30% of the expected value of 70% at 2m of head. As the head across the turbine increases, an increase is observed for both the maximum efficiency and the speed for the bep. Test results obtained at 4m of head show an improvement in turbine speed and efficiency by 20% and 6% respectively.

The shape of the efficiency curve is very interesting. It would normally be expected that a fixed geometry propeller turbine would have a very pointed efficiency curve and would be very sensitive to changes of flow rate. On the contrary, the performance of *runner 1* shows a rather flatter shape, when compared with the performance of *runner 2* for example presented in section 4.3.2, which is geometrically more accurate than *runner 1*. The reason for this change in the performance characteristics of *runner 1* can only be the result of the shape of the fabricated blades. As it has been already described in Chapter 3, section 3.4.5, the fabricated blades appear to have an S-shape with the leading and trailing edges very different to what the hydraulic design of the blades produced. The power output and flow rate characteristics are shown by *Figure 4.4* and *Figure 4.5* respectively.

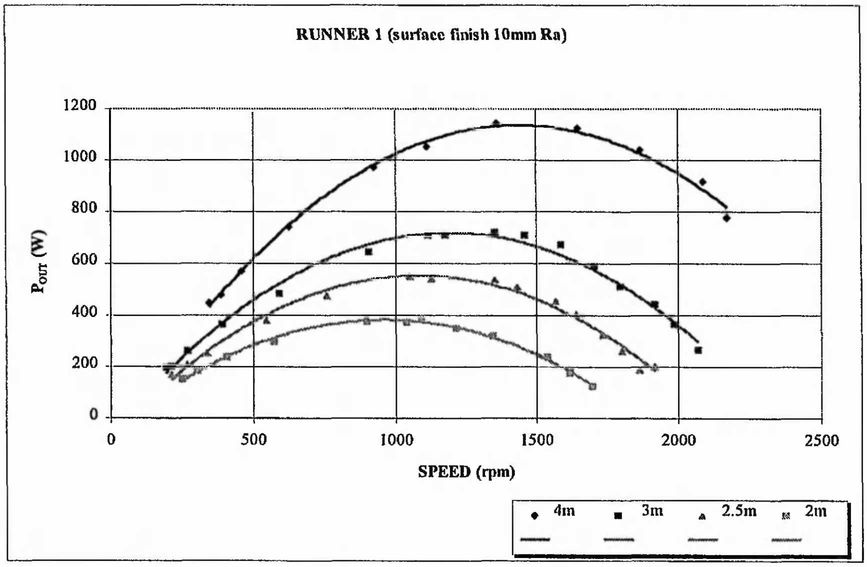


Figure 4.4 : Power output test result for runner 1.

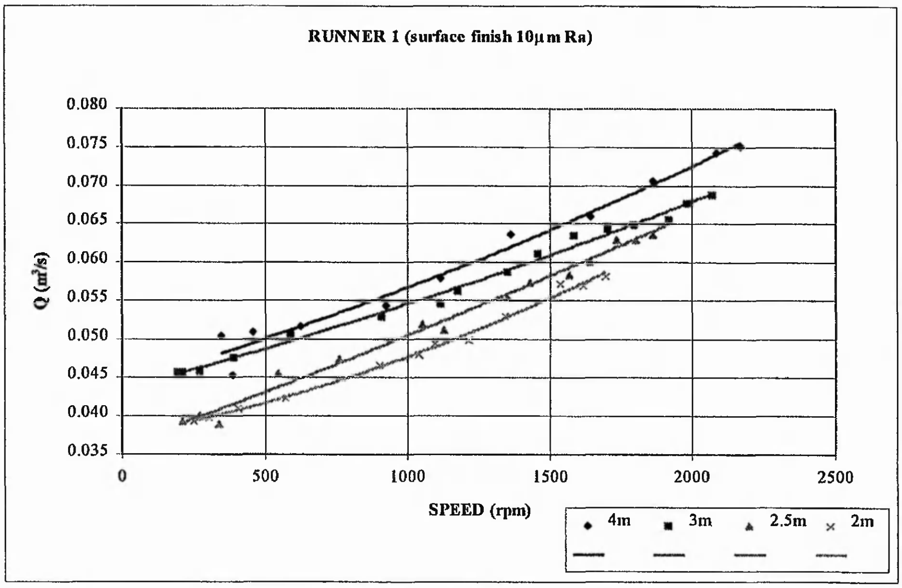


Figure 4.5 : Flow rate test results for runner 1.

Figure 4.6 shows the *unit torque* and *power output* test results versus *unit speed* and *Figure 4.7* shows the *efficiency* test results versus *unit speed.*

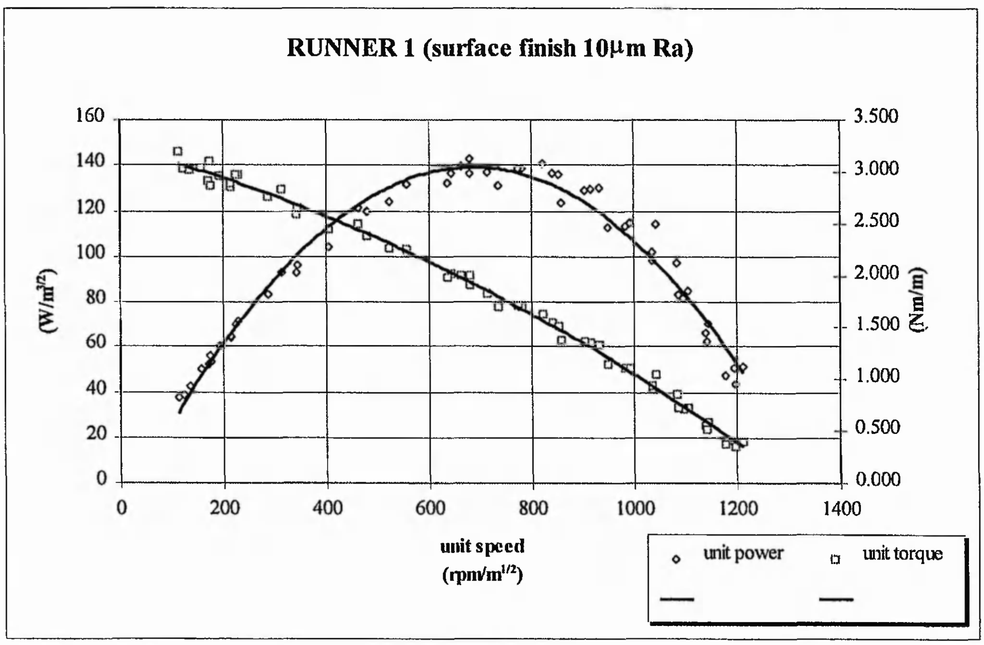


Figure 4.6 : Runner 1 - unit torque and power output versus unit speed (10μm Ra).

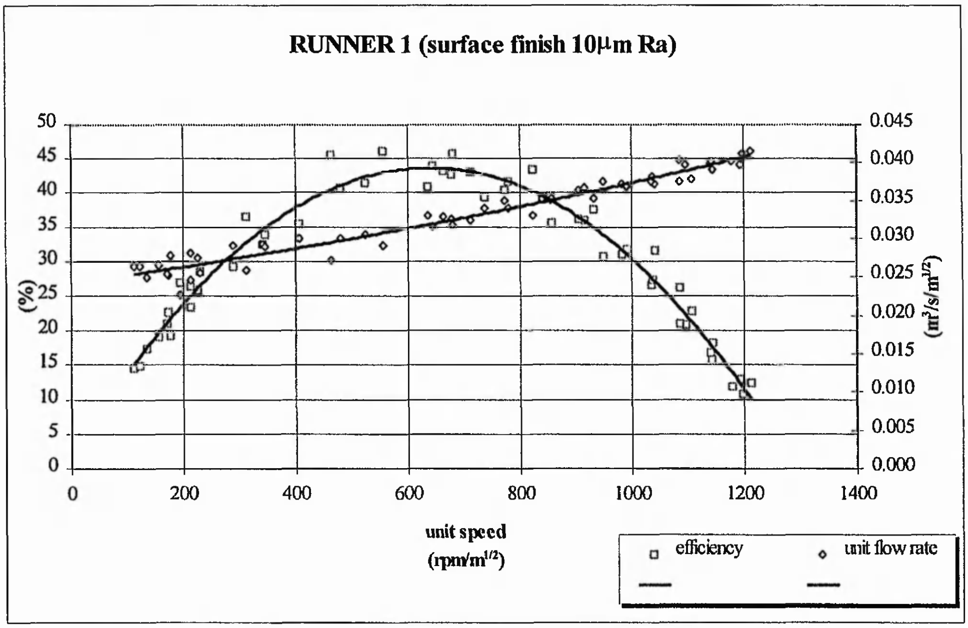


Figure 4.7 : Runner 1 - efficiency and unit flow rate versus unit speed (10μm Ra).

4.3.2 Performance characteristics of runner 2

Table 4.2 shows selected normalized test results for *runner 2*.

Roughness	H_N	Q	N	P_{out}	$\eta_{o(bep)}$(%)
10μm Ra	3	0.056	944	887	53.8
10μm Ra	2.5	0.050	789	654	53.3
10μm Ra	2	0.049	922	470	48.9
1.4μm Ra	3	0.059	922	966	55.6
1.4μm Ra	2.8	0.057	919	860	54.9
1.4μm Ra	2.6	0.058	1076	809	54.7
1.4μm Ra	2.5	0.054	958	725	54.7
1.4μm Ra	2.4	0.052	870	672	54.7
1.4μm Ra	2.2	0.048	936	556	53.7
1.4μm Ra	2	0.048	953	500	53.2

Table 4.2 : Best efficiency characteristics for runner 2 (normalized values).

The turbine performance for *runner 2* shows a considerable improvement over the performance of *runner 1* which was a result of the increased geometrical accuracy of the runner blades. A performance comparison between *runner 1* and *runner 2* is presented in section 4.3.3. The efficiency curves for tests carried out with blade surface finishes of 10μm and 1.4μm Ra roughness, are shown by *Figure 4.8* and *Figure 4.9* respectively.

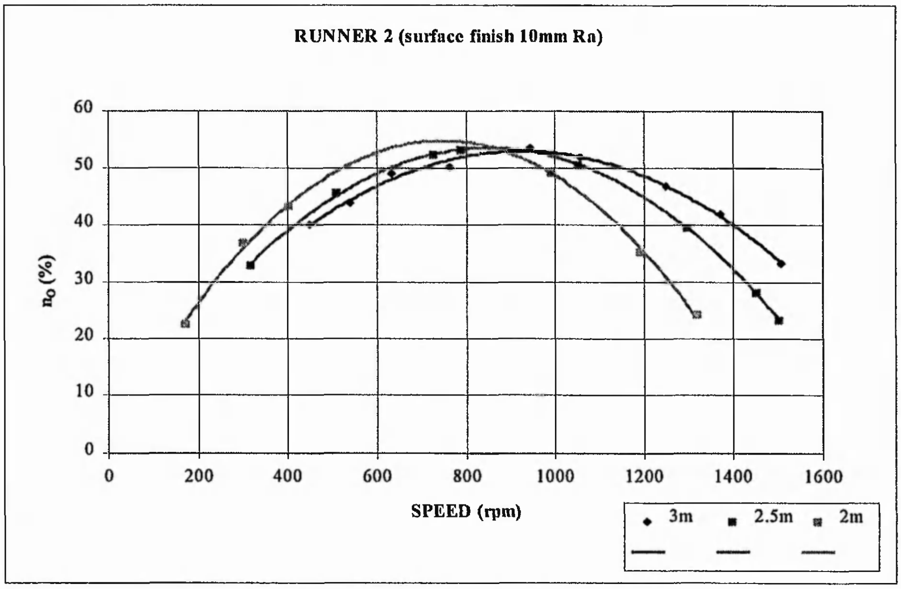


Figure 4.8 : Efficiency test results for runner 2 with rough surface finish.

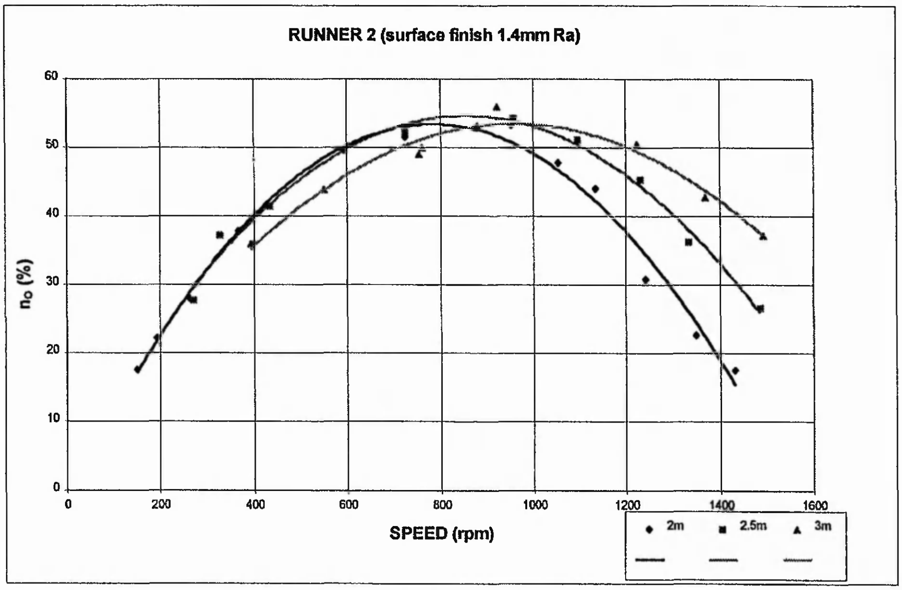


Figure 4.9 : Efficiency test results for runner 2 with smooth surface finish.

An improvement in performance is also observed when the surface finish of the runner blades in improved. Such a comparison is shown by *Figure 4.10* and *Figure 4.11* for a turbine head of 2 and 2.5m respectively.

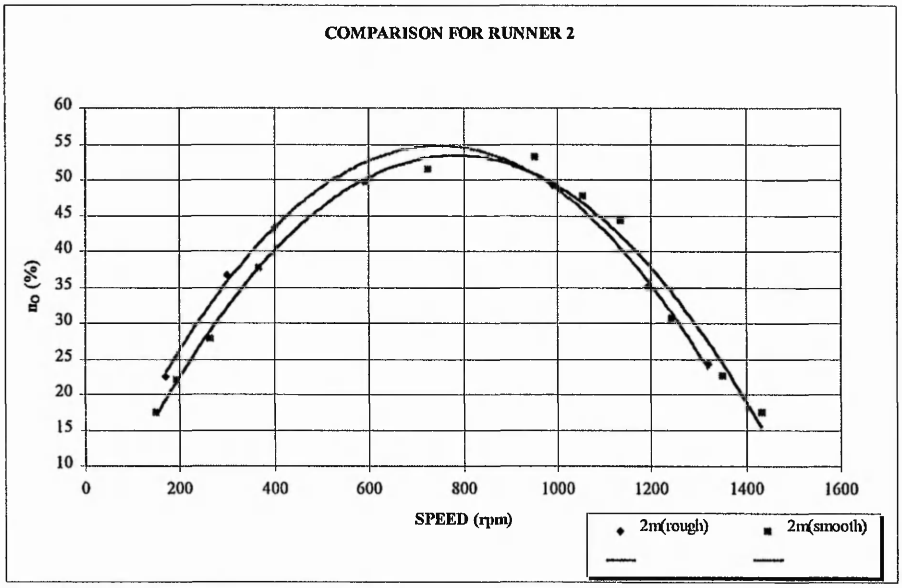


Figure 4.10 : Comparison of efficiency test result for runner 2 for a net head of 2m.

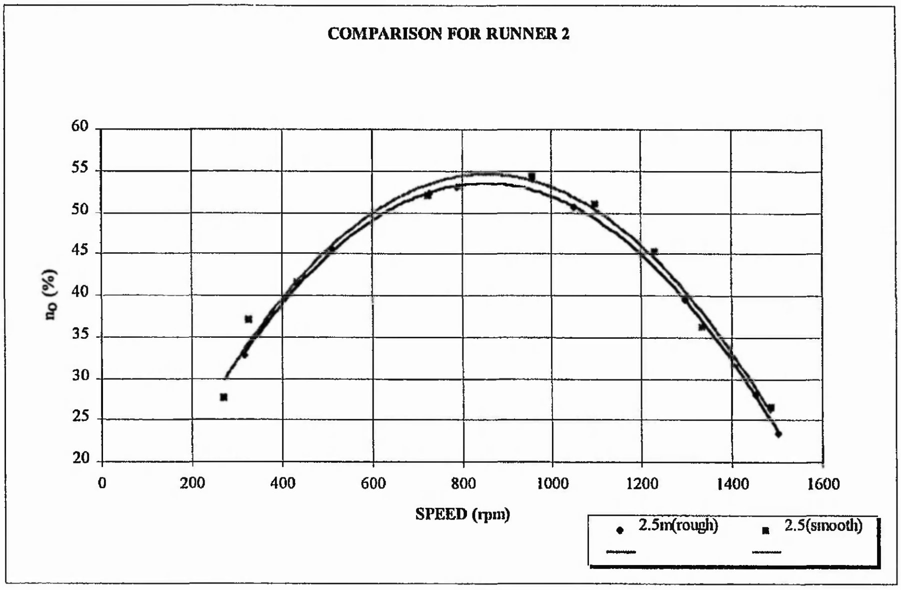


Figure 4.11 : Comparison of efficiency test result for runner 2 for a net head of 2.5m.

The power output characteristics of *runner 2* for a blade surface finish of 10μm Ra and 1.4μm Ra are presented by *Figure 4.13* and *Figure 4.13* respectively

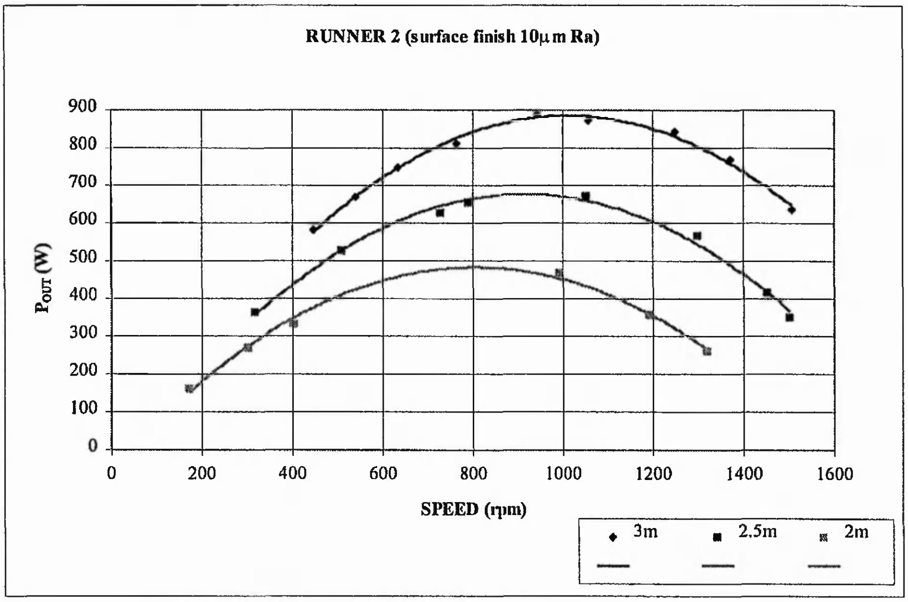


Figure 4.12 : Power output versus speed for rough surface finish.

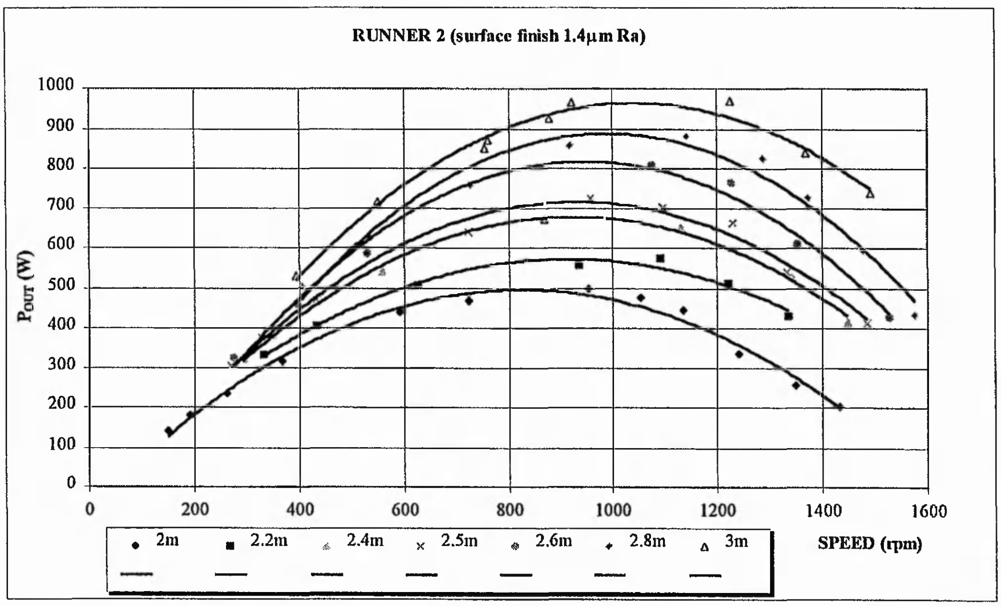


Figure 4.13 : Power output versus speed for smooth surface finish.

Tests were performed for a maximum head of 3m due to the improved performance of *runner 2* and the limited capacity of the 1kW load bank.

The *unit torque, flow rate, efficiency* and *power output* characteristics versus *unit speed* for *runner 2* are shown in the next few pages.

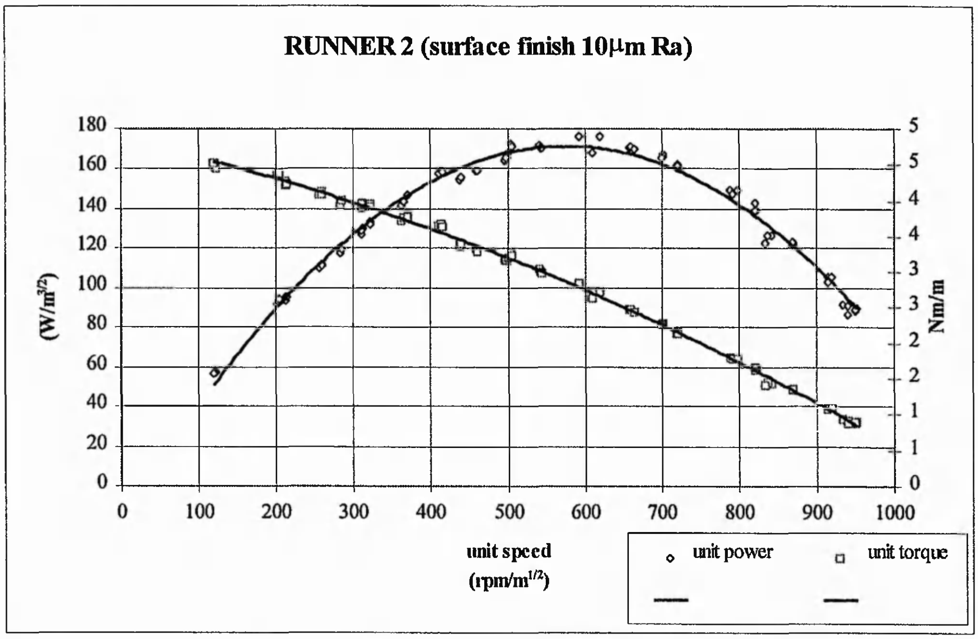


Figure 4.14 : Runner 2 - unit torque and power output versus unit speed (10μm Ra).

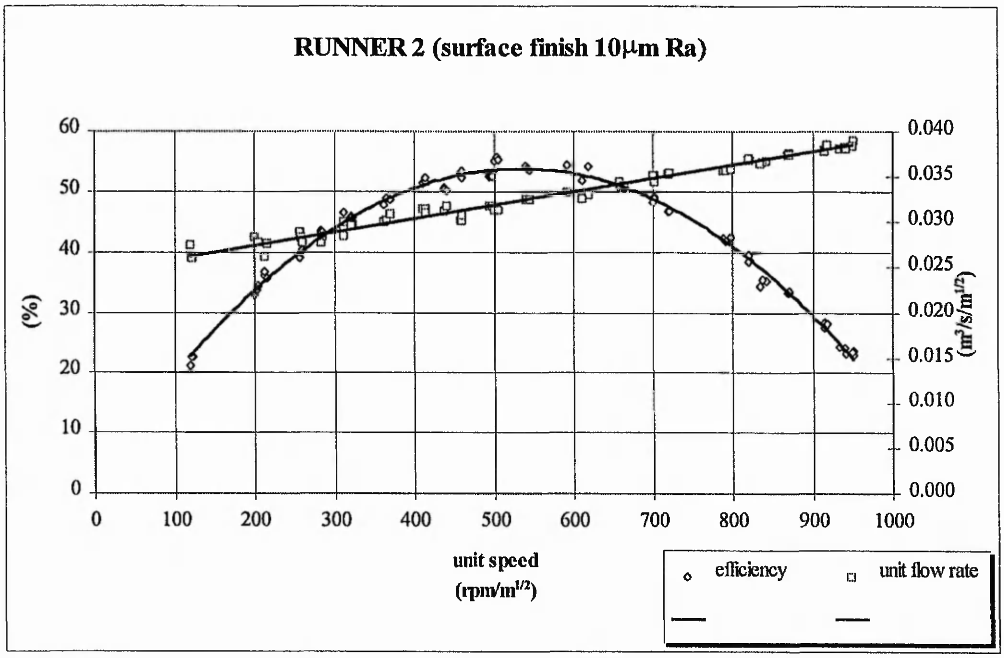


Figure 4.15 : Runner 2 - efficiency unit and unit flow rate versus unit speed (10μm Ra).

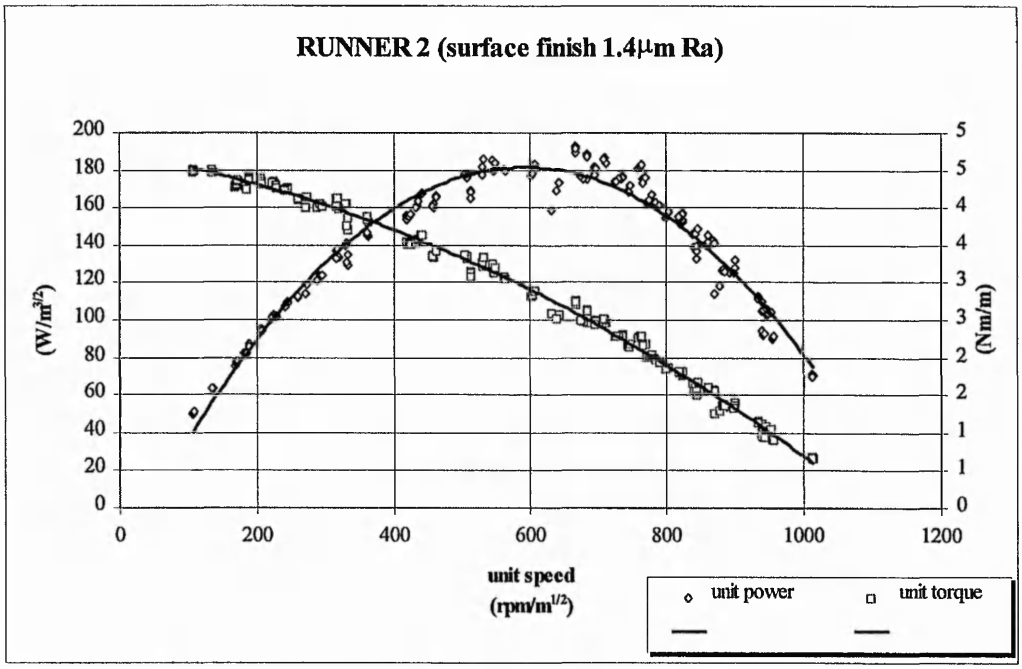


Figure 4.16 : Runner 2 - unit torque and power output versus unit speed (1.4μm Ra).

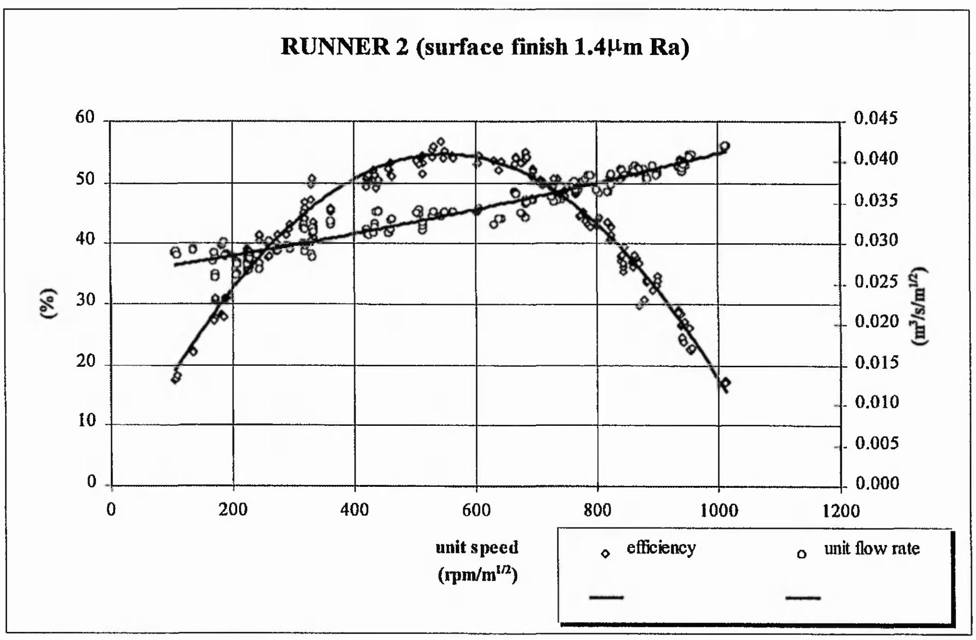


Figure 4.17 : Runner 2 - efficiency and unit flow rate versus unit speed (1.4μm Ra).

4.3.3 Comparison between runner 1 and runner 2

Test results for *runner 2* show a considerable improvement in turbine efficiency compared to those obtained for *runner 1*. The efficiency improvement for *runner 2* is 8.3% and 11.9% at 2 and 3m of head respectively. The comparison of the performance characteristics for the two runners is shown by *Figure 4.18*.

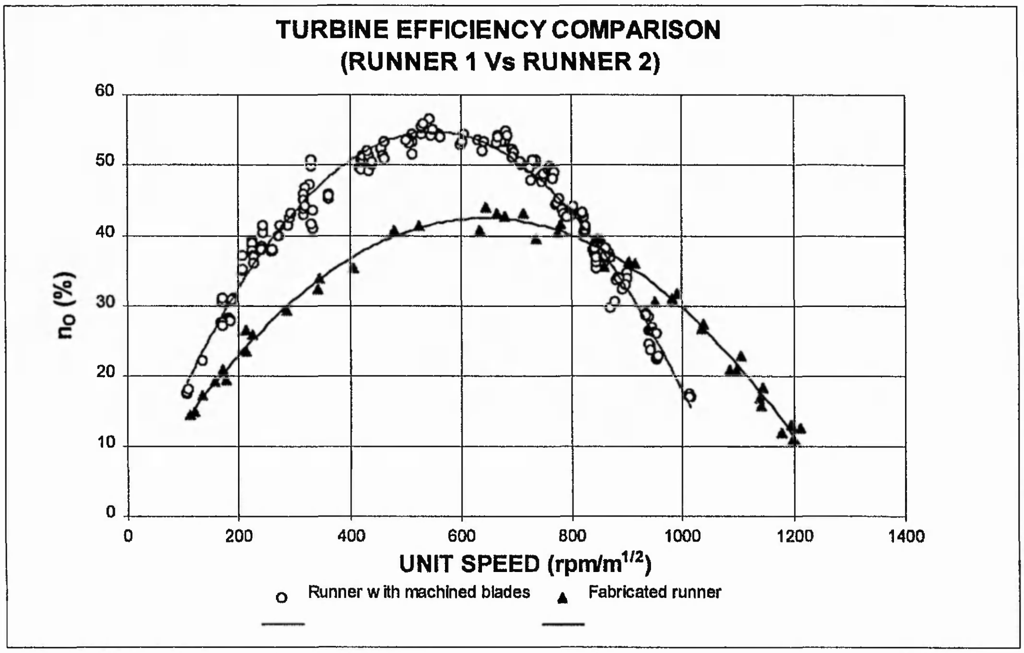


Figure 4.18 : Efficiency comparison between the fabricated runner and the one with machined blades.

As it is clearly demonstrated by the efficiency curves presented by *Figure 4.18*, the performance of *runner 2* is more sensitive to speed and therefore flow rate changes to that of *runner 1*. The efficiency curve for *runner 2* is what would normally be expected from a fixed geometry machine.

An improvement is also observed by the flow rate capacity of *runner 2* shown by *Figure 4.19*.

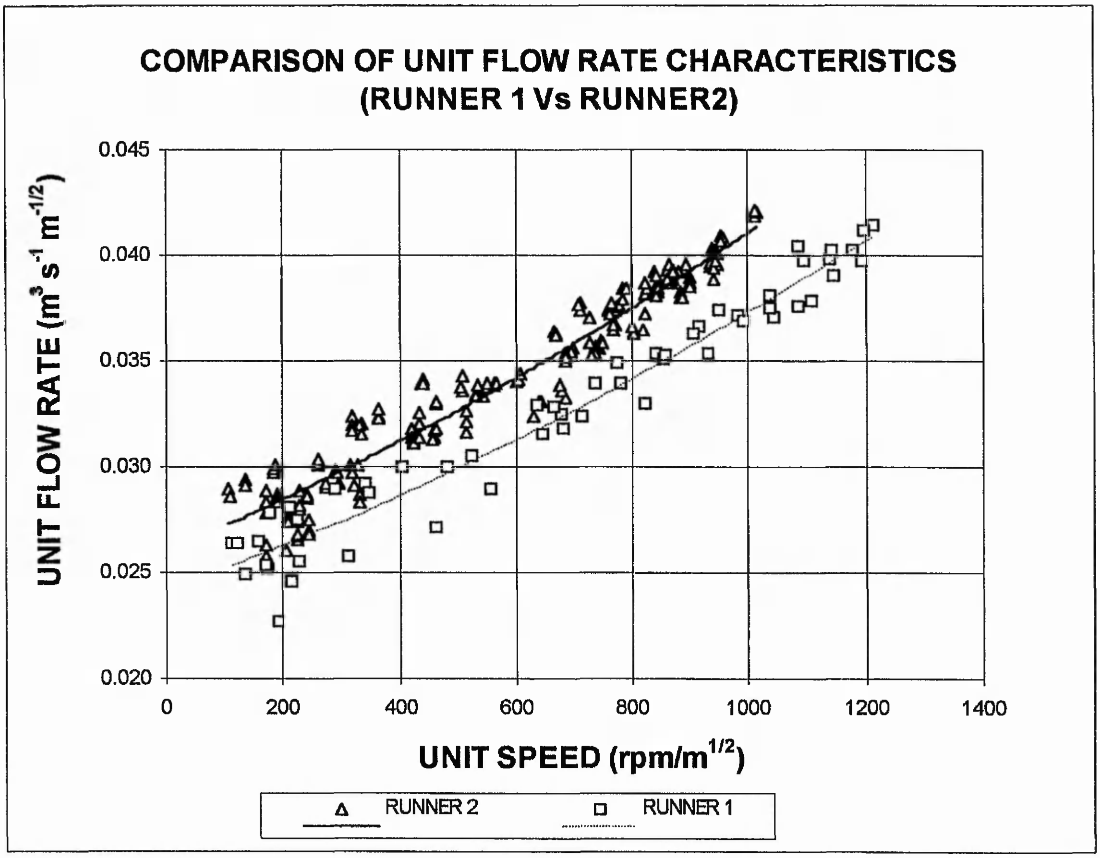


Figure 4.19 : Flow rate versus speed.

With reference to the test results and the performance characteristics of the *runner1* and *runner 2*, the following are concluded :

- The geometrical accuracy of the runner blades has a significant effect on the overall turbine performance.
- The amount of twist from hub to tip has to be kept to a minimum when runner blades are fabricated in order to maintain geometrical accuracy.
- Runner blades with smooth surface finish perform better than the ones with a rough surface finish.
- The current prototype turbine would be suitable for direct coupling to a 6 pole rather than a 4 pole induction generator.

4.4 Nature of real fluid effects and loss analysis

Turbomachinery flows are among the most complex flows encountered in fluid dynamic practice. Both the geometry of the fluid flow domain and the physical processes present are extremely complicated. Turbomachinery flows are always three dimensional, viscous and unsteady. Regions of laminar, transitional, and turbulent flows, separated flows, and fully developed viscous profiles may all be present simultaneously due to the multiplicity of length scales introduced by the complicated geometry of the flow field. The equations describing the flow physics are found in the textbooks by Lakshminarayana [1996] and Vavra [1960] and published papers [Denton, 1993 and Vu, 1986]. The analysis of fluid flow based on these methods requires complex numerical methods that are used mainly by large turbine manufacturers who have much greater resources for investigating such phenomena.

The approach used in this section adopts a simpler way of looking at the flow through the propeller turbine unit. This method is capable of calculating hydraulic losses, in more detail than in the original design methodology did, as a result of which the overall efficiency of the turbine was lower than the expected value of 70%. The analysis resorts to the established relations for pipe friction losses, losses through diffusing sections, boundary layer theory and the flow over flat plates, pressure distribution and characteristics of aerofoil sections. With reference to *Figure 4.20* the present analysis would investigate the hydraulic head losses as follows :

- ❑ *Intake pipe.*
- ❑ *Spiral casing.*
- ❑ *Guide vane area.*
- ❑ *Turbine runner.*
- ❑ *Draft tube.*

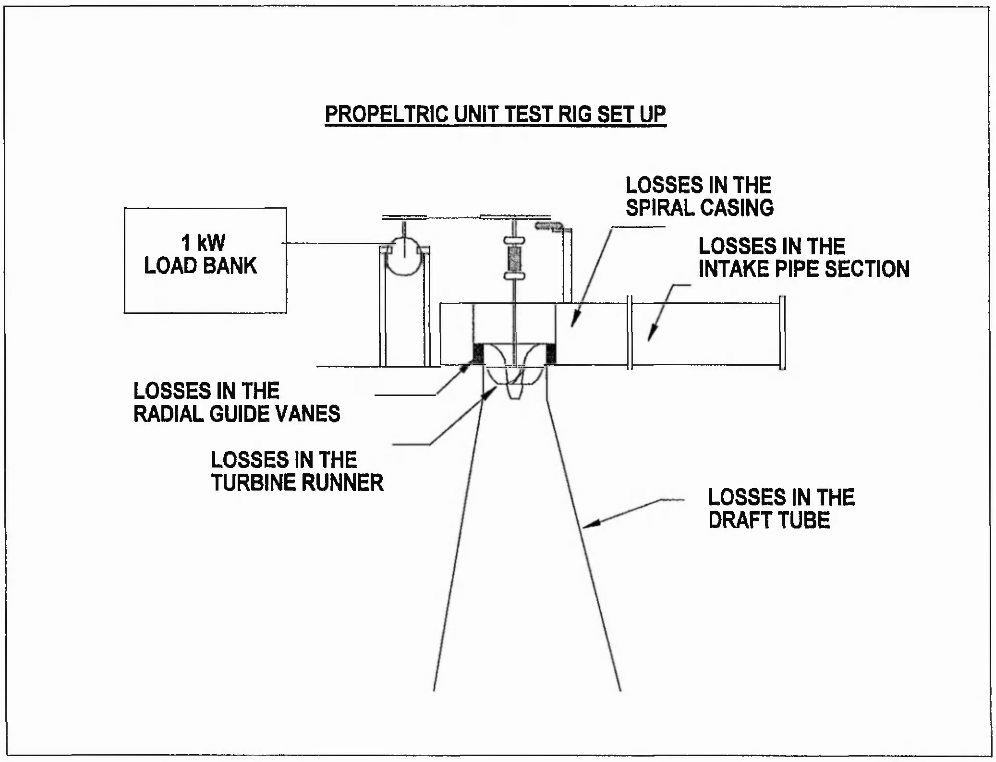


Figure 4.20 : Areas where losses are considered.

Test results obtained at 2m of head for *runner 2* are used for the current analysis. *Runner 1* is not considered in this section due to the incorrect geometry of the runner blades. An evaluation of the design method used and design alterations to assist fabrication are presented in section 4.5.

4.4.1 Fluid flow and loss analysis for the spiral casing

The understanding and prediction of the flow in a water turbine spiral casing is very important. Until recently, the design of spiral casings was based on numerous assumptions and design models that have been tested in fluid laboratories [Fitero, 1972 and Gubin, 1975]. However, with the widespread use of numerical methods and the use of powerful computers that enable simulation of the flow through the complex three dimensional geometry of the spiral casing [Bruttin, 1996; Désy, 1987 and Song, 1995], the losses and performance can be predicted and optimized very accurately.

Either method requires skills and equipment not available in the small workshops of developing countries. A modified spiral casing shape was used instead, shown by *Figure 421,* to assist fabrication.

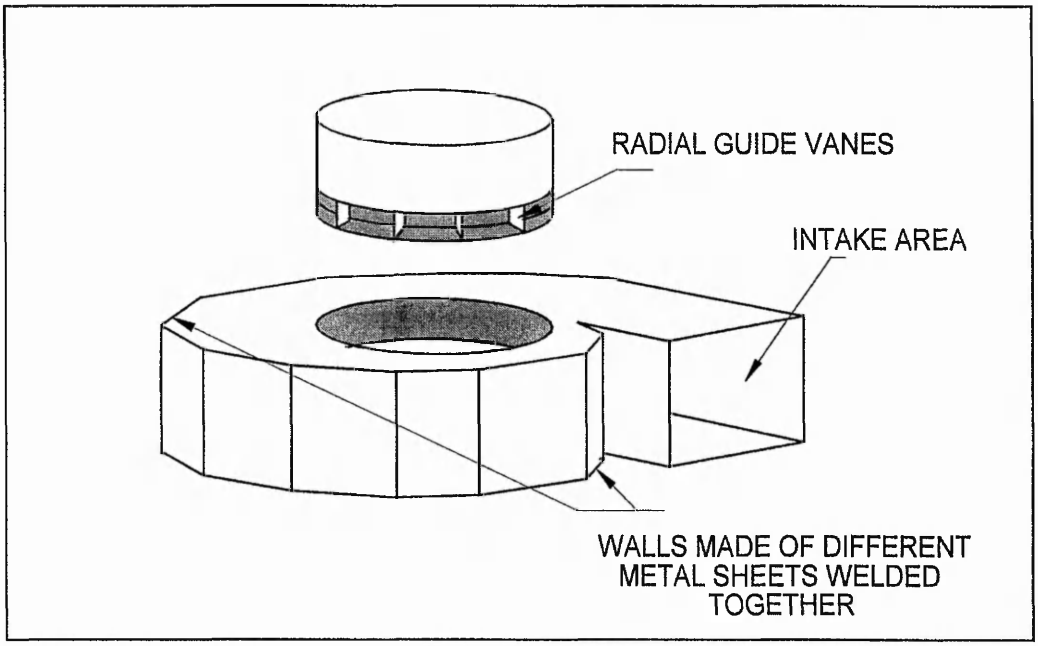


Figure 421 : The fabricated spiral casing.

As a result of simplified flow surfaces with sharp edges and square sections performance was therefore sacrificed for low cost solutions. The analysis of the hydraulic losses for the intake system aims at establishing the effect of the current design layout on the overall performance of this prototype turbine, based on the test data, as follows :

- ❑ Determine the friction losses in the casing based on the resistance laws of flow through pipes.
- ❑ Calculate the head loss due to the reducing area between the spiral casing and the guide vanes.
- ❑ Calculate the velocity distribution in the spiral casing and compare this to the original design assumptions.

4.4.1.1 Friction losses in the spiral casing

With reference to *Figure 4.1,* the head across the turbine is measured by a pressure transducer which is located 0.5m from the intake of the casing. Head losses would therefore include those losses leading to the casing and the casing itself. Frictional losses through the guide vanes were not considered mainly due to their short length.

The head loss due to friction in the pipe work leading to the spiral casing, ξ_p, is given by *Table 4.3*. These losses are very small in value, less than 1% of the net head available. When a short penstock, less than 2m in length, is used for feeding the water into the spiral casing, these losses can therefore be ignored.

Diameter of pipe, D_P	0.25m	Length of pipe, L_P	0.5m
Flow rate, Q	$0.048 m^3/s$	Average velocity, V_P	0.978 m/s
Roughness of cast iron, k	0.25mm	Reynolds number, Re	2.45×10^5
Friction coefficient, f	0.005	$\xi_P = 4 \frac{f_P \, L_P}{D_P} \frac{V_P^2}{2g}$	0.0022m or 0.11%

Table 4.3 : Head loss in the pipe work leading to the casing.

The losses in the spiral casing itself are calculated and presented in *Table 4.4*.

Section width, X_{sp}	0.24m	Length of casing, L_{sp}	0.5m
Section height, Y_{sp}	0.24m	Hydraulic mean depth, m_{sp}	0.06 m
Flow rate, Q	$0.048 m^3/s$	Average velocity, V_{sp}	0.833 m/s
Relative Roughness,	0.625×10^{-3}	Reynolds number, Re	4.16×10^5
Friction coefficient, f	0.005	$\xi_{sp} = \frac{f_{sp} \, L_{sp}}{m_{sp}} \frac{V_{sp}^2}{2g}$	0.0015m or 0.075%

Table 4.4 : Head loss in the spiral casing.

A similar result is obtained using the work of Osterwalder [1972]. The method makes use of parameters related to the geometry of the casing as follows :

$$\frac{\xi_{sp}}{H_N} = \left\{\frac{V_{av}}{\sqrt{2gH_N}}\right\}^2 f\pi\left[\frac{4\,\varepsilon\,D_T}{D_{sp}}+1\right] = 787.4\times10^{-6}\ \text{or}\ \xi_{sp} = 0.0016m \qquad \{4.1\}$$

where $\varepsilon = \dfrac{\text{Guide vane radius (outside)}}{\text{Turbine tip diameter}}$, D_{sp} = *spiral casing inlet diameter*,

$V_{av} = \dfrac{Q}{\text{area at inlet of casing}}$ and f = *friction coefficient taken as above*

4.4.1.2 Losses due to area reduction between casing and guide vanes

The analysis presented here is based on the hydraulic losses due to reducing sections. With reference to *Figure 4.22*, the casing is considered to have been unfolded and the through flow is examined as it would have been for a duct with a sudden contraction.

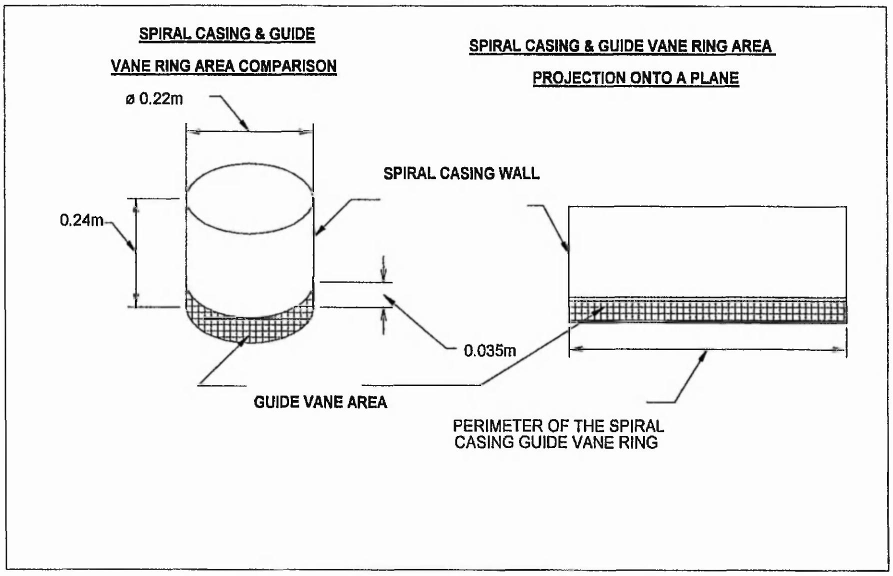


Figure 4.22 : Guide vane area reduction loss.

The equivalent area of a circular duct is considered and therefore standard pipe loss data presented by Massey [1986] are used. The ratio of the equivalent areas, shown by *Figure 4.22*, produces a diameter ratio of 0.382.

The loss coefficient ζ_g for this diameter ratio is 0.38. The head loss due to the area change is calculated as follows :

$$\xi_g = \zeta_g \frac{V_{go}^2}{2g} = 0.12m \qquad \{4.2\}$$

4.4.1.3 Velocity distribution in the spiral casing

Figure 4.23 shows the velocity distribution of the incoming flow that was originally assumed for the design of the spiral casing. The dimensions of each section were determined by considering constant average velocity in the spiral casing. The law of constant angular momentum was not used for the initial casing design.

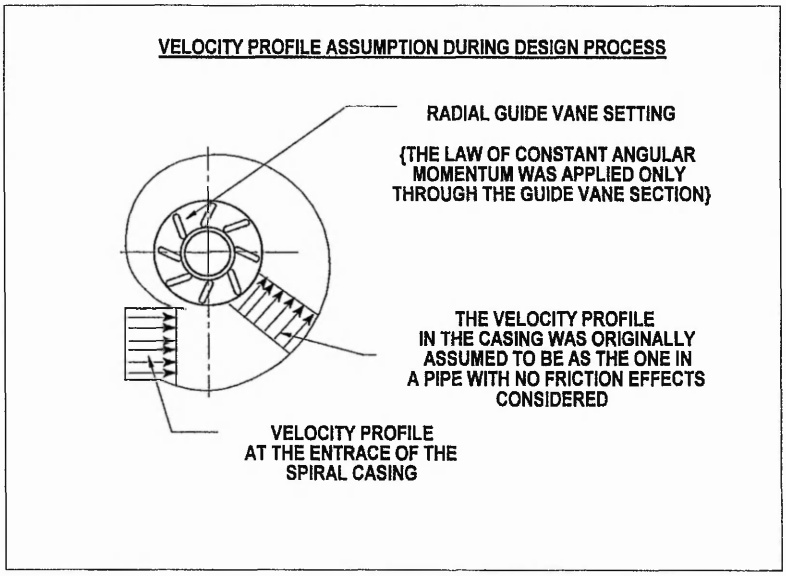


Figure 4.23 : Assumed velocity distribution in the spiral casing.

As a result of the simplified design assumptions for the spiral casing, the runner blades were approached by greater tangential velocity components to those produced by the original design calculations. The mismatch between the flow distribution in the spiral casing and the required velocity components for the turbine runner led to the increase of hydraulic losses. Flow observations at the draft tube exit, described in section 4.4.5.5, identified eddies in the water. These eddies were the result of increased whirl at the exit of the runner blades when operating at speeds close to the design speed of 1560rpm.

The reduction of the turbine speed for the bep was a further indication of mismatch between the flow in the casing and the design flow geometry. The method described below provides a simple and yet reliable way to determine the size of any spiral casing and flow distribution to the runner blades. The method is based on the work of Audisio [1991] and Ntoko [1985] as well the methods described by Eck [1973] and Lazarkiewicz [1965].

The incoming flow into the casing, whose dimensions are known, provides information of the average velocity at any section of the casing. The law of constant angular momentum at any section of the casing is therefore used to determine the exact position, i.e. radius, that this average velocity is situated. Provided the radius is known, the tangential component of the flow and the velocity distribution in the runner blades are therefore determined. With reference to *Figure 4.24*, applying the law of constant momentum at any section of the casing, the velocity of flow is given by

$$V = \frac{V_{ugo} R_{go}}{R} \qquad \{4.3\}$$

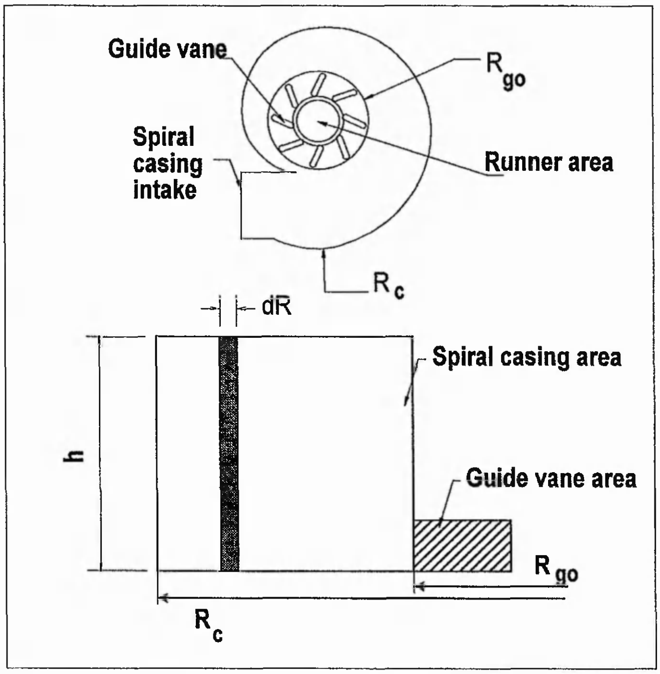


Figure 4.24 : Details for the velocity profile calculation in the spiral casing.

Taking a small element of a section in the casing, as shown by *Figure 4.24*, and integrating between R_{go} and R_c, then

$$Q = \int V\, d(area) = \int_{R_{go}}^{R_c} V\, h\, dR = hV_{ugo}R_{go}\left[\ln R_c - \ln R_{go}\right] \qquad \{4.4\}$$

Assuming that at any point in the casing V is the same as the V_u component and using equations {4.3} and {4.4} the following expression is derived

$$\frac{V_{ugo}}{V_u} = \frac{Q}{hV_u\, R_{go}} \frac{1}{\left[\ln R_c - \ln R_{go}\right]} \qquad \{4.5\}$$

So in order to find the location of the average velocity in the spiral casing i.e. the radius R of the average velocity, equation {4.5} is substituted in equation {4.3}. With the aid of the above analysis, the tangential component of the absolute velocity is therefore calculated.

Table 4.5 shows the results for a flow rate of 0.048 m^3/s at 2m of head.

Q	0.048m^3/s	V_{ugo}	1.57m/s	V_{ugi}	2.16m/s	V_{u1T}	2.30m/s
V_{av}	0.833m/s	V_{ago}	1.98m/s	V_{agi}	2.73m/s	V_{a1T}	3.23m/s
$\frac{V_{ugo}}{V_{av}}$	1.89m	V_{go}	2.53m/s	V_{gi}	3.4m/s	V_{u1H}	5.76m/s
R_{av}	0.21m	α_{go}	51.63°	α_{gi}	51.63°	V_{a1H}	3.23m/s

Table 4.5 : Velocity distribution in the spiral casing and turbine runner.

Based on the findings of this analysis, the flow distribution in the spiral casing is shown by *Figure 4.25*.

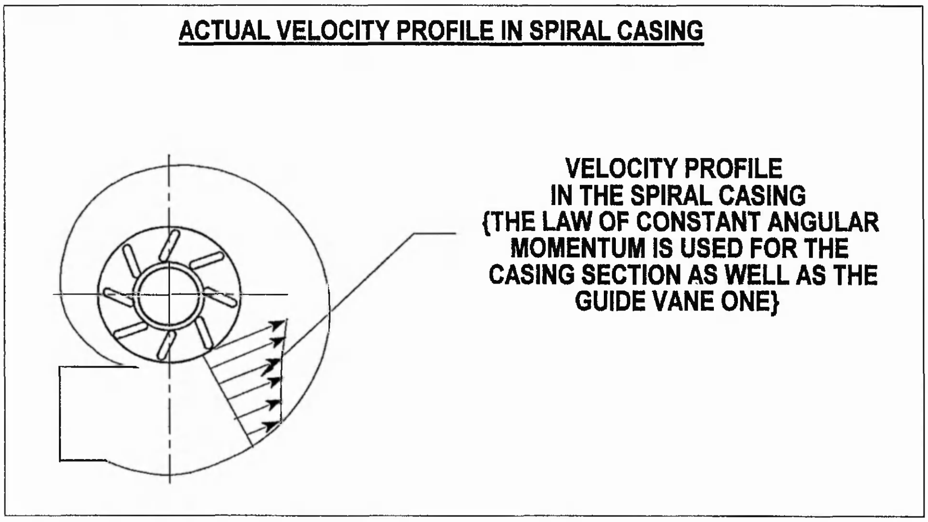


Figure 4.25 : Actual velocity distribution in the spiral casing.

The conservation of angular momentum applies to frictionless motion in a space where there are no forces acting on the fluid in the tangential direction, i.e. the guide vanes are in the direction of flow and cause no momentum loss. The use of the method presented in this section enables the designer to calculate the characteristic angle of the incoming flow into the guide vane area. Based on the calculated data presented by *Table 4.5*, the following are concluded :

- ❑ Test results show that the original assumptions for the flow distribution assumption in the spiral casing were incorrect. A modified spiral casing design to match the characteristics of the present runner is presented in section 4.5.1.

- ❑ The incoming flow is approaching the guide vanes, which are set at 75°, at an angle[10] of 51.7°. This comparison pinpoints to momentum losses as a result of which the angular momentum into the runner blades would be different to what is presented by *Table 4.5*. A method to calculate this loss is introduced and explained in section 4.4.2.

[10] This angle is measured between the absolute velocity and its tangential component.

4.4.2 Fluid flow though the guide vanes

One of the biggest problems that the guide vane system presents to the turbine designer is the uneven distribution of the whirl component along the length of the vane and consequently through the runner and towards the draft tube. Since the guide vanes have a constant discharge angle, the whirl component of the flow velocity is different from tip to hub. According to work of Gokhman [1996], this causes the absolute flow into the runner to be unsteady. Thus the design conditions are ambiguous - does one design for the guide vane to aim at the mid section of the runner blades or some other location ?

Modified mathematical models for the guide vane design are available. However, such systems require complicated hydrodynamic design approaches for the guide vane profiles which often use twisted sections that allow even distribution of whirl along the blade. The investigation of these parameters is beyond the scope of the current research.

The aim of this section is the calculation of the angular momentum of the absolute velocity at the exit of the guide vanes. To do so, the guide vane arrangement is treated as a cascade of stationary blades. As it is shown by *Figure 4.26*, the flow into the guide vane entrance has an incidence angle of 23.3°. The lift coefficient for a cascade composed by flat plates, using the Weinig coefficient κ [Turton, 1984 and Wislicenus, 1965] is found by

$$C_L = 2\pi\, \kappa\, \sin \alpha_{g\infty} \qquad \{4.6\}$$

The value of the lift coefficient is calculated in the direction of the average velocity through the cascade, the angle of which is given by

$$\alpha_{g\infty} = \frac{\alpha_{go} + \alpha_{gi}}{2} \qquad \{4.7\}$$

The cascade lift coefficient for the guide vane arrangement is equal to 1.08.

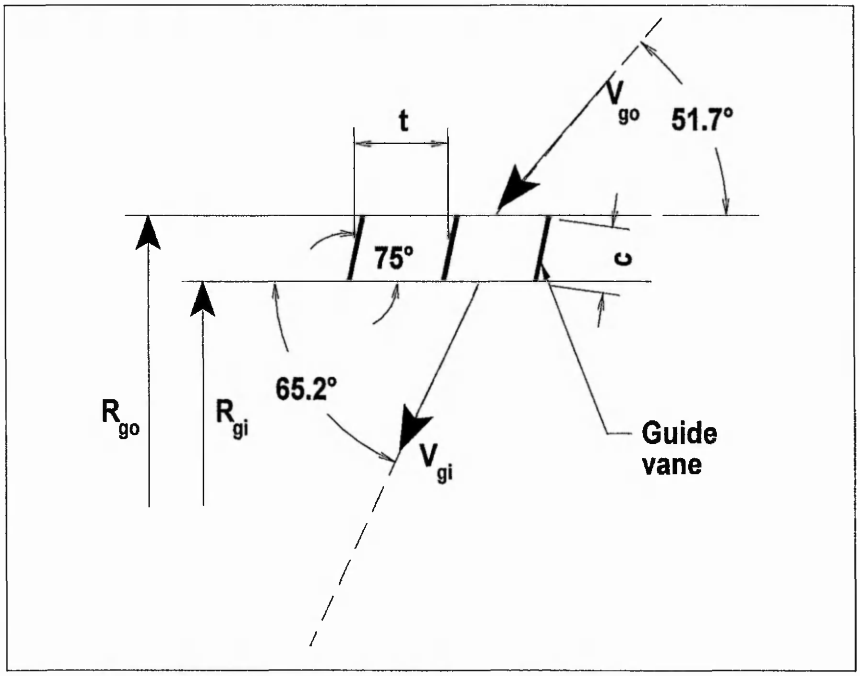


Figure 4.26 : Calculating the momentum loss in the guide vanes.

The change in momentum is calculated using the analysis, presented in Chapter 3, for the lift coefficient for a cascade of blades. Since the blades are stationary the absolute velocity components are used as follows :

$$C_L = 2\left\{\frac{t}{c}\right\}\frac{V_{ugo} - V_{ugi}}{V_{g\infty}} \qquad \{4.8\}$$

Based on these calculations, the following are concluded :

- ❑ The angular momentum of the absolute velocity has decreased by 0.9m/s to a value of 1.26m/s at best efficiency, due to the incorrect setting of the guide vanes.

- ❑ Despite using guide vanes of only 30mm in length, the direction of the guide vane exit flow angle is increased to 65.2° in comparison with angle of 51.7° at the inlet of the guide vane cascade.

4.4.3 Velocity diagrams based on experimental results

In order to determine the tangential component of the angular momentum into the runner blades, the law of constant angular momentum is applied for the bladeless area between the guide vanes and the runner blades.

The Euler equation for an axial flow turbine is therefore used to determine the tangential component of the absolute velocity at the runner exit which is given by :

$$gH_E = u\left[V_{u1} - V_{u2}\right] \qquad \{4.9\}$$

The Euler head is found using the test results for the turbine overall efficiency and the calculated values of volumetric and mechanical efficiencies presented and explained in Chapter 3.

Table 4.6 shows the calculated velocity components for the hub, middle and tip blade section based on the test results of *runner 2* operating at 2m of head. The values presented by *Table 4.6* show that the direction of the absolute velocity is very close to the axial direction. The exit velocity head loss at the draft tube is therefore minimum as a result of a small exit whirl component. These losses are calculated in section 4.4.5.3. The exit whirl has a negative value because it is in the same direction as the peripheral velocity of the blade. The velocity diagrams for the hub, middle and tip section of the runner are shown by *Figure 4.27* using the values presented by *Table 4.6*.

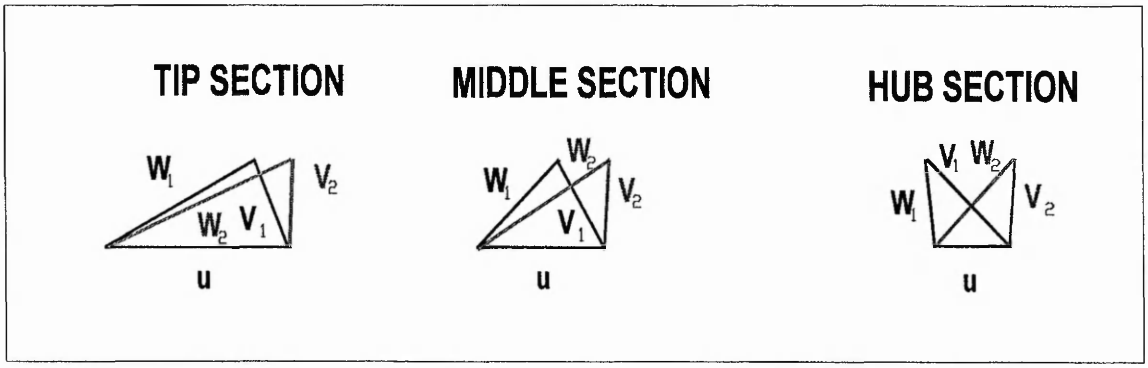


Figure 4.27 : Velocity diagrams based on experimental data at bep, 2m head.

TIP SECTION		MID SECTION		HUB SECTION	
u_T	7.48m/s	u_m	5.24m/s	u_H	2.99m/s
V_{uT1}	1.34m/s	V_{um1}	1.92m/s	V_{uH1}	3.36m/s
V_{1T}	3.50m/s	V_{1m}	3.76m/s	V_{1H}	4.66m/s
α_{1T}	67.4°	α_{1m}	59.27°	α_{1H}	43.90°
W_{uT1}	6.14m/s	W_{um1}	3.32m/s	W_{uH1}	0.36m/s
W_{1T}	6.93m/s	W_{1m}	4.63m/s	W_{1H}	3.25m/s
β_{1T}	27.76°	β_{1m}	44.21°	β_{1H}	96.36°
$\Delta\iota_T$	3.70°	$\Delta\iota_m$	8.70°	$\Delta\iota_H$	27.5°
V_{uT2}	-0.15m/s	V_{um2}	-0.21m/s	V_{uH2}	-0.37m/s
V_{2T}	3.24m/s	V_{2m}	3.24m/s	V_{2H}	3.25m/s
α_{2T}	92.68°	α_{2m}	93.79°	α_{2H}	93.47°
W_{uT2}	7.48m/s	W_{um2}	5.24m/s	W_{uH2}	2.99m/s
W_{2T}	8.15m/s	W_{2m}	6.16m/s	W_{2H}	4.41m/s
β_{2T}	23.35°	β_{2m}	31.65°	β_{2H}	47.11°

Table 4.6 : Revised flow velocity calculations based on the test results at 2m head.

Based on the same method described in section 4.4.2 and the Euler equation, the velocity diagrams at speeds other than bep can be determined. Selected values of the flow geometry using test results obtained at 593rpm and 1453rpm are shown for the middle section of the blades by *Table 4.7*.

SELECTED VALUES FOR VELOCITY DIAGRAMS

Test result obtained at 593rpm				Test results obtained at 1453rpm			
V_{um1}	1.80m/s	α_{1m}	59.30°	V_{um1}	2.37m/s	α_{1m}	59.20°
V_{um2}	-1.39m/s	α_{2m}	114.63°	V_{um2}	1.91m/s	α_{2m}	64.33°
W_{1m}	3.37m/s	β_{1m}	64.27°	W_{1m}	6.88m/s	β_{1m}	35.27°
$\Delta\iota_m$		28.7°		$\Delta\iota_m$		0.2°	
W_{2m}	4.45m/s	β_{2m}	43.93°	W_{2m}	6.08m/s	β_{2m}	33.18°

Table 4.7 : Flow geometry obtained at speeds other than the bep.

As it is clearly demonstrated by the results presented by *Table 4.7*, at speeds lower than the bep, the exit whirl is clearly in the direction of the peripheral velocity of the blades whereas a change in direction is observed at higher speeds. Moreover, the angle of incidence for the relative velocity reduces as the operating speed of the turbine approaches the design value of 1560rpm.

The conclusions drawn from this section are summarized as follows :

- ❑ The velocity diagrams for the turbine best efficiency point show that the velocity head loss at the runner exit is very small due to the fact that the absolute velocity is close to the axial direction.

- ❑ The reduced turbine performance at speeds other than the bep is justified by the velocity diagrams and the geometry of the flow. At slower speeds, the whirl exit loss is smaller than at high speed while there are increased shock losses, defined and calculated in section 4.4.4.1, due to high angles of incidence between the incoming flow and the runner blades.

4.4.4 Fluid flow in the runner cascade

Inefficiencies in turbine runners arise from losses that are caused by friction, which creates slowly moving boundary layers, the mixing of these boundary layers with the main flow, shock losses at off design conditions due to incorrect flow approach at the runner leading edge and uneven pressure distribution over the runner blades that causes separation of the flow.

Existing performance prediction methods for axial flow runners are limited to gas/steam turbine and compressor designs and very little data and information are available for hydraulic machines. Moreover, the complexity of the flow approaching the turbine runner can fully be analyzed by computational methods that are beyond the scope of the present thesis.

The aim of this section is to calculate the shock losses at the runner entry using the bep test data. Moreover, frictional losses due to boundary layer formation over flat plates are investigated. Finally, the pressure distribution over the cascade is presented using the original design data as well as the ones obtained by the performance tests.

4.4.4.1 Runner shock losses

Shock losses occur when the relative velocity approaches the runner blades at angles different to those determined by the design of a turbine cascade. With reference to *Figure 4.28*, the *shock loss* is given by

$$\xi_{shock} = \frac{W_{shock}^2}{2g} \qquad \{4.10\}$$

Using the flow geometry for the middle section[11] of the runner blades, presented in the previous section, the shock loss is equal to 0.075m or 3.75% of the available head.

[11] It is considered standard practice to perform calculations for the middle section of the runner blades.

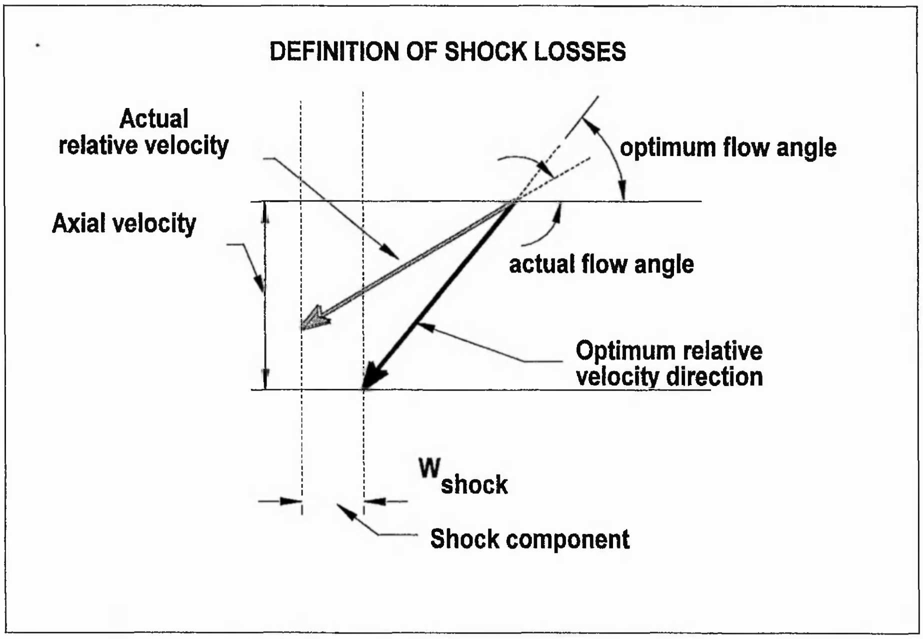


Figure 4.28 : Definition of shock losses using the relative velocity direction.

4.4.4.2 Reynolds number effects and boundary layer considerations

Investigations of losses in cascades performed over the last few decades and the refinements in boundary layer theories are used to provide an understanding of the nature of these losses and their relationship with the blade spacing and camber.

The flat plate[12] provides the most elementary vehicle for studying the relation between losses and *Re*. The exact solution of flow problems over flat plates have been considered by Trunckenbrod and are presented by Schlichting [1968]. In boundary layer theory two important parameters are considered :

- ❑ The boundary layer displacement thickness δ^* which expresses the effective displacement of the outer flow due to the boundary layer.

- ❑ The boundary layer momentum thickness ϑ which expresses the effect of the loss in momentum.

[12] The analysis presented here is for flat plates set at very small or zero incidence.

During separation the numerical value of ϑ becomes small whereas the numerical value of δ^* tends to be large [Balje, 1964]. Thus the ratio of the displacement thickness to the momentum thickness, defined as *form factor*

$$H = \frac{\delta^*}{\vartheta} \tag{4.11}$$

can be used as separation criterion. The critical value for H is about 2 for turbulent boundary layer. With reference to the work presented by Newman [1977], for a flat plate at zero incidence ϑ and δ^* are given by

$$\vartheta = \frac{0.0363 \times L}{R_e^{0.2}} \tag{4.12}$$

$$\delta^* = \frac{0.0467 \times L}{R_e^{0.2}} \tag{4.13}$$

Using equations {4.12}and {4.13} the shape factor for a flat plate is equal to 1.3. This shows that flat plates set at an incidence close to zero are not likely to experience separation of the boundary layer. The Reynolds number for the runner blades is defined as

$$R_e = \frac{[W_\infty \times c]}{\nu} \tag{4.14}$$

With reference to the experimental results at 2m of head, at best efficiency the Reynolds number for the tip section of the runner blades is 9.36×10^5 whereas for the middle section is 4.9×10^5. The boundary layer momentum thickness is therefore equal to 0.32 and 0.25mm for the tip and middle section respectively.

The relationship between cascade loss and boundary layer momentum thickness has been explained by Csanandy [1956] and very recently by Lakshmirayana [1996]. A satisfactory and simple basis for defining the loss though a cascade is given by :

$$\xi_c = \zeta_c \frac{V_a^2}{2g} \tag{4.15}$$

where ζ_c is the loss coefficient based on the through flow velocity. The value of ζ is related to the boundary layer momentum thickness and the cascade geometry by the following formula

$$\zeta_c = \frac{2\,\vartheta}{t\cos^3(\beta_2)} \qquad \{4.16\}$$

Using the design flow exit angle, the value of ζ_c becomes 0.0067 for the blade tip and 0.0095 for the blade middle section. The cascade loss is therefore calculated by equation {4.15}. The maximum loss occurs at the middle of the blade section and is equal to 0.005m.

4.4.4.3 Pressure distribution over the runner blades

The pressure distribution across the cascade of a turbomachine is very important and allows the designer to determine whether under certain conditions the pressure and/or suction side of the blades are likely to experience separation of the boundary layer. Pressure distribution also allows to identify the limits imposed by the geometry of the blades with regard to the flow incidence, camber angle and maximum camber position.

Figure 4.29 shows a comparison between the design and experimental values for the angles by which the relative velocity is approaching the runner blades using the data presented in Chapter 3 and *Table 4.6* respectively.

Details of mathematical approaches for determining the pressure distribution for blades arranged in a cascade are available in the work of Hawthorne [1964] and Vavra [1960]. The analysis presented here uses a computer software developed by Lewis [1996] to investigate pressure distribution over the tip, middle and hub sections of the runner blades using test and design data for the geometry of the propeller turbine cascade.

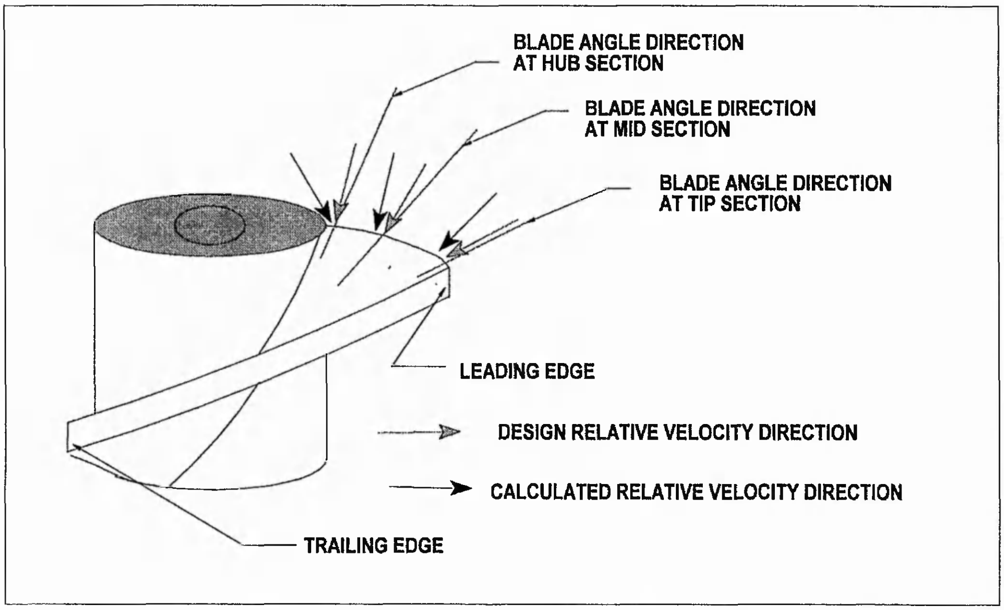


Figure 4.29 : Graphical representation of the revised velocity diagrams.

The design of the cascade was based on the findings of Eckert [1944] who has shown that the same efficiency could be attained by blades made of constant thickness sheets of metal as for aerofoil sections provided the geometry is preserved. The cascade pressure distribution analysis presented in this section therefore uses the C4 profile with a circular arc camber line defined as follows :

$$\frac{y}{c} = 4\frac{f}{c} \times \frac{x}{c}\left\{1 - \frac{x}{c}\right\} \qquad \{4.17\}$$

Using the geometry for any blade section and the relative velocity flow angles, the pressure distribution for the pressure and suction side of the cascade are determined and presented by a dimensionless coefficient defined as the pressure coefficient [Carter, 1961 and Hutton, 1959] given by

$$C_P = \frac{P - P_1}{\frac{1}{2}\rho\, W_1^2} \qquad \{4.18\}$$

where P is the static pressure on the blade surface and P_1 is the pressure upstream.

Figure 4.30, *Figure 4.31* and *Figure 4.32* show the pressure distribution for the hub, middle and tip section of the blade respectively based on the geometry determined by the design calculations.

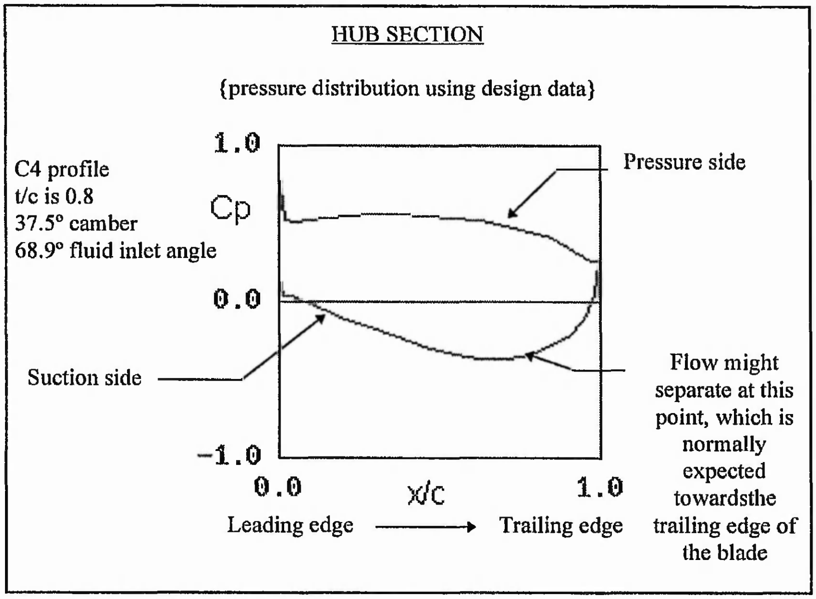


Figure 4.30 : Pressure distribution over the blade hub section - design.

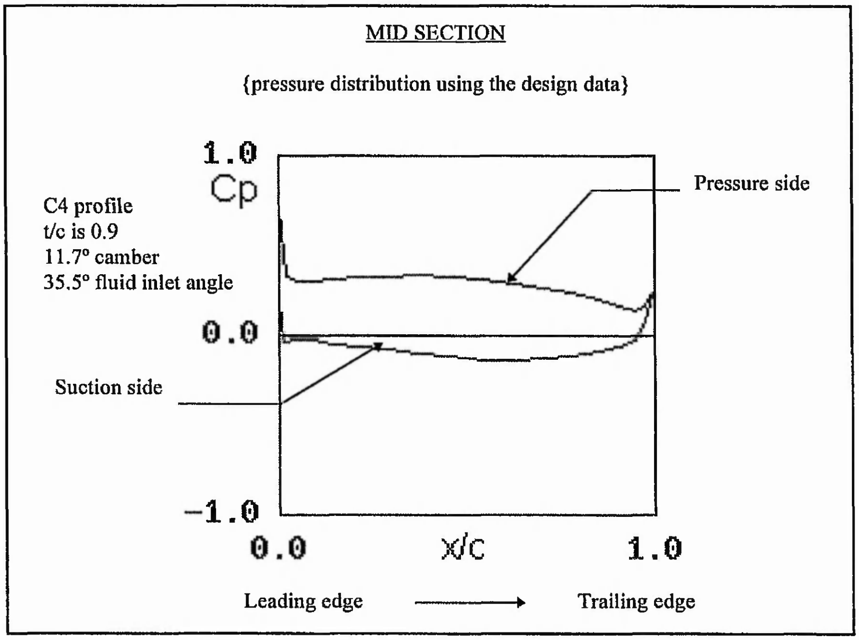


Figure 4.31 : Pressure distribution over the blade middle section - design.

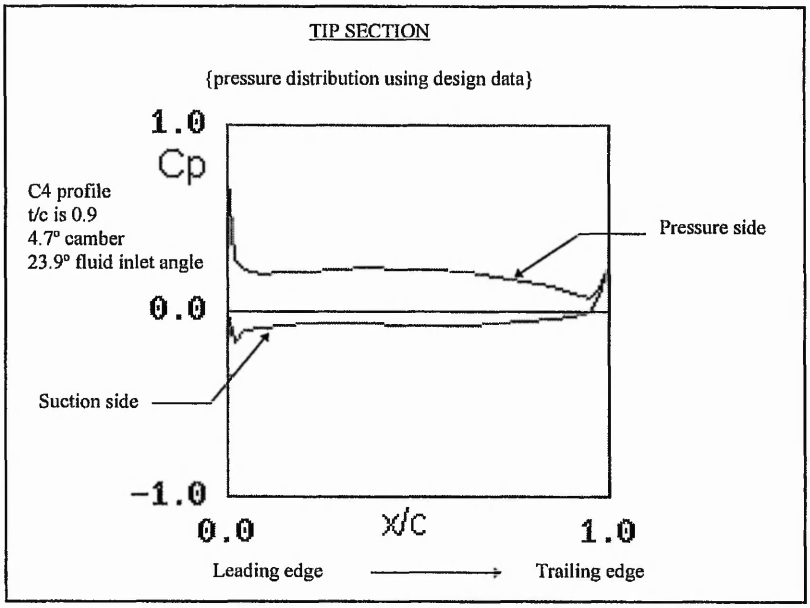


Figure 4.32 : Pressure distribution over the blade tip section - design.

In a paper published by Hutton [1959] it is suggested that pressure peaks should be avoided over blade surfaces as these lead to flow separation and therefore decrease in performance. The initial pressure peak at the leading edge is followed by a smooth horizontal curve for both the pressure and suction side of the blades. The results presented here show what is normally expected of a well designed circular arc profile.

The suction side of the hub section has a slight concave shape towards the trailing edge, a pressure peak that would cause separation in the event of increased incidence angle. This pressure peak can be avoided by using a parabolic blade shape[13], where the maximum camber is closer to the leading edge of the blade [Carter, 1946 and Hutton 1959]. It is therefore concluded that the design of the cascade would not experience any flow separation provided that the relative velocity direction is maintained.

Using the same profile and blade geometry as before, the pressure distribution for the flow angles determined by the revised velocity diagram calculations is shown by *Figure 4.33*, *Figure 4.34*, and *Figure 4.35* for the hub, middle and tip sections respectively.

[13] The use of such a shape is not practical for the intended application of the current turbine design.

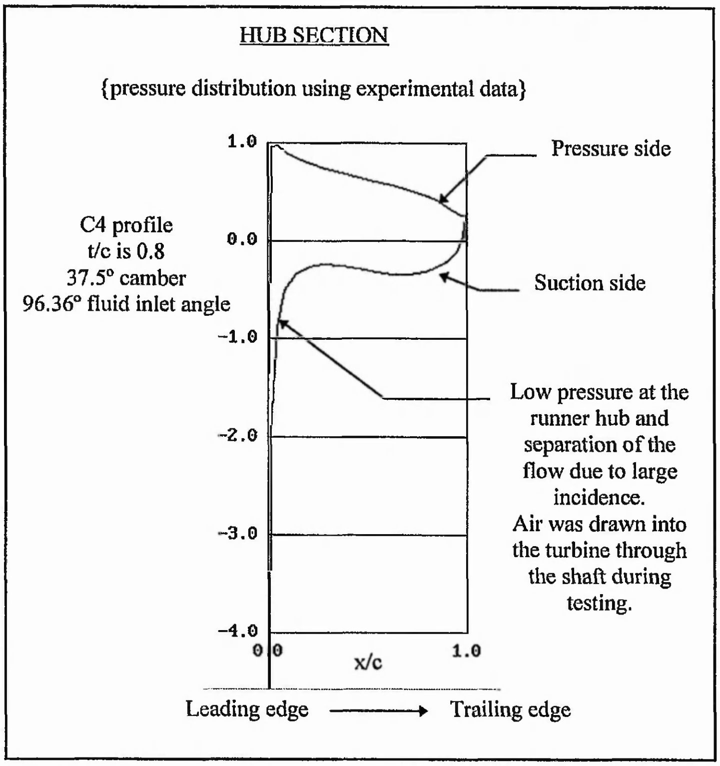


Figure 4.33 : Pressure distribution over the blade hub section - results.

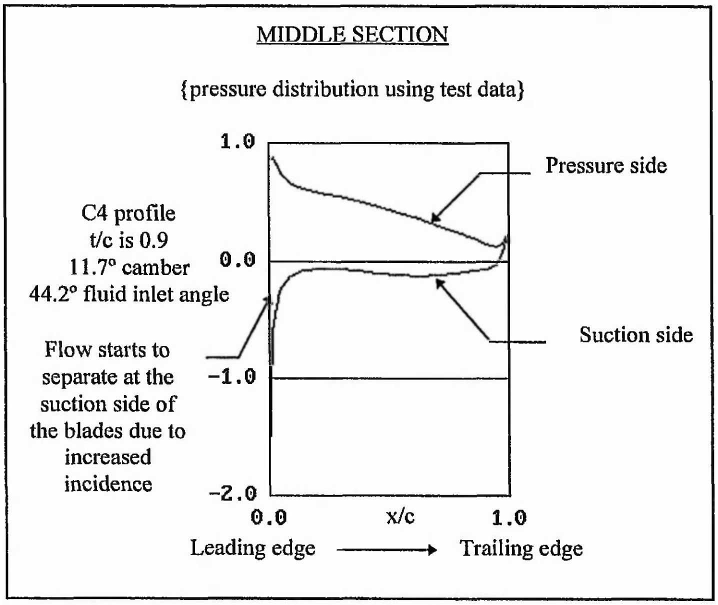


Figure 4.34 : Pressure distribution over the blade middle section - results.

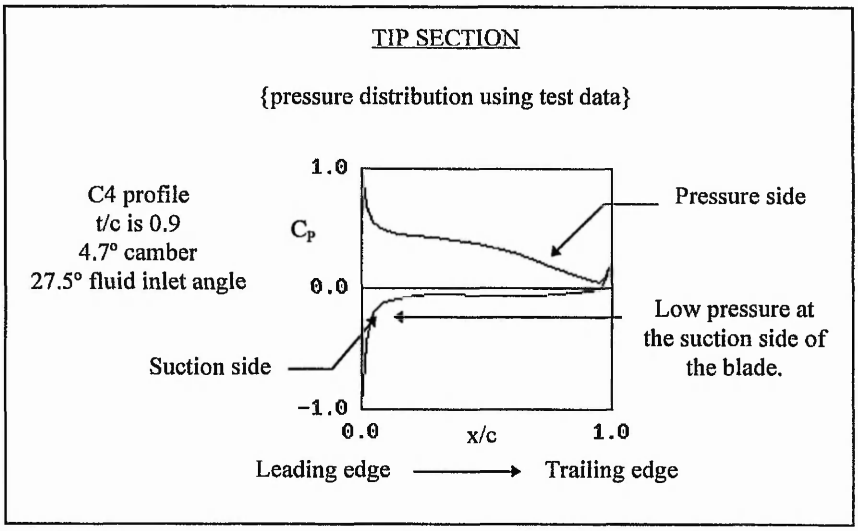


Figure 4.35 : Pressure distribution over the blade tip section - results.

Based on the data presented by *Table 4.6*, the relative velocity is approaching the runner blades at an angle of incidence of 3.7° and 27.5° at the tip and hub sections respectively. Flows at angles of incidence larger than 6° to 8° cause considerable drop in aerodynamic performance of an aerofoil [Turton, 1984]. Flow separation is also known to occur at these large angles of incidence which is verified by the reduced pressure at the suction side of the leading edge of the runner blades.

The conclusions drawn from the pressure distribution analysis are as follows :

- ❑ Blades whose camber lines are based on the circular arc can be used for turbine blade design. Pressure distribution over such shapes shows no sign of sudden pressures changes which result in flow separation and loss in performance.

- ❑ Flow separation and low pressures result when the large flow angles are experienced.

- ❑ Large blade camber can upset the flow distribution when circular arc profiles are used. As it is shown in section 4.5.2, smaller camber angles and blade twists can be obtained when lower operating speeds are used.

4.4.5 Fluid flow and loss analysis for the draft tube

The performance of the draft tube is also important because it has a significant effect on the efficiency and power output of a hydraulic turbine. As already explained in Chapter 3, the role of the draft tube is to reduce the velocity of water exiting from the turbine, thereby converting the excess kinetic energy of the exhaust stream into a rise in static pressure. Since the velocity head recovered by the draft tube represents a sizable fraction of the total effective head on the turbine, especially for heads as low as 2 to 3m, it is important for good turbine performance to minimize the exit losses from the draft tube.

For large machines with curved draft tube designs, numerical and computational methods are used to examine and analyze the complexity of flows [Shyy, 1986 and Kubota, 1996].

The analysis presented in this section uses information published for straight conical tubes based on the experimental work of Mehta [1977], Vuskovic [1966] and more recently the work of Hothersall [1988] and Monacelli [1987]. Hydraulic losses in the draft tube are grouped and examined as follows :

- Frictional losses, ξ_{frdt}.
- Expansion losses, ξ_{expdt}.
- Exit losses, ξ_{extdt}.

4.4.5.1 Frictional losses in the draft tube

The draft tube frictional losses are calculated using the same equations for friction losses through a circular pipe. The friction loss coefficient and hydraulic radius are considered for the half way through the draft tube length, as it is often described in literature [Ida, 1989]. The mid section of the draft tube has a diameter of 0.225m and for $0.048m^3/s$, the Reynolds number is 1.7×10^5.

Using the data for mild steel pipes, the friction coefficient is 0.005. The frictional losses in the draft tube are therefore calculated as follows :

$$\xi_{frdt} = 4\frac{fLV_{amid}^2}{2gD_{mid}} = 0.01m \qquad \{4.19\}$$

The friction losses in the draft tube are only 0.5% of the net head available.

4.4.5.2 Expansion losses in the draft tube

The expansion losses are determined by the following empirical formula which is based on the work of Butaev [1965] and Gubin [1973] :

$$\xi_{exp\,dt} = \zeta_{exp}\frac{\left[V_{a2} - V_{a3}\right]^2}{2g} \qquad \{4.20\}$$

The coefficient of expansion loss for a circular cross section straight draft tube design is related to the half draft tube angle and an empirical coefficient which is obtained using the following expression

$$\zeta_{exp} = 3.2\ tan\left\{\frac{\vartheta}{2}\right\}^{1.25} = 0.32 \qquad \{4.21\}$$

Using the information presented by *Figure 4.36* the draft tube expansion loss is equal to 0.1m which is 5% of the net head available.

4.4.5.3 Losses at the draft tube exit

As it is clearly pointed out by Soundranayagam [1979] and Sayann [1979] the main concern to a designer is the amount of whirl in the draft tube that is caused by the flow immediately after the trailing edge of the runner blades. This is of particular importance with machines of fixed geometry which are highly sensitive to any off design operating conditions. Some whirl in the draft tube is desirable as explained by Moses [1982] and can benefit performance by increasing local pressures on the draft tube walls thereby reducing the tendency towards separation [Monacelli, 1987].

However, every draft tube design would respond in a different way to a particular turbine runner geometry and thus the effects of whirling motion in draft tubes requires extensive experimental investigation.

The dimensions and diffusing angle of the draft tube were selected along the lines that any draft tube would have been designed for maximum pressure recovery. The revised velocity diagram calculations as well as the flow observations, described by section 4.4.3, indicate very clearly that there is an exit whirl component from the runner blades. The flow in the draft tube is therefore not axial, as originally assumed, and energy is wasted due to the tangential velocity leaving the runner which cannot be recovered. The exit loss is therefore calculated using the absolute velocity component at the draft tube exit given by

$$\xi_{extdt} = \frac{V_3^2}{2g} \qquad \{4.22\}$$

The mid section of the blade is used for the above calculation. The absolute velocity at the exit of the draft is 0.81m/s which account for a loss of 0.033m or 1.65% of the net head available to the turbine.

4.4.5.4 Pressure measurements in the draft tube

The calculation presented here shows that the losses in the draft tube can be calculated using the Bernoulli equation. If the velocity through the draft tube is not axial due to an exit whirl from the runner blades, the values of the absolute velocities are used.

Pressure tappings were used to measure the pressure immediately below the turbine runner. With reference to *Figure 4.36*, the Bernoulli equation is applied between the section where the pressure is measured and the exit of the draft tube as follows :

$$\xi_{loss} = \frac{P_2 - P_3}{\rho g} + \frac{V_2^2 - V_3^2}{2g} + z \qquad \{4.23\}$$

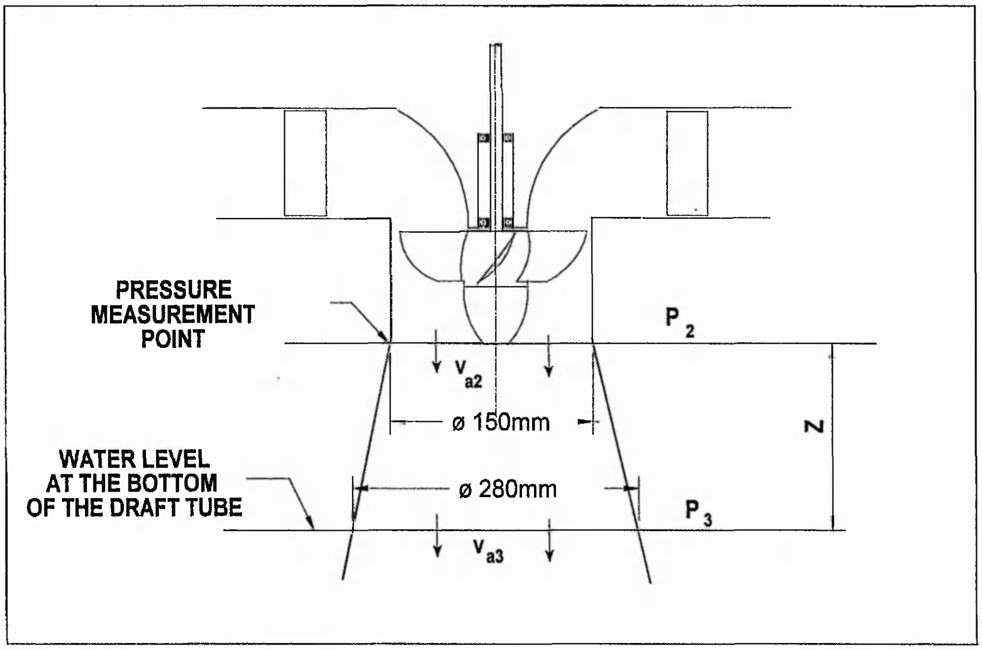


Figure 4.36 : Pressure measurements in the draft tube.

The pressure head below the runner was measured -1.35m at bep. Using the data obtained in section 4.4, with the distance between points 2 and 3 being 1m the head loss is equal to 0.15m or 7.5% of the net head available.

4.4.5.5 Flow observations during testing

Several performance tests were repeated for a number of times during which the following observations were made :

- Swirling of flow was clearly visible coming out at the bottom end of the draft tube.

- Sudden pressure fluctuations and vibration of the draft tube were observed at flows before and after the best efficiency point.

The effects of swirling and the significance of eddies at the exit of the draft tube have been explained earlier in the thesis.

Unsteady flow behind water turbine runners and vortex break down have long been recognized with fixed runner blades. The unsteady flow is characterized by fairly regular oscillations of pressure in the turbine, resulting in power swings, vibration and noise. Surge is generally attributed to eccentric rotation of a vortex core formed in the draft tube when whirling of flow is high. The whirl at the runner exit increases as the point of operation moves away from the point of zero whirl, and at a certain high whirl the flow stagnates along the draft tube axis, a phenomenon usually referred to as vortex breakdown. Further increase in whirl develops a reversed flow, thus a dead water core (fluid column that does not take part in the main flow) is formed in the draft tube.

As a result of this dead core that is formed along the axis of the draft tube, the area of the draft tube is decreased, and obviously velocities become higher and pressure surges cause power swings. A detailed explanation of the vortex break down phenomenon as well as experimental investigation of pressure surges in draft tube can be found in the work of Mehta [1977], Sayann [1979] and Soundranayagam [1979].

4.4.6 Discussion

The major causes of increased hydraulic losses have been identified as follows :

- ❑ Incorrect flow distribution in the spiral casing due to simplified design assumptions.
- ❑ Incorrect setting of the guide vanes.
- ❑ Head losses due to the area change between the spiral casing.
- ❑ Large angles of incidence at the runner blades.

Figure 4.37 shows the head losses produced by the systematic analysis which was presented in this chapter. *Figure 4.38* shows a comparison between the results obtained by the analysis presented in this chapter and the head losses that were originally estimated during the design of the turbine, described in Chapter 3.

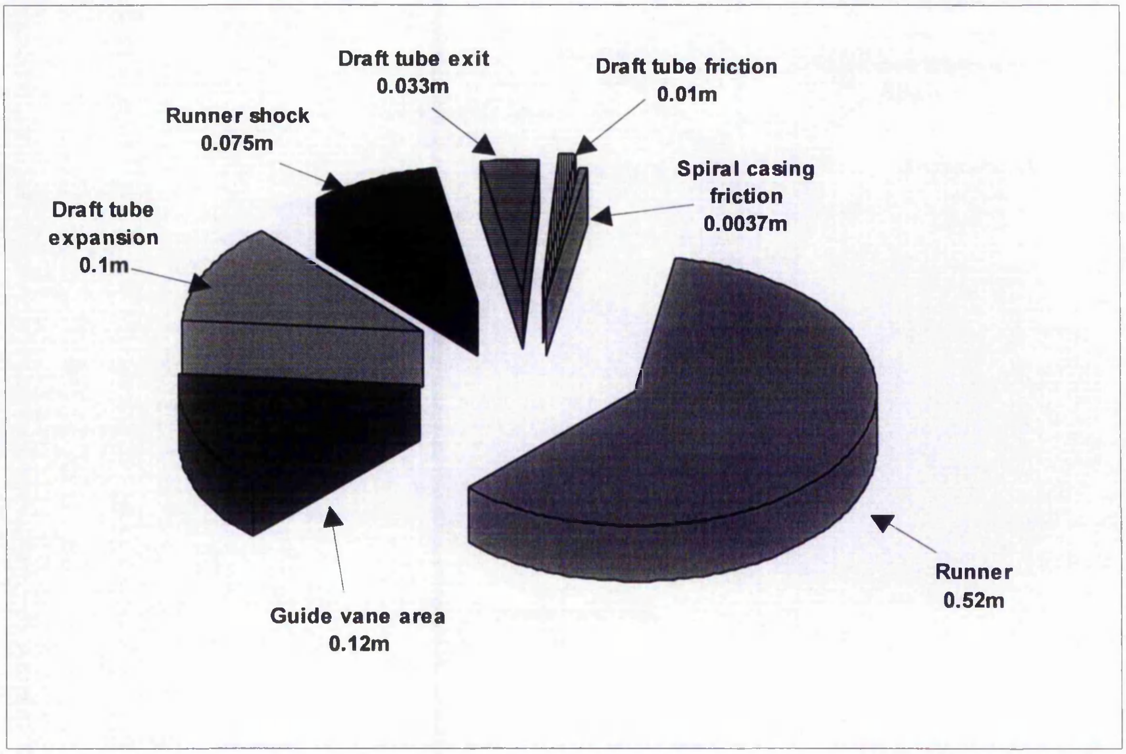


Figure 4.37 : Loss breakdown using the calculated values for bep at 2m of head.

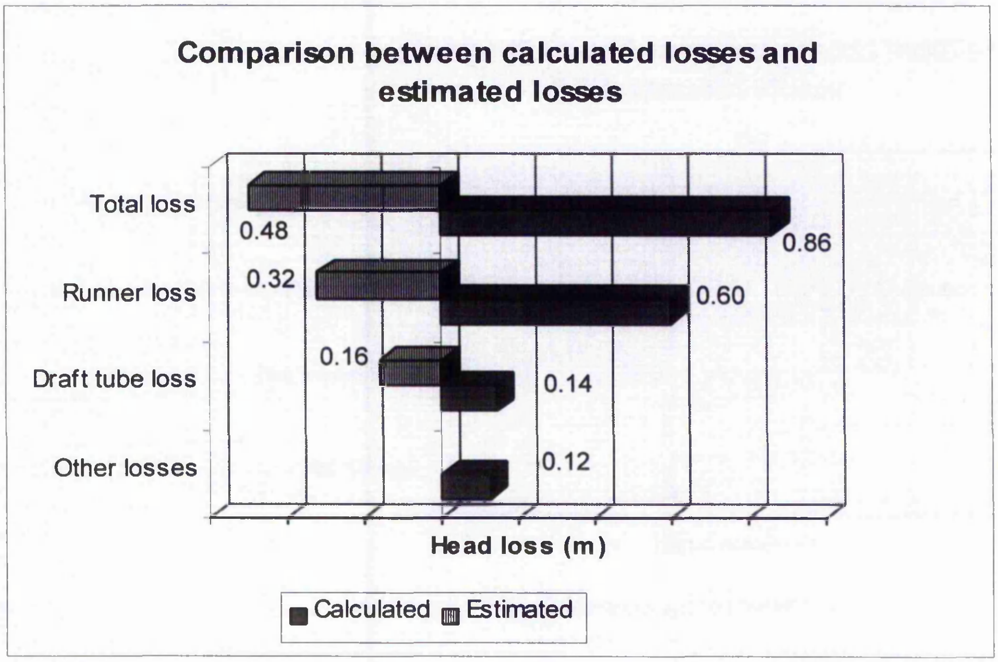


Figure 4.38 : Estimated and calculated head losses at 2m of head.

With reference to *Figure 4.38*, it is therefore concluded that an estimate of the hydraulic losses for small propeller turbines can be produced during the design process as long as the following losses are incorporated into the design :

- ❑ *Head losses due to friction in the penstock, spiral casing and the draft tube.*
- ❑ *Head losses due to the expansion in the draft tube.*
- ❑ *Head losses through the guide vane area.*
- ❑ *Head losses due to the exit velocity from the draft tube.*

According to the published work of Hutton [1954 and 1983], for large Kaplan turbines the losses are divided into 1/3 and 2/3 between the draft tube and runner respectively. This method of division of the losses can still be used for small machines to provide an estimate for the runner losses. However, as it is seen by the proportion of the runner losses to those of the draft tube, a more accurate estimate would be obtained by dividing the losses between the draft tube and the runner into 1/4 and 3/4 respectively.

4.5 Assessment of the design methodology

Despite increased hydraulic losses, a maximum turbine efficiency of 53% and 55% was obtained at 2m and 4m of head and a speed of 1000rpm.

With reference to the literature review presented by Chapter 2, the following are concluded :

- ❑ The efficiency of the turbine compares well with existing designs.
- ❑ The turbine is suitable for direct coupling to a 6 pole induction generator.

The aim of this section is to provide design alterations and solutions to improve the flow distribution in the spiral casing and the guide vane effectiveness. Moreover, a new propeller turbine design is introduced to assist fabrication of the runner blades.

4.5.1 Assessment of the spiral casing and guide vane design

The design of the spiral casing is one of the key elements for a successful low head propeller turbine design, provided the runner blades are made to the geometrical accuracy produced by the design. Despite the simplified shape of the spiral casing, that would assist manufacturing and minimize costs, the velocity distribution can only be determined by the method described in section 4.4.1.3.

Based on the existing runner geometry, *Table 4.8* shows the modified casing dimensions that would provide the correct flow distribution into the runner blades. The dimensions are larger than the existing design and would therefore result into additional manufacturing and material costs. Moreover, transportation would be difficult due to the increased weight. Both the guide vane setting angle and the guide vane height would be kept at 75° and 35mm respectively.

TURBINE DESIGN		CASING DESIGN	
Q	$0.072m^3/s$	Q	$0.072m^3/s$
V_{uT}	1.20m/s	V_{uT}	1.20m/s
V_{uH}	3.0m/s	V_{uH}	3.0m/s
α_1	75°	α_{go}	75°
casing to be of square cross section 0.48m × 0.48m			

Table 4.8 : Modified spiral casing for correct flow distribution .

4.5.2 Assessment of turbine runner design

Table 4.9 presents a modified design of a propeller turbine using the performance characteristics of the current prototype turbine. The design speed was selected to be 1040rpm for direct coupling to a 6 pole induction generator while the net head available was maintained at 2m.

	Original Design	Modified Design
H (m)	2	2
Q (m^3/s)	0.072	0.032
N (rpm)	1560	1040
n_s (min^{-1})	248	110.6
D_H/D_T	0.4	0.6
D_T (mm)	150	150
	Tip section	
β_1 (°)	23.9	22.7
β_2 (°)	21.8	19.1
β_{∞} (°)	22.8	20.9
θ (°)	4.7	8.2
	Hub section	
β_1 (°)	68.9	48.3
β_2 (°)	44.9	29.9
β_{∞} (°)	55.3	39.1
θ (°)	37.5	28.8
Blade no :	4	8

Table 4.9 : Comparison in blade twist and camber angle for varying hub to tip ratio.

The combined effect of slower operating speed and higher hub to tip ratio reduces the twist required between the hub and tip section of the runner blades. Moreover, with the camber angle at the hub section reduced, the pressure distribution over the hub section of the cascade would be improved compared with the one obtained for the current prototype.

This modified turbine design proves that the methodology used for the purposes of the current research project can be applied successfully for the design of propeller turbines operating at 1040rpm as well as 1560rpm for direct coupling to 6 and 4 pole induction generators respectively. Furthermore. the method allows the turbine dimensions to be tailored to those of standard pipe dimensions at any speed and flow rate.

4.5.3 Assessment of draft tube design

The design of the draft tube requires no further improvement or alterations. Future designs should consider the loss analysis presented in section 4.4.5.

4.5.4 Assessment of non contact seal performance

Pressure readings across the non contact seal were not taken due to limitations imposed by the existing test rig set up. However, the performance of the seal was monitored during testing with the following observations :

- ❑ *Runner 1* performance testing : Some water leakage was observed for an operating speed of 0rpm to 1200rpm.

- ❑ *Runner 2* performance testing : For a speed range of 0rpm up to about 850rpm there has been some water leakage but not as much as with *runner 1*. At speeds over 900rpm, air was drawn into the turbine through the turbine shaft. The observation of air suction at the seal occurred at the same time as a reduction in turbine efficiency. This observation is also verified by the low pressure distribution across the blade cascade at the hub section of the blade, discussed in section 4.4.4.3.

In order to maintain a constant sealing gap, a recess was machined onto the shaft of the current turbine runner as it is shown by *Figure 4.39*. When the turbine runner is attached to an extended induction generator shaft, thin metal shims would be used to adjust the vertical position of the generator and therefore the sealing gap, as shown by *Figure 4.39*.

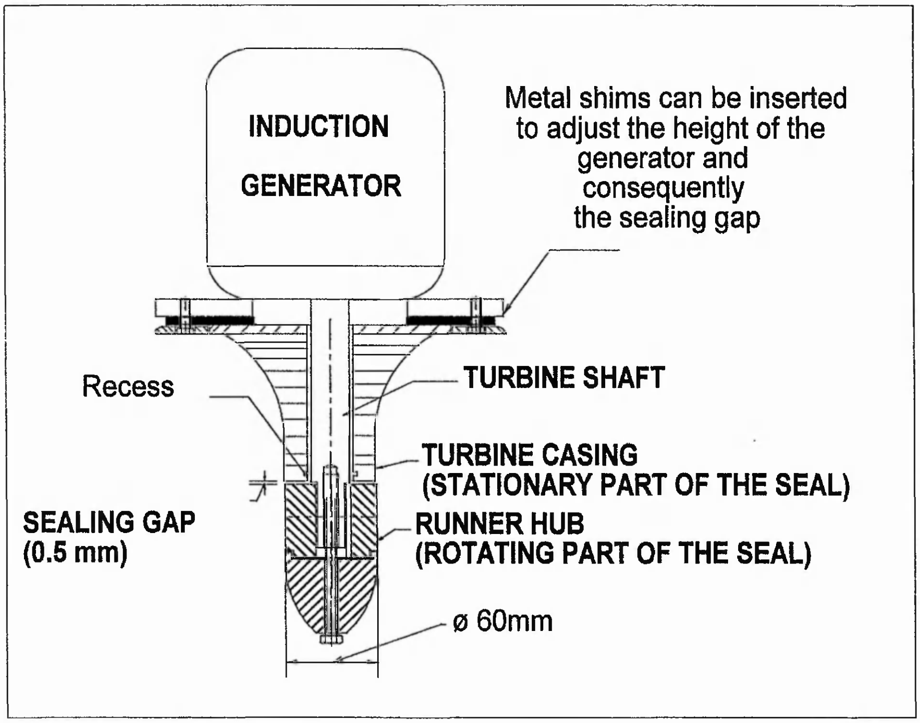


Figure 4.39 : Fine adjustments for the non contact seal.

The non contact seal is an attractive idea for reaction turbines, but its performance is limited to a certain range of operating speeds. Moreover, as it was demonstrated by the calculations performed in Chapter 3, the performance depends on the gap between the rotating and stationary surface. In addition to the non contact seal, another form of sealing is required to prevent water leakage and/or air drawn into the turbine system. Mechanical seals are not suitable for use as they experience reliability problems due to silt and sand in the water. As an alternative a stuffing box can be used. Such an arrangement was successfully tried by Katmandu Metal Industries for their micro hydro turbine designs. However, modifications to the layout of the turbine casing would be required to accommodate the stuffing box.

4.5.5 Assessment of the current design using the Cordier diagram

The design of the prototype propeller turbine runner was dominated by the following parameters :

- ❑ The turbine had to be designed at 1560rpm for direct coupling to a 4 pole induction generator which is standard practice for 1kW micro hydro schemes.
- ❑ The dimensions of the turbine had to be kept to those of standard pipe work fittings.
- ❑ Flow surfaces had to be kept simple to assist fabrication.

Over the years, designers have used the dimensionless quantities of specific speed and specific diameter in an attempt to develop guidelines for optimum designs. Based on the information published by Bohl [1991] and Sigloh [1993], the following dimensionless parameters are used to determine the position of a turbine on the Cordier diagram :

- ❑ Flow coefficient, $\Phi = \dfrac{Q}{u\, D_T^2\, \dfrac{\pi}{4}}$
- ❑ Head coefficient, $\Psi = \dfrac{gH_N}{\dfrac{u^2}{2}}$
- ❑ Dimensionless specific speed, $n_s = \dfrac{\Phi^{1/2}}{\Psi^{3/4}}$
- ❑ Dimensionless specific diameter, $d_s = \dfrac{\Psi^{1/4}}{\Phi^{1/2}}$

The dimensionless characteristics for the current turbine design as well as the proposed new design are shown by *Table 4.10*.

Current turbine design		New turbine design	
n_s	1.59	n_s	0.42
d_s	1.24	d_s	2.22
Operating speed	1560rpm	Operating speed	1040rpm

Table 4.10 : Dimensionless specific speed and diameter for the current turbine and the proposed new design.

Using the information presented by *Figure 4.40*, it is therefore concluded that both prototype turbine and the new design are within the optimum design recommendations.

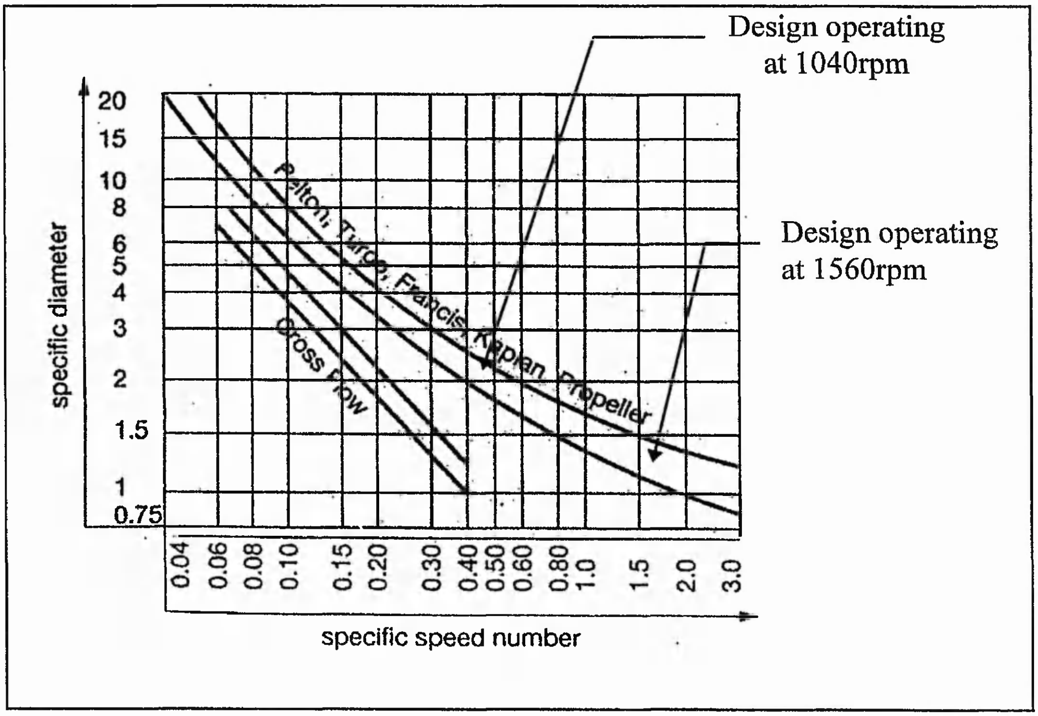


Figure 4.40 : Cordier diagram.

4.6 Assessment of fabrication and material selection

Manufacture of the prototype propeller turbine has been carried out using methods and equipment available in workshops of developing countries. The turbine dimensions were purposely tailored to those of standard sizes to allow sections of the turbine to be made of pipe sections and therefore minimize manufacturing time and costs. Mild steel was used for every component of the prototype propeller turbine with the exception of the nose of the turbine hub. This was made of aluminum to reduce the weight of the runner assembly and therefore the axial forces on the supporting bearings. The processes involved during manufacturing are as follows :

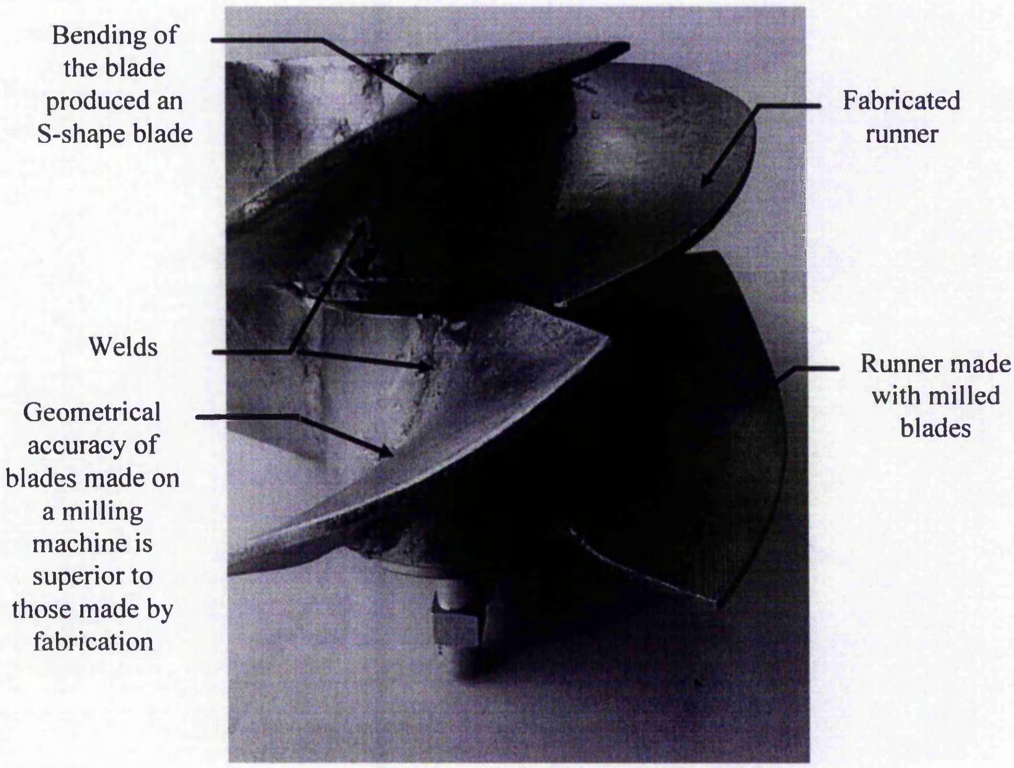
- ❑ Turning on a lathe the turbine hub and the shaft extension.
- ❑ Cutting sheets of metal on a power saw.
- ❑ The use of a welding machine.
- ❑ Bending and twisting sheets of metal of constant thickness for the runner blade forming with a wrench.

A number of limitations have been identified when fabricating propeller turbines. These can be summarized as follows :

- ❑ Complex geometrical shapes of runner blades with a considerable twist from hub to tip, cannot be fabricated by bending and twisting metal sheets to the desired geometrical accuracy produced by blade design, refer to *Picture 4.1*.
- ❑ The workshop is required to have an angle measuring device, e.g. a protractor, or a pattern with the correct blade shape, to assist fabrication to the desired blade setting angles from hub to tip.
- ❑ The use of welding causes excessive thermal stresses which cause considerable geometrical distortion of the fabricated runner blades.

In order to avoid or minimize these problems, the designer has the following options :

- ❑ Redesign of the turbine so that the runner blades have less twist. Turbine redesign has been introduced in section 4.5.2.
- ❑ Use other manufacturing methods, e.g. mechanical forming and/or casting. These are discussed further in Chapter 5.



Picture 4.1 : The fabricated runner and the runner manufactured with blades made on a milling machine.

When fabrication is used to manufacture propeller turbines, the choice of material is limited to mild steel. Though widely available in developing countries at low cost, mild steel has poor corrosion resistance. The use of anti-corrosion paint is not appropriate due to the sand and silt content in the water. Galvanizing the turbine runner, spiral casing and draft tube is an alternative. However, the costs involved as well as the equipment required need to be examined before final conclusions are drawn. Such methods are part of the further work described in Chapter 5.

4.7 Cost analysis

Table 4.11 shows a cost breakdown of the prototype propeller turbine prepared by the Department of Mechanical and Manufacturing Engineering.

COMPONENT	TIME(hr)	Cost £20/hr
Draft tube & flanges	7.5	£150
Material, mild steel		£50
Guide vanes, runner blades & hub, casing, flanges	70	£1400
Material		£170
Shafts, key ways, assembling, generator support.	32.5	£650
Material		£40
TOTAL		£2460

Table 4.11 : Cost break down for the propeller turbine development.

The first prototype turbine has taken a total of 110 workshop hours to be made and has a weight of about 70 kg (with no penstock). Based on the information presented by *Table A.1* in Appendix A, the cost for the same turbine to be made in a Nepali workshop is as follows :

- ❑ Material : 70kg of mild steel, Rs36/kg.
- ❑ Workshop time : 110hrs, Rs300/day for a skilled work person.

The total cost for the turbine to be made in Nepal is estimated to be Rs6645[14]. At 2m of head, the fabricated runner is producing 0.37kW at 39% efficiency. A typical Pelton wheel would cost Rs12000 (buckets, runner, material, spear valve and runner housing).

[14] Information supplied by Intermediate Technology Nepal, PO Box 2325, Katmandu, Nepal and Katmandu Metal Industries Ltd., Cha3-812 Nagal Quadon, Chhetrapati, Katmandu 3, Nepal.

Typical efficiencies for 1kW micro hydro Pelton turbines are estimated around 60% to 70%. Based on these figures, the cost per kW is :

- Rs18462 or US$374 per kW for the Pelton wheel.
- Rs17960 or US$364 per kW for the propeller turbine.

An increase in the efficiency of the propeller turbine by 15% would result in a decrease in the cost of the propeller turbine by Rs5878 or US$119 per kW.

4.8 Summary

The results presented in this chapter show that the turbine performance is sensitive to the geometrical accuracy of the runner blades. Bending and twisting sheets of metal to form turbine blades to the design angles has been difficult to achieve and the inaccurate manufacture reduced the turbine efficiency. The S-shape of the fabricated blades has produced a flatter performance characteristic curve to what is normally expected from a propeller turbine. A second runner, whose blades were machined to the correct geometry produced by the design calculations, has shown an improvement in efficiency from 40% to 53% at 2m of head.

The analysis presented in section 4.4 has shown that the performance of the turbine has been affected by the use of a simplified velocity distribution in the spiral casing, and the losses associated with it. As a result of the flow distribution in the spiral casing, the effectiveness of the guide vanes was reduced. This was also accompanied by a loss of angular momentum.

The design of the current turbine assists fabrication at an estimated US$364/kW. This is very close to the cost/kW that of a Pelton wheel runner which is currently being used for high head micro hydro schemes. It is therefore concluded that the current propeller turbine design is a promising technology and compares well with other propeller turbine designs which operate at slower speeds and/or have higher manufacturing costs.

Chapter 5

Conclusions and

Recommendations

for Further Work

"Give a man a fish, as the saying goes, and you are helping him a little bit for a very short while; teach him the art of fishing, and he can help himself all his life...but teach him to make his own fishing tackle and you have helped him to become not only self supporting but also self reliant and independent"

E. F. Schumacher

5.1 Appropriate design methodology

A detailed appraisal of the potential and use of existing propeller turbine technology for low head micro hydro electric schemes was undertaken and has been presented in Chapters 1 and 2. The main conclusions drawn from these chapters are :

- ❑ Propeller turbines directly coupled to induction generators can be both an appropriate and beneficial technology in developing countries.
- ❑ Existing propeller turbine technology is inappropriate for use in developing countries because the technology is unsuitable to be manufactured and maintained locally.
- ❑ There are reliability problems with existing technology. High silt content in water, a frequent condition in developing countries due to deforestation and storms, causes bearing and contact seal failure.

It was therefore clear that a new design approach had to be adopted in order to :

- ❑ Utilize the local skills and materials available.
- ❑ Use engineering standards that would maintain simplicity of the design and yet provide reliability for silty environments.
- ❑ Reduce manufacturing costs.

In order to accomplish these, the whole concept of design, which was introduced in Chapter 3, was based on the following :

- ❑ The use of tools and processes that are currently employed by local workshops for manufacturing Pelton wheels and cross flow turbines.
- ❑ The use of higher operating speeds than current designs to allow direct coupling of the turbine runner onto the shaft of either a 4 or 6 pole induction generator.
- ❑ Tailoring the turbine dimensions to standard pipe sizes to assist fabrication and therefore minimize manufacturing time and costs.
- ❑ The use of simplified flow surfaces to assist fabrication.

- ❑ The use of the hydrodynamic effects between the stationary housing of the turbine runner and the rotating runner hub to develop a maintenance free non contact seal suitable for use with reaction turbines. Non contact seals are currently used with impulse turbines in Nepal.

The turbine arrangement used has a scroll casing, radial guide vanes and a straight conical draft tube. This arrangement enables the direct coupling of the propeller runner onto the generator shaft. This arrangement requires a small extension of the generator shaft which is successfully used with Peltrc sets [Smith, 1992a]. Moreover, the generator's bearings are used to take the axial and radial loads. Less components, such as bearings and bearing housings are used, less workshop time is required for manufacturing and costs are therefore reduced.

5.2 Turbine manufacture and testing

Conventional axial flow turbine runners have aerofoil shape blades. These shapes require skills and machinery, e.g. milling machines, that are not easily found in small workshops in developing countries. The runner blades were therefore made of constant thickness metal sheets.

Bending and twisting sheets of metal to form turbine blades has been difficult to achieve, and inaccurate manufacture reduced turbine efficiency. The geometrical accuracy of the runner blades was further affected by the thermal distortion as a result of arc welding the blades onto the runner hub. A second runner was manufactured, for comparison, with blades machined on a CNC milling machine. Test results show that the turbine performance is sensitive to the geometrical accuracy of the runner blades.

The turbine maximum efficiency of 53% and 55% at 2m and 4m of head respectively, although low in comparison with large axial flow turbines designs, is acceptable for village hydroelectric schemes. Similar low head propeller turbine designs have either poor performance [Green, 1993a] or run at much lower speeds [Parker, 1996].

From the test results and analysis of losses, it can also be concluded that :

- ❑ The range of best efficiency is limited by the fixed geometry of the runner blades and the guide vanes.

- ❑ The main losses in the turbine are within the runner.
- ❑ The smaller the runner diameter the bigger the leakage losses for a certain tolerance between the runner tip and the runner enclosure.

A revised design for a propeller turbine operating at 2m head has been developed, based on the information from the test results. This turbine is designed for direct coupling to a 6 pole induction generator. The runner tip diameter is maintained at 150mm. As a result of using a slower specific speed, the hub to tip ratio is increased and the twist between the hub and the tip section of the blade is therefore reduced. This design will be easier to fabricate accurately.

A non contact seal was designed and incorporated into the existing arrangement of the prototype propeller turbine. The non contact seal requires no special tooling for manufacture. Despite the fact that the performance of the non contact seal is dependent on the operating speed of the turbine, the hub diameter and the sealing gap, the idea of a non contact seal is very attractive for use with reaction turbines intended for micro hydro power generation.

5.3 Range of application

The prototype propeller turbine is suitable for low head applications for which no existing turbine is appropriate. *Figure 5.1* shows the range of application for a cross flow and propeller turbine based on the dimensionless specific speed values for a speed range of 300rpm to 1000rpm.

The values of these speed limits are realistic for using a 6 pole generator either directly coupled to the turbine shaft or driven by a speed increasing mechanism. As is clearly seen, the propeller turbine offers the advantage of direct coupling to the generator shaft where the cross flow turbine requires a belt drive. The propeller turbine should also be less expensive than a cross flow turbine. The estimated cost/kW for the current prototype is US$364 and compares with the manufacturing costs of a Pelton wheel.

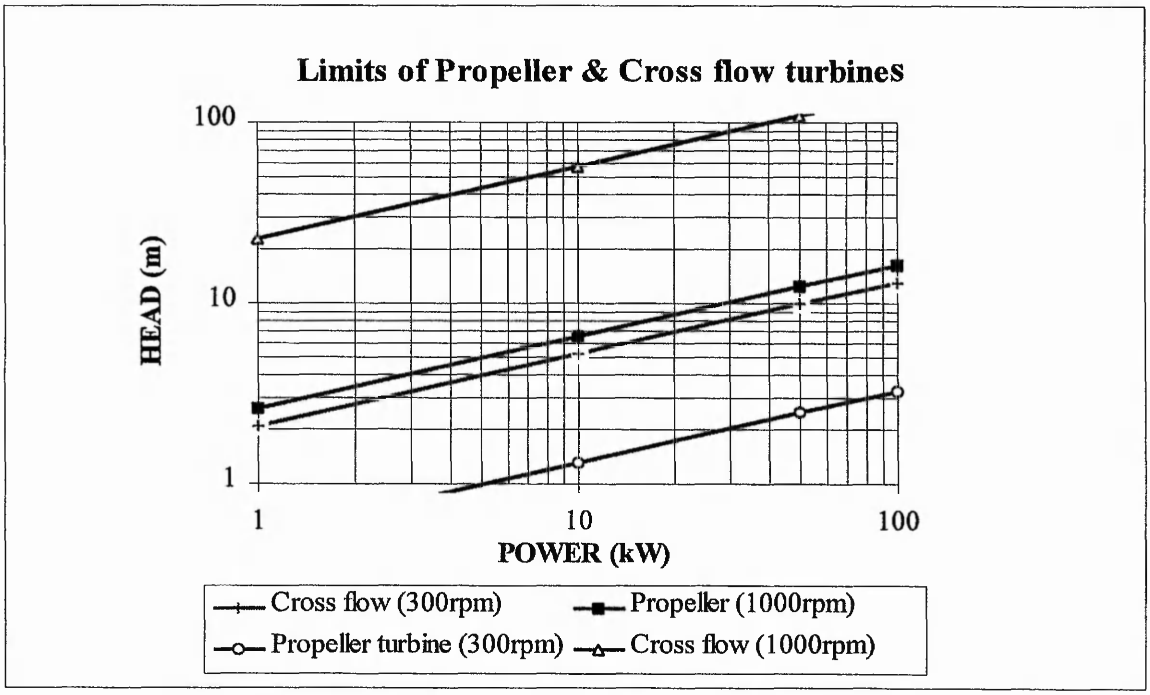


Figure 5.1 : Limits of Propeller and Cross flow turbines.

5.4 Recommendation for further work

Based on the knowledge and expertise gained from this research project, in the author's opinion, further work is required for performance optimization and improved manufacturing accuracy at low cost as follows :

Propeltric unit development.

- ❑ Design and test a new runner and an adjustable guide vane system to match the existing spiral casing geometry. The runner hub and runner blades should be machined as a single piece on a suitable CNC milling machine for improved geometrical accuracy. Alternatively, a new spiral casing should be designed to match the characteristics of the current turbine runner.

- ❑ Introduce pressure and velocity measurements to the existing test rig by means of pressure transducers and Acoustic Doppler Velocity meters. These will allow further experimental data to be obtained from the spiral casing, the turbine runner and the draft tube.

- Use numerical methods, such as those described by Chen [1986], Holmes [1982] and Morfidakis [1991], and finite element packages, such as ANSYS, to perform a three dimensional analysis of the flow using the existing propeller turbine geometry. Moreover, flow visualization methods, using for example FLUX - a computer software developed for turbomachinery flow visualization [Roth, 1996], will enable further design improvements to be introduced for increased runner performance.

- The performance evaluation and further development of the non contact seal requires setting up of a purpose built test rig. Further testing is required to ensure reliable sealing over a wide range of operating speeds.

- Field testing of propeltric units in developing countries.

Manufacturing methods

- Investigate the use of purposely built jigs and/or patterns for improved accuracy of fabrication by mechanical forming of the runner blades. The adaptation of existing technology for the needs of local workshops also requires further study.

- Investigate the use of casting as an alternative to fabrication.

- Derive solutions that would provide effective corrosion resistance and minimize silt erosion, such as galvanizing and/or ceramic coatings, with emphasis given on cost and availability of materials in the poorest countries.

- Despite the compact design of the propeller turbine presented in this thesis, ease and costs of transportation depend on the weight and size of the spiral casing. It is therefore necessary to find ways that would allow the spiral casing to be incorporated into the concrete floor of the power house. This is widely used for large hydroelectric schemes where special forming and concrete making skills are required to ensure accuracy of dimensions, good surface finish of the flow surfaces and vertical alignment of the guide vanes.

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Appendix A : Cost Analysis

A.1 Cost breakdown of a Nepali Peltric unit

The information presented here was provided by a Nepali manufacturer[1] and the prices quoted are in Nepal Rupees (exchange rate Rs80 : £1). The prices were quoted for a Peltric unit of 1kW power output, 150 pitch circle diameter (PCD).

COMPONENT	COST	COMPONENT	COST
Runner 150 pcd18 buckets	Rs1800	Training	Rs150
Skilled labour / day	Rs300	Testing & Assembly	Rs500
Semi skilled labour / day	Rs200	Guarantee	Rs2500
Unskilled labour / day	Rs100	**Selling price**	**Rs29000**
Metal for flanges, nuts and bolts	Rs300	Controller 1kW	Rs12000
Selling price	**Rs3000**	100m penstock 63mm HDPE pipe	Rs5000
Housing for runner	Rs2000	Transmission cost 100m ($6mm^2$) cable	Rs1200
Spear Valve	Rs7000	1 Bag of cement	Rs500
Motor 1.5HP	Rs7000	Labour	Rs500
Controller box	Rs700	Power house	Rs2000
Capacitors	Rs1200	Trash rack / penstock accessories	Rs1500
Voltmeter	Rs300	Transport	Rs200
MCB 3phase	Rs600	**Total**	**22900**
(or MCB 1phase)	Rs300	**Total cost per kW installed**	**Rs51900**
Other components	Rs500		

Table A.1 : 1kW Peltric unit cost breakdown made in Nepal.

At the time of this quote, mild steel prices for metal sheets were in the range of Rs32 to Rs36 per kg. Mild steel pipes cost more and range from Rs50 to Rs60 per kg[2].

[1] Katmandu Metal Industries Ltd., Cha3-812 Nagal Quadon, Chhetrapati, Katmandu 3, Nepal.

[2] Information supplied by Drona Upadhyay, Intermediate Technology Nepal, PO Box 2325, Katmandu, Nepal.

A.2 Cost comparison of various energy options

Component	Description	Life (years)	Initial costs	life - cycle cost
Micro Hydro				
Hydropower scheme	5 kW plant - turbine made locally	15	\$ 10,000	\$ 10,000
Transformer / switchgear	step up and step down transformers @ \$ 1000	15	\$ 2,000	\$ 2,000
Secondary distribution system	7 km @ \$ 4,500 / km	15	\$ 32,000	\$ 32,000
Service drops / house wiring	100 @ \$ 50 / household	15	\$ 5,000	\$ 5,000
Operation & maintenance	\$ 100 / month			\$ 9,000
Parts	\$ 1000 / year			\$ 8,000
		Initial capital cost	\$ 49,000	
		Life - cycle cost		\$ 66,000
		Life - cycle cost / consumer		**\$ 660**
Diesel Engine				
Diesel genset / powerhouse	two 6 kW gensets @ \$ 7000; powerhouse \$ 4000	8	\$ 18,000	\$ 24,000
Secondary distribution line	4 km @ \$ 4,500 / km	15	\$ 18,000	\$ 18,000
Service drops / house wiring	100 @ \$ 50 / household	15	\$ 5,000	\$ 5,000
Overhaul	major overhaul of genset every 2 years @ \$ 5,000			\$ 17,000
Operation & maintenance	fuel cost : \$ 0.30 / kWh; parts & labor :\$ 2,000 / year			\$ 36,000
		Initial capital cost	\$ 41,000	
		Life - cycle cost		\$100,000
		Life - cycle cost / consumer		**\$ 1,000**
Grid Extension				
Primary distribution line	15 km @ \$ 4,500 / km	15	\$ 68,000	\$ 68,000
Transformer / switchgear	Step - down transformer @ \$ 1,000	15	\$ 1,000	\$ 1,000
Secondary distribution line	4 km @ \$ 4,500 / km	15	\$ 18,000	\$ 18,000
Service drops / house wiring	100 @ \$ 50 / household	15	\$ 5,000	\$ 5,000
Cost of energy	750 kWh / month @ \$ 0.15 / kWh			\$ 10,000
Operation & maintenance	service man @ \$ 40 / month			\$ 4,000
		Initial capital cost	\$ 92,000	
		Life - cycle cost		\$ 106,000
		Life - cycle cost / consumer		**\$ 1,060**
Household PV units				
PV array	55 Wp output	15	\$ 380	\$ 380
Controller		8	\$ 100	\$ 140
Support / conductors / misc.		15	\$ 60	\$ 60
Installation	labour		\$ 40	\$ 40
Battery	100 Ah deep - discharge lead acid	4	\$ 140	\$ 340
Operation & maintenance	\$ 1.50 / month for technician to monitor			\$ 140
		Initial cost	\$ 720	
		Life - cycle cost / consumer		**\$ 1,100**

Table A.2 : Breakdown of costs for various electricity supply options to serve a Nepali village with 100 consumers, [adopted from Inversin, 1994].

Appendix B : Field Trips and Visits

B.1 Micro hydro in developing countries

The development of micro hydro schemes in many countries has been a process of improvement and modernization of traditional technology that has been used for agro-processing for many centuries. This section presents specific aspects of this development with reference to Nepal, Sri Lanka and Peru.

B.1.1.1 Micro hydro in Nepal

In Nepal, several centuries ago, agro-processing was mechanized with the development of vertical axis water turbines known as *pani ghattas* which can produce powers up to about 2kW from 2m-4m of head. This wooden machine has flat plates and runs on a vertical axis, driving the millstones located directly above it, see *Figure B.1*. Today there are an estimated 20,000 small mills still in operation.

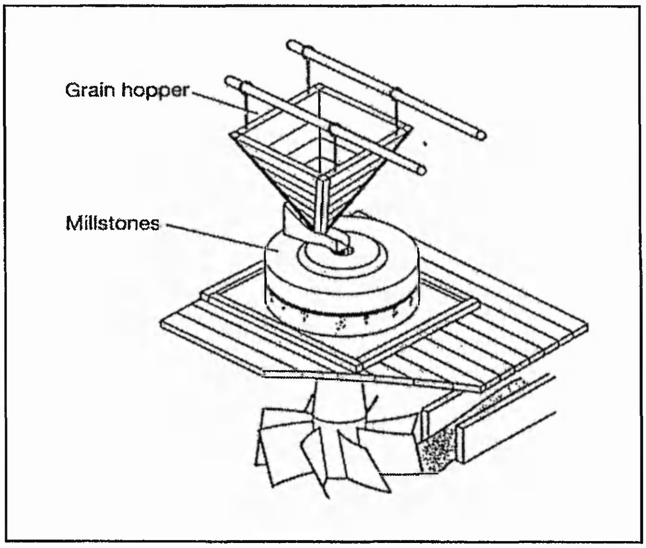


Figure B.1 : The Nepalese pani ghatta.

Towards the end of 1960s, two organizations, the United Mission to Nepal and Balaju Yantra Shala, started the development of more suitable and more efficient low cost turbines. Nepalese manufacturers have worked side by side with foreign experts in developing low cost, suitable technology for poor rural communities. Work began with up grading the pani ghattas which were made of wood, to higher performance steel machines with shaped runners [Brown, 1992; Hislop, 1987a & b and Shrestha, 1989]. Moreover, the aid agencies introduced metal cross flow turbines, which could produce greater power outputs for the same site where a pani ghatta was previously used.

Akkal Man Nakarmi[1] developed an intermediate solution to the high efficiency/low cost problem by designing a metal turbine based on the traditional ghatta blades, which is well know as the multi purpose power unit (MMPU), shown by *Figure B.2*, which has been widely used for driving agro-processing machinery. Today, more micro hydro turbines are made in Nepal than probably any other country, excluding China. Local demand and appropriate low cost micro hydro technology has encouraged many Nepalese workshops through their own applied experience to create small industries at the forefront of micro hydro technical adaptation [Cromwell, 1992 and Oram, 1995]. The principle role of external technical assistance has been in training and providing certain know how with regard to the design skills involved by technology transfer.

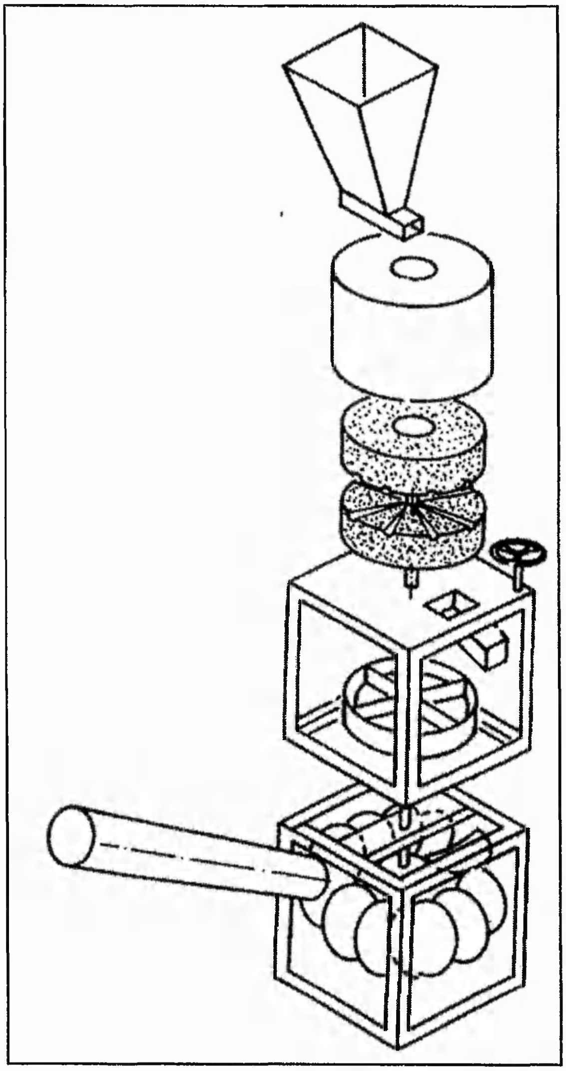


Figure B.2: A Nepalese multipurpose power unit.

[1] Owner of the Katmandu Metal Industries LTD, Cha3-812 Nagal Quadon, Chhetrapati, Katmandu 3, Nepal.

According to Brown,

> *"There are now at least ten small workshops in Nepal that are manufacturing and installing micro hydro plants. There are about 750 plants in operation and about 50 new projects are implemented annually. All the expertise and about 85% of hardware comes from within Nepal"* [Brown, 1992].

Cross flow turbines are dominating the medium head sites. More recently, there has been a growing interest in the use of Pelton wheels for those sites with relatively high heads and low flow rates, although the use of two or more jets enables their use on some sites where cross flow turbines can be used. According to Smith,

> *"An important contribution to the promotion of Pelton wheels was the introduction of Peltric sets by Katmandu Metal Industries which are very small machines (1-5 kW power) and have the Pelton wheel directly coupled on the shaft of an induction generator*" [Smith, 1992a].

Figure B.3 shows a typical Peltric set. Pelton wheels are considered to be the simplest type of turbine [Waltham, 1993], have high efficiencies and are more reliable than cross flow turbines. However, unlike cross flow turbines, not every part of a Pelton wheel can be fabricated or welded together out of standard sizes of steel sheet or tube. Pelton buckets are an intricate shape and the only feasible method for manufacturing them is casting.

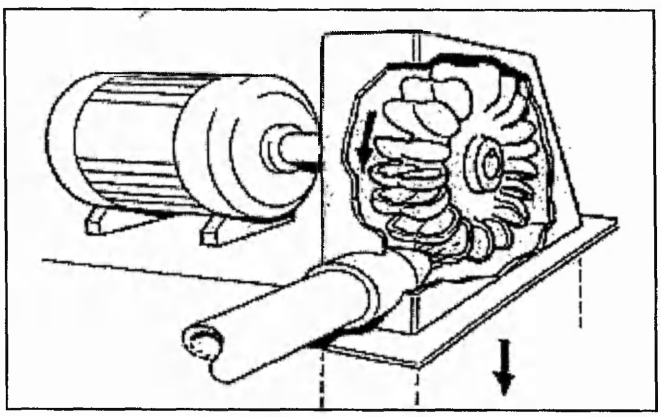


Figure B.3 : A typical Peltric set.

B.1.1.2 Micro hydro in Peru

In many of the Latin and South American countries, water wheel technology was introduced by the Spanish in the 16^{th} and 17^{th} centuries. By the beginning of the 20^{th} century, locally manufactured Pelton wheels were introduced to harness the power available from medium and high head sites, which are mainly located in the Andes mountains [Burton, 1988 and Viani, 1993]. The power from low head sites has been utilized by means of horizontal and vertical axis water wheels made of wood or steel.

ITDG has been working in training and technical assistance for local installation of micro hydro schemes and in research into methods of increasing the access of rural people to the benefits which micro hydro can bring, particularly through low cost innovations in supply and use of energy, since 1985. Work is carried out in the north of the country, in the Cusco and Cajamarca areas [Fisher, 1994]. These projects have used Pelton and cross flow turbines manufactured in Cajamarca and have been installed, maintained and managed by local people with support from ITDG. Moreover, a pilot project on the design and development of a low head propeller turbine has recently been initiated [Viani, 1993] which is described in more detail in section 2.3.8 of Chapter 2.

B.2 Report on the field trip to Sri Lanka

A three week visit to Sri Lanka - March 1994 - was organized to investigate the suitability of a low head axial flow turbine which can provide relatively cheap power for remote communities. In addition, various Peltric sites were visited in rural areas to get familiar with their characteristics, problems faced during their installation, operation and how these are dealt with. Information was also gathered on the existing workshop and manufacturing facilities of local industries. Finally, the installation and testing of a prototype axial flow machine designed and manufactured locally was carried out.

B.2.1 Introduction

The island of Sri Lanka is located south of India between 6° and 10° north of the equator. With an area of 66000 km^2, Sri Lanka is divided into three topographical regions. The *mountainous region* is mainly located in the central southern part with altitudes above 1000 m. This is then surrounded by the *upland region* with altitudes varying from 300 to 1000m.

The rest of the island and mainly the northern part forms the *flat region*. Rainfall in Sri Lanka varies from 1 to 4m per calendar year. The May - August *monsoon* brings rain to the south west and central regions whereas the north and east areas have rain mainly during the November - January *monsoon*. 93% of the country's electricity is produced by large scale hydro electric schemes, some of which are said to be facing technical and silting problems.

Sri Lanka has a population of about 17.5 million, 70% of which live in the rural areas. However, only 30% of the population has access to mains electricity even though the majority of the people live within a few km away from grid lines. However, with the majority of the houses in the rural areas made of coconut wood and sun dried clay and with roofs made of coconut tree leaves, the Ceylon Electricity Board (CEB), consider these dwellings unsuitable for mains power supply for safety reasons.

As an alternative energy source for these rural communities biomass and particularly fuel wood is used, mainly for cooking. In the tea estates, fuel wood is used also used for the drying process of tea.

B.2.2 Micro hydro in Sri Lanka

Micro hydro in Sri Lanka was introduced during the 1920's in the tea estates mainly to provide power for the processing and drying of tea. The number of micro hydro schemes installed on tea estates in Sri Lanka during the first half of this century is estimated at about 500 [Ariyabandu, 1994]. However, most of these schemes fell into disuse as a result of the introduction of large scale hydro electric schemes and the extension of the national grid alongside the use of diesel generators for electricity generation. The rising prices of fuel oil and electricity and the interest for utilizing renewable energy sources to avoid extreme deforestation, have contributed to the revival of micro hydro schemes.

In the early 1980s however, rising electricity and diesel prices, and to a lesser degree, the interest of the rural communities in decentralized renewable energy sources, has led to attempts to revive micro hydro power generation [Carrasco, 1993].

However, the majority of the projects ended in failure with the main reason being the use of technology that was imported into the country; local communities were forced to use equipment without building up local capabilities for their correct installation, operation and maintenance.

ITDG became involved in a number of projects in order to develop local technical capabilities to manufacture, repair, operate and maintain equipment for power generation in rural areas. The tea estates became ideal demonstration sites. ITDG has set up a training unit in Sarasavigama, a site a few miles outside the mountain city of Kandy. The aim of this center was to train local workshops to make turbines from locally available materials and encourage local manufacturers to test their machines [Ariyabandu, 1994]. *Picture B.1* shows the Sarasavigama training centre.

Picture B.1 : The Sarasavigama training center, Kandy - Sri Lanka.

Since 1991, ITDG initiated its micro hydro program in the hilly areas of the Matara District working closely with local innovators and manufacturers. The majority of the schemes in Sri Lanka are domestic type using locally made Peltric sets. The work so far in Sri Lanka has proved that the main factors which determine the viability of a micro hydro scheme are *affordability*, *availability* and *durability* of the machinery used.

In addition, the careful consideration of the management structure and social and cultural implications of the project are equally important [Carrasco, 1993]. *Picture B.2* shows the opening of Isahawatte Dola scheme in the Akuressa district.

Picture B.2 : Official opening of Isahawatter Dola scheme in the Akuressa district.

B.2.3 Turbine technology for micro hydro sites in Sri Lanka.

Most micro hydro sites in the tea estates have high heads (greater than 30m). For such heads, Pelton wheels directly coupled to induction generators have been used for a number of years now. Peltrics require only small flow rates to run as opposed to axial or Francis turbines, and come with a number of jets that accommodate the varying flow rates all year round or power demand between day and night time. The power produced is in the range of 1kW to 3kW, and supplies a village of about 20 houses, with 100W allocated per house. Typical loads are electric light bulbs of 40W, 60W or even 100W. With the encouragement to use low wattage - long life fluorescent lamps as opposed to the high wattage incandescent lamps, more power is available for each house household. During daytime, the power is sometimes used to run a TV or a radio, and sometimes even water heaters.

Most of the design work for a particular scheme is carried out by the ITDG engineers in Sri Lanka. The simplicity of the design and installation, make it possible to use Peltric units anywhere in a remote community. With the houses being a few hundred meters apart, transmission costs are minimum. For those houses for which connection would be too costly as a result of distances, batteries are used for power supply which are charged during daytime. With the entire Peltric technology being available in the country itself and the fact that the majority of the equipment such as turbine runner and casing, induction generator controller (IGC) and civil works are locally made, costs are kept to an affordable level.

B.2.4 Turbine manufacturing industry in Sri Lanka.

All Peltric systems are now made in Sri Lanka by local workshops. Bronze casting is used for making each bucket individually, which is then assembled to the main wheel using bolts and nuts. This technique makes it easier to change individual buckets which get damaged during operation, mainly by the eroding action of silt. The casings and nozzles of the Peltric system are also made and assembled together locally. The policy of ITDG is to use the skills and facilities of small local industries found scattered round the remote areas. The benefits are numerous. Small communities are given jobs which provide a reasonable income to live on. Designs tend to be simple and less costly.

Local people are also trained in the making of turbines which means that there will be more people capable of servicing and maintaining these units in the future. Moreover, if the turbines are made locally, repairs can take place locally, thus the time that a unit is out of operation is kept to a minimum.

B.2.5 Installation-testing of the low head axial flow turbine.

According to the ITDG office in Colombo, there is an increasing interest by owners of low head sites in utilizing power at heads below 5m and with relatively high flow rates. At the moment, cross flow turbine are used which tend to be bulky and heavy and require a speed increasing mechanism to drive a generator as a result of their low operations speeds.

B.2.5.1 The turbine

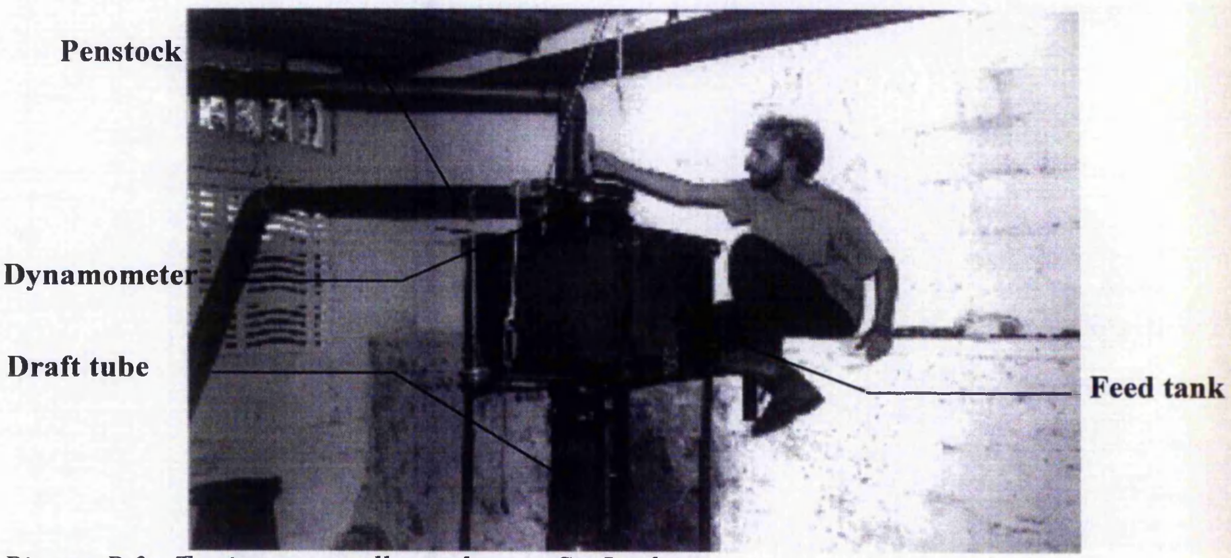
Chris Taylor, a student from Warwick University spent a period of 10 months in Sri Lanka doing voluntary work. His work involved the design and manufacturing of a prototype low head axial flow turbine, its installation and testing at a site near Sarasavigama village used by ITDG for training and testing purposes. During the three week period in Sri Lanka some time was taken to study the design of the turbine and assist Chris with the turbine installation and testing.

The turbine was an open flume design with a feed tank on top of the five guide vanes which were arranged axially. The four runner blades were designed to rotate at 1000rpm with a head of 2m and a flow rate of $0.05m^3/s$.

The runner tip and hub diameters were 200 and 150 mm respectively. For the lower bearing of the turbine shaft, a wooden bearing was used which has previously been soaked in oil. The top end of the shaft was supported by a self aligning thrust bearing. The method used to design this particular turbine, suggests that emphasis was given on low cost rather than performance. The blades of this particular design were made by twisting and bending flat sheets of metal to the desired shape which were then welded onto the hub. The direction of the relative velocity component was used as the basis for the design of the runner blades. This method has been explained in Chapter 3. Twisting of the blades was kept minimum by using a hub to tip ratio of 0.75 and a speed of 1000rpm. An assumption was made for a design hydraulic efficiency of 85%.

B.2.5.2 Testing of the turbine

The test site at Sarasavigama tea estate has been set up by ITDG specifically for demonstration and testing of micro hydro units. A Pelton turbine, with 100mm PVC penstock pipe feeding from a near by stream, was already installed which provided electrical power for the site's needs. In order to provide enough flow rate for testing the propeller turbine, another feed tank was used while tests were carried out. Flow rate measurements were carried out with the aid of a V - notch weir, which was built into the outlet channel of the propeller turbine. The power output from the turbine was measured with a torque drum, see *Picture B.3,* made of a car wheel rim directly coupled to the turbine shaft and loaded with a spring balance and weights.



Picture B.3 : Testing a propeller turbine in Sri Lanka.

Table B.1 shows some of the test results.

Turbine speed (rpm)	Flow rate (l/s)	Power output (W)	Efficiency (%)
880	36.7	197	31
810	36.7	200	32
750	37.5	214	33

Table B.1 : Test results of the propeller turbine for a constant head of 1.75m.

B.2.5.3 Conclusions from the propeller turbine installation

This particular prototype has produced a maximum efficiency of 33%. Inefficiencies occurred due to the simplicity of design. The 90° sharp intake at the bottom of the feed tank, located just above the guide vanes, was considered to be the major cause of inefficiency. Sharp edges are known to cause considerable whirl in the water, eddies and sharp rises in velocities. As a result of these disturbances in the incoming flow, the guide vanes were not performing as they were designed to. Moreover, as it has been explained in Chapter 4, any mismatch at the inlet of the runner would cause increased shock losses at the blades leading edges and velocity head losses at the draft tube exit. As a result of using an open flume layout, the turbine shaft had to be 0.5m long to keep the water drum out of reach of water. Such an arrangement produced excessive vibration during testing. Moreover, air was drawn into the turbine guide vanes from the top of the tank.

B.3 Report on the site visit at Pedley Wood Conservation Trust - Addlington, Peak District, UK

Pedley Wood is a privately owned conservation trust in Cheshire. The site is located on a stream leaving moorland to enter the Cheshire plain. The trusts objectives are mainly related to the promotion of renewable energy from wind and low head hydro power by setting up projects closely related to local schools, universities and conservation enthusiasts. As a registered charity, the trust is able to generate money that facilitate the setting up of the different projects.

The Pedley Wood site is well known for its low head water wheel, a project that was initiated back in 1990. The wheel was designed to run at 12rpm with a flow rate of $0.07m^3/s$ and a head of 2m. The water wheel buckets were made of plywood and are mounted on oak spokes. The high torque low speed characteristic of the wheel is converted to a low torque high speed power by the combined use of a Ferguson tractor back axle and the integral gearbox of an AC induction motor which is run as a generator. The AC induction motor and integral gearbox has been obtained from Brown Pestell ltd. The generating speed is 1600rpm and produces 240 volts at 50 Hz single phase.

The above site is also used as a demonstration site for low head propeller turbines. A prototype 1kW propeller turbine was designed and constructed by a student of the Mechanical Engineering Department of Nottingham Trent University [Heitz, 1993]. The turbine has been installed and tested by another student of Nottingham Trent University alongside with the water wheel [Mali, 1994]. The prototype has four runner blades which are made of constant thickness stainless steel plates bent and twisted to the desired angles and is running at 2100rpm. The design head and flow rate are 2.9m and $0.06m^3/s$ respectively. Tests of this particular design suggest that it is very inefficient. According to Mali,

> *"The turbine overall efficiency and performance was very poor over a certain range of operating speeds and flow rates. Though a turbine was an open flume design, the tank feeding the turbine was not big enough and therefore the head varied a lot during testing and at occasions air used to be drawn into the turbine inlet*" [Mali, 1994].

The efficiency of this particular design has never exceeded 20%, and according to the literature available, there have been a considerable amount of frictional losses due to the high velocities of water and incorrect selection of bearings. Moreover, there have been problems with blocked guide vanes either because of leaves or plastic bags.

The design methodology of this turbine was based on the assumption that the coefficient of lift would remain constant across the whole blade section, an approach usually adopted in fan designs [Bohl, 1991]. As a result, the blades were developed with long chord lengths at the hub section and very small at the tip.

The result was a low hydraulic efficiency of the turbine. Moreover, a very long transmission shaft was required due to the open flume arrangement, which experienced severe vibrations during testing.

During an open day at Pedley Wood - June 1994 - a great opportunity was given to talk to other people involved with micro hydro, exchange ideas and experiences and investigate the design considerations of low head axial flow turbines.

B.4 Report on the site visit at River Wandle micro hydro Project - South London, UK

In the South - West of London, the River Wandle used be one of the most important energy sources, providing power for nearly twenty mills to produce corn, snuff and many other products during the industrial revolution [Williams, 1995b]. Back in 1992, Platform, an interdisciplinary arts and ecology group, set out to initiate a number of projects aiming to the revival of water power schemes in the London area. In 1993, near the Delta of River Wandle, a site was chosen for the installation of a low head propeller turbine. The drop at this point is about 2 meters and the scheme has been incorporated into the existing sluice gate arrangement, *Figure B.4*, which was initially installed 60 years ago.

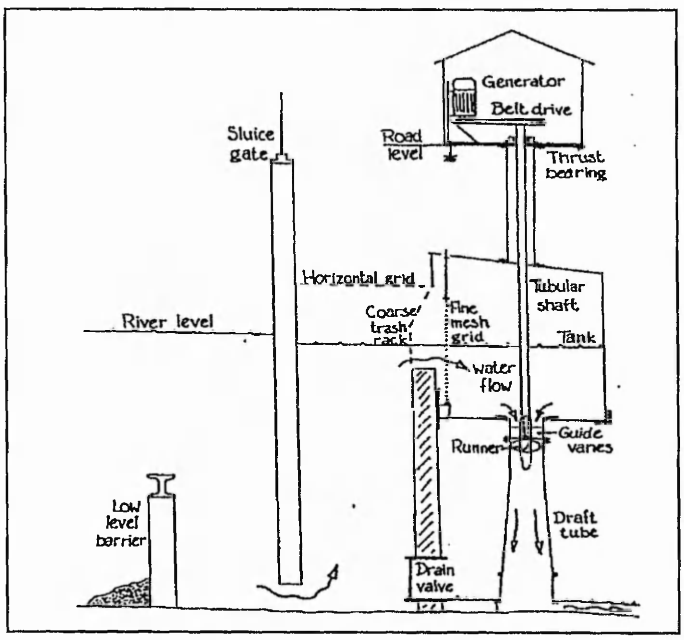


Figure B.4 : The low head site in London.

The open flume layout was used, with fixed geometry axial guide vanes. The turbine was mounted vertically inside a tank. It has fixed geometry, four runner blades made of constant thickness sheet of metals and runs at 650rpm. With the aid of belt drive, an induction generator was able to produce single phase power at 240 volts AC. According to Williams [1995b] the performance of this turbine has been very poor. Though proper trash rack grills were fitted to the intake of the tank to make sure that leaves would not enter the turbine, the maximum efficiency of the machine was only 24%.

A two day visit - April 1994 - was organized to investigate the causes of excessive vibration and noise during the operation of this low head axial flow turbine, a design similar to the one installed at Pedley Wood Trust site. After dismantling the turbine unit, signs of wear on the shaft were noticed and the water lubricating bearing located at the bottom of the turbine was totally damaged. The shaft also seemed to be badly bent due to the imbalance loads exerted on it during 30 days of operation. It was evident that the shaft had to be replaced with a new one which would have a thrust bearing at the top in order to take all the torsional vibrations and thus allow a smoother operation of the turbine.

A new runner has been designed but has yet to be tested. The power produced is currently been used by the nearby primary school of St. Joseph, for battery charging and low wattage lighting for some of the rooms in the school building.

B.5 Conclusions from the field trips and site visits

The conclusions from this field trip can be summarized as follows :

☑ The key elements to a successful scheme are minimum maintenance requirements, durability, reliability and cheap spare parts locally available. Since every village is responsible to maintain and service its own plant, the maintenance procedures have to be as simple as possible. In case spare parts are required, these should be available locally in order to minimize the time the community is without power and to make sure that the costs are kept low.

☑ The participation of the whole village in a micro hydro project is of great importance. This involves the measurement of available head, flow rates and carrying out the majority of the civil works.

☑ Difficulties in transportation over rough terrain make the use of minimum civil works essential for cost reduction.

☑ Lightweight turbine sections are easier to transport rather than complete turbines.

☑ Need to know how much water is available before carrying out the design of a turbine unit. Over sized schemes operate at low efficiencies whereas undersized ones do not utilize the power available from a particular site.

☑ The cost of micro hydro schemes should be kept minimum. The majority of the expenses is covered by charity organizations like for example the Rotary Club. However, it is often the case that the village communities contribute a considerable amount to the total cost of a project.

☑ Simplicity in the design and simplified flow areas would result in decrease in overall efficiency of a propeller turbine.

☑ The use of open flume layout requires long turbine shafts that are likely to experience vibration if incorrectly balanced. Moreover, when the size of the tank is small, air is drawn into the turbine.

☑ It is crucial to assemble the turbine in the workshop before is send out to the fields in order to make sure that every thing fits together.

☑ Blade geometry is affected by thermal distortion due to welding. One option is to use thicker plates for fabricating the runner blades, which would then make bending and twisting more difficult. However, blade twist can be minimized by using higher hub to tip ratios as well as lower operating speeds.

Appendix C : Induction Motors as Generators

C.1 Sample calculation for a 4 pole induction generator

The shafts of synchronous generators turn at a fixed speed which is known as synchronous speed. For a 4 pole generator at 50Hz the synchronous speed is given by :

$$N_{synch.} = \frac{120}{p} f = \frac{120}{4}\ 50 = 1500\ rpm \tag{C.1}$$

Shafts of induction machines do not turn at synchronous speed due to slip given by :

$$s = \frac{N_{synch.} - N_m}{N_{synch.}} \tag{C.2}$$

The motor speed is therefore given by :

$$N_m = \frac{120}{p} f(1-s) \tag{C.3}$$

Slip is approximately the same for both motor and generator mode. *Figure C.1* shows the torque - speed characteristics for induction motors and generators [Bell, 1995]. When a motor is run as a generator, the negative value of slip is taken [Smith, 1996 and Williams, 1995a]. For a 4 pole induction generator (induction motor rated at 1450 rpm, 50Hz) the generating speed is given by

$$N_g = \frac{120}{p} f(1+s) = \frac{120}{4}\ 50\ (1.0333) = 1550\ rpm \tag{C.4}$$

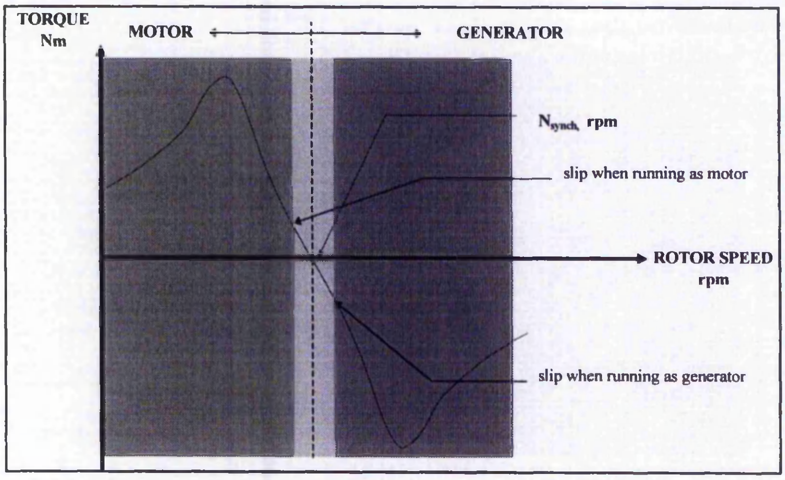


Figure C.1 : Torque - speed characteristics of induction machines on the grid.

Appendix D : Useful Addresses - Contacts

D.1 Turbine manufacturers

America

Turbinas Hydraulicas
✉ Calle M. Eterovic 1241,
Casilla : 2083,
Cochabamba,
Bolivia.

Canyon Industries ltd.
✉ 5346 Mosquito Lake Rd,
Deming WA 98244,
USA.

Asia

Jyoti ltd., International Division
✉ 103 Kakad Chambers,
132 Dr. Annie Besant Rd,
Bombay-400 018,
India.

Katmandu Metal Industries
✉ P.V.T. ltd.,
Cha3-812 Nagal Quadon,
Chhetrapati, Katmandu 3,
Nepal.

Australia

TAMAR DESIGNS PTY ltd.
✉ Deviot, Tasmania 7275,
Australia.

Europe

Gugler
✉ A-4085,
Niederranna 41,
Austria.

CINK m.v.e.
✉ Chebská 48,
CZ - 36006 Karlovy Vary,
Check Republic.

AWT Limited
✉ PO BOX 10,
Tetbury,
Gloss.,
GL8 8RA,
England.

Biwater Hydropower
✉ Clay Lane, Old Bury,
Warley, West Midlands,
BG9 4TF,
England.

Turbines Hydrauliques EMTH
✉ 27500 Tourville-Sur-Pont-Audemer,
France.

Ossberger-Turbinenfabrik GmbH & Co.
✉ Otto-Rieder St. 7,
D-8832 Weissenburg I. Bay,
Germany.

Ganz-Mavag
✉ Ganz Machinery & Energy Co. ltd.,
H-1087 Budapest, VIII,
Kõbanyai út 21,
H 1475 Budapest, PO BOX 276,
Hungary.

Orengine s.r.l
✉ Via Gretodi Cornigliano, 6 r.,
Genova,
Italy.

MacKellar Engineering ltd.
✉ Strathpey Industrial Estate,
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Railway Terrace,
Rugby,
CV21 3HT,
UK.

ETSU
✉ Harwell, Didcot,
Oxfordshire,
OX11 0RA,
UK

International Water Power & Dam Construction (Journal)
✉ Quadrant House,
The Quadrant,
Sutton,
Surrey SM2 5AS, UK.

Library & Information services
✉ Institution of Civil Engineers,
Great George St.,
Westminster, London
SW1P 3AA,
UK.

Wilson Energy Associates ltd.
✉ Engineering Consultants,
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Bramhall, Stockport, Cheshire,
SK7 1AH, UK.

Mr. V. Meier
✉ SKAT,
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CH-9000 St. Gallen,
Switzerland.

Mr. R Metzler
✉ Stepham Blattmann Str. 11,
D 7743 Furtwagen,
Germany.

Appendix E : Mechanical Losses

E.1 Mechanical efficiency calculations

The bearings of the propeltric unit are subjected to *axial* and *radial forces*. *Axial forces* act along the shaft of the turbine/generator set and they are caused by the hydraulic thrust of the water onto the turbine runner plus the weight of the runner. *Radial forces* are caused by out of balance masses which can be due to incorrect attachment of the runner blades onto the runner hub and/or incorrect dynamic balancing of the runner. RHP *cast iron self lube* bearing units were used for the prototype setup. The frictional torque given by the RHP Technical Handbook is given by

$$T_{frictional} = 0.5\,\mu\,P\,d$$

The equivalent load P is given by

$$P = XF_r + YF_a$$

The axial force is

$$F_a = F_{Hydraulic} + F_{Weight}$$

$$F_{Hydraulic} = \frac{\pi}{4}\left[D_T^2 - D_H^2\right]\rho\,g\,H_N = \frac{\pi}{4}\left[0.15^2 - 0.06^2\right]9.81\times10^3\times 2 = 291.24\ N$$

$$F_{Weight} = g\left[mass\ of\ runner + mass\ of\ nose\ cone\right] = 9.81\times 2 = 19.62\ N$$

$$F_a = 19.62 + 291.24 = 313.86\ N$$

$$F_r = \left[mass\ out\ of\ balance\right]\omega^2 R = 0.25\left[\frac{2\,\pi\,1560}{60}\right]^2 0.03 = 200.16\ N$$

Using the data from the RHP Handbook

$$\begin{aligned} P &= 0.56\,F_r + 1.768\,F_a \\ &= 0.56\times 200.16 + 1.768\times 313.16 = 665.75\ N \end{aligned}$$

$$T_{frictional} = 0.5\times 0.003\times 665.75\times 0.02 = 0.02\ N$$

The power lost due to bearing friction is therefore **3.26W**.

Appendix F : Design Calculations

F.1 Hydraulic design of the propeltric unit

The design details from the hydraulic design of the turbine are presented below. The calculations were performed with the aid of Microsoft Excel spreadsheet.

SITE INPUT SPECIFICATION

H, (m)	Q, (m^3/s)	N, (rpm)	Ns, (min^{-1})
2	0.072	1560	248.896

RUNNER DIMENSIONS

K_{uT}	D_H / D_T	D_T, (m)	D_H, (m)
1.950	0.400	0.150	0.060

Z	u_H, (m/s)	u_T, (m/s)	V_m, (m/s)
4.000	4.887	12.216	4.879

n_V, (%)	n_M, (%)	n_H, (%)	n_O, (%)
94.500	98.000	75.200	69.643
P_{IN}, (w)	P_{OUT}, (w)	gH	gH_E
1412.640	983.801	19.620	14.754

VELOCITY CALULATIONS "TIP" SECTION

D_T, (m)	V_1, (m/s)	α_1, (deg.)	W_{u1}, (m/s)
0.150	5.026	76.095	11.008
u_T, (m/s)	V_2, (m/s)	α_2, (deg.)	W_{u2}, (m/s)
12.216	4.879	68.230	12.216
V_{uT1}, (m/s)	W_1, (m/s)	β_1, (deg.)	ΔW_u, (m/s)
1.208	12.041	23.902	1.208
V_{uT2}, (m/s)	W_2, (m/s)	β_2, (deg.)	$\beta_1 - \beta_2$, (deg.)
0.000	11.200	21.770	2.132
V_m, (m/s)	W_{oo}, (m/s)	β_{oo}, (deg.)	t_T, (m)
4.879	12.595	22.789	0.117

VELOCITY CALULATIONS "HUB" SECTION

D_H, (m)	V_1, (m/s)	α_1, (deg.)	W_{u1}, (m/s)
0.060	5.737	58.248	1.867
u_H, (m/s)	V_2, (m/s)	α_2, (deg.)	W_{u2}, (m/s)
4.887	4.879	45.047	4.887
V_{uH1}, (m/s)	W_1, (m/s)	β_1, (deg.)	ΔW_u, (m/s)
3.019	5.224	69.055	3.019
V_{uH2}, (m/s)	W_2, (m/s)	β_2, (deg.)	$\beta_1 - \beta_2$, (deg.)
0.000	6.905	44.953	24.102
V_m, (m/s)	W_{oo}, (m/s)	β_{oo}, (deg.)	t_H, (m)
4.879	5.933	55.309	0.047

VELOCITY CALULATIONS "A" SECTION

D_A, (m)	V_1, (m/s)	α_1, (deg.)	W_{u1}, (m/s)
0.120	5.107	72.806	8.264
u_A, (m/s)	V_2, (m/s)	α_2, (deg.)	W_{u2}, (m/s)
9.773	4.879	63.472	9.773
V_{uA1}, (m/s)	W_1, (m/s)	β_1, (deg.)	ΔW_u, (m/s)
1.510	9.596	30.557	1.510
V_{uA2}, (m/s)	W_2, (m/s)	β_2, (deg.)	$\beta_1 - \beta_2$, (deg.)
0.000	10.923	26.528	4.029
V_m, (m/s)	W_{oo}, (m/s)	β_{oo}, (deg.)	t_A (m)
4.879	10.254	28.412	0.094

VELOCITY CALULATIONS "B" SECTION

D_B, (m)	V_1, (m/s)	α_1, (deg.)	W_{u1}, (m/s)
0.090	5.278	67.580	5.317
u_B, (m/s)	V_2, (m/s)	α_2, (deg.)	W_{u2}, (m/s)
7.330	4.879	56.353	7.330
V_{uB1}, (m/s)	W_1, (m/s)	β_1, (deg.)	ΔW_u, (m/s)
2.013	7.216	42.537	2.013
V_{uB2}, (m/s)	W_2, (m/s)	β_2, (deg.)	$\beta_1 - \beta_2$, (deg.)
0.000	8.805	33.647	8.891
V_m, (m/s)	W_{oo}, (m/s)	β_{oo}, (deg.)	t_B (m)
4.879	7.987	37.650	0.070

BLADE GEOMETRY CALCULATIONS

SECTION : TIP

t/c	0.800	0.900	1.000	1.200	1.300	1.400	1.500	1.600
c, (m)	0.147	0.131	0.117	0.098	0.090	0.084	0.078	0.073
C_L	0.153	0.172	0.191	0.229	0.249	0.268	0.287	0.306
μ	0.500	0.450	0.400	0.325	0.280	0.255	0.225	0.215
θ, $(\beta'_2-\beta'_1)$, (deg.)	4.252	4.724	5.314	6.541	7.592	8.336	9.448	9.887
R	1.979	1.584	1.267	0.858	0.682	0.577	0.475	0.426
f	0.001	0.001	0.001	0.001	0.001	0.002	0.002	0.002
f/c	0.009	0.010	0.012	0.014	0.017	0.018	0.021	0.022

SECTION : A

t/c	0.800	0.900	1.000	1.200	1.300	1.400	1.500	1.600
c, (m)	0.117	0.104	0.094	0.078	0.072	0.067	0.063	0.059
C_L	0.235	0.264	0.294	0.352	0.382	0.411	0.440	0.470
μ	0.550	0.490	0.440	0.360	0.325	0.300	0.280	0.250
θ, $(\beta'_2-\beta'_1)$, (deg.)	7.303	8.197	9.129	11.158	12.359	13.389	14.346	16.067
R	0.922	0.730	0.590	0.403	0.336	0.288	0.251	0.210
f	0.002	0.002	0.002	0.002	0.002	0.002	0.002	0.002
f/c	0.016	0.018	0.020	0.024	0.027	0.029	0.031	0.035

SECTION : B

t/c	0.800	0.900	1.000	1.200	1.300	1.400	1.500	1.600
c, (m)	0.088	0.078	0.070	0.059	0.054	0.050	0.047	0.044
C_L	0.402	0.452	0.503	0.603	0.653	0.704	0.754	0.804
μ	0.585	0.530	0.490	0.410	0.375	0.350	0.325	0.300
θ, $(\beta'_2-\beta'_1)$, (deg.)	15.149	16.721	18.086	21.615	23.632	25.320	27.268	29.540
R	0.334	0.269	0.224	0.157	0.132	0.115	0.100	0.086
f	0.003	0.003	0.003	0.003	0.003	0.003	0.003	0.003
f/c	0.033	0.036	0.039	0.047	0.051	0.055	0.059	0.064

SECTION : HUB

t/c	0.800	0.900	1.000	1.200	1.300	1.400	1.500	1.600
c, (m)	0.059	0.052	0.047	0.039	0.036	0.034	0.031	0.029
C_L	0.812	0.913	1.015	1.218	1.319	1.420	1.522	1.623
μ	0.640	0.580	0.550	0.480	0.450	0.420	0.380	0.360
θ, $(\beta'_2-\beta'_1)$, (deg.)	37.530	41.413	43.672	50.041	53.377	57.189	63.209	66.721
R	0.091	0.074	0.063	0.046	0.040	0.035	0.030	0.027
f	0.005	0.005	0.004	0.004	0.004	0.004	0.004	0.004
f/c	0.080	0.088	0.093	0.106	0.112	0.120	0.131	0.137

F.2 Load calculations for generator bearings

The design head for the propeltric unit is 2m. The bearings of the generator would be subjected to an axial thrust load due to the hydraulic force of the water on the runner which combined with the weight of the runner would give a total axial load as follows :

$$F_a = F_{Hydraulic} + F_{Weight}$$

$$F_{Hydraulic} = \frac{\pi}{4}\left[D_T^2 - D_H^2\right]\rho\, g\, H_N = \frac{\pi}{4}\left[0.15^2 - 0.06^2\right] 9.81\times 10^3 \times 2 = 291.24\ N$$

$$F_{Weight} = g\left[mass\ of\ runner + mass\ of\ nose\ cone\right] = 9.81\times 2 = 19.62\ N$$

$$F_a = 19.62 + 291.24 = 313.86\ N$$

Using information supplied by various small induction motor manufacturers[1], the value of this axial force is well within the specified ranges of standard bearings used for vertical mounting of induction motors.

F.3 Structural design of the runner blades

The assumption made for the structural design of the blades is that each blade resembles a simply supported cantilever with hydraulic thrust acting as shown in *Figure F.1*. The blades are treated as if there is no twist.

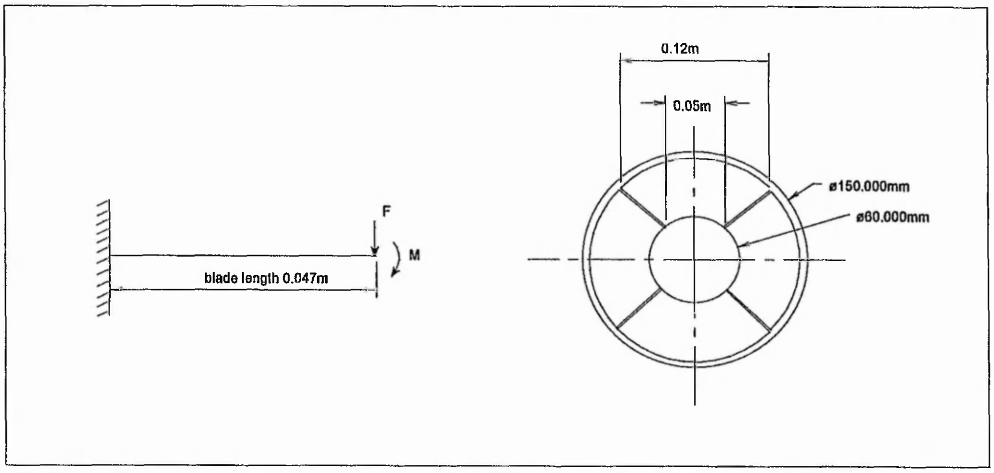


Figure F.1 : Structural design of the runner blades.

[1] GEC Small Machines Ltd. give thrust load limits for 2 pole induction motors, vertically mounted, of 660.3N and 649.4N for 0.75 and 1.1kW machines respectively.

Assuming that the maximum turbine operating head would be 4m, the hydraulic thrust is given by

$$F_{Hydraulic} = [\rho\, g\, H_N] \times \frac{\pi}{4}[D_T^2 - D_H^2] \approx 582.5N$$

The force on each blade is therefore equal to 145.625N and the moment is equal to 6.8Nm. The permissible direct stress is given by

$$\sigma_p = \frac{\text{Ultimate Yield strength of mild steel}}{\text{safety factor}} = \frac{215}{3} = 71.67MPa$$

The maximum stress for the blade is given by

$$\sigma_{mac} = \frac{M}{\frac{l\,t^2}{6}} = \frac{6 \times 6.8}{0.047\ t^2}\ \text{MPa}$$

Equating the maximum stress with the permissible stress, the blade thickness is 3mm.

F.4 Structural design of the turbine shaft

When a circular shaft is subjected to longitudinal thrust or tension, as well as twisting, the direct stresses due to longitudinal load must be combined with the shearing stress due to torsion in order to evaluate the principal stresses in the shaft. The shaft is loaded axially in tension due to the hydraulic thrust and the weight of the runner by a longitudinal direct stress σ . The shearing stress at any point is denoted as τ. The principal stresses in the shaft are given by $\frac{1}{2}\sigma \pm \frac{1}{2}\sqrt{\sigma^2 + 4\tau^2}$ and the maximum shearing stress is given by $\tau_{max} = \frac{1}{2}\sqrt{\sigma^2 + 4\tau^2}$.

For the design of the turbine shaft, it is assumed that the turbine is operating at the full capacity of the test rig, i.e. 6m if head and a flow rate of 0.14m^3/s. The power available on the turbine shaft is 5.8 kW (70% turbine overall efficiency). The torque for a design speed of 1560 rpm is

$$T = \frac{Power}{rotational\ speed} = 35.5\ Nm$$

The shaft is made of mild steel with the tensile strength (s_u) equal to 430 MPa and the yield strength (s_y) being 215 MPa. Using a safety factor of 3, the permissible stress is

$$\sigma_p = \frac{215}{3} = 71.67\ MPa$$

The hydraulic thrust is given by $\left[\rho\, g\, H_N\right] \times \frac{\pi}{4}\left[D_T^2 - D_H^2\right] \approx 874N$

For a shaft diameter of 0.02m

$$\tau = \frac{16\,T}{\pi\, d^3} = \frac{16 \times 35.5}{\pi \times 0.02^3} = 22.6MPa \text{ and } \sigma = \frac{Thrust}{\frac{\pi\, d^2}{4}} = \frac{874}{\frac{\pi \times 0.02^2}{4}} = 2.8MPa$$

The maximum shearing stress becomes

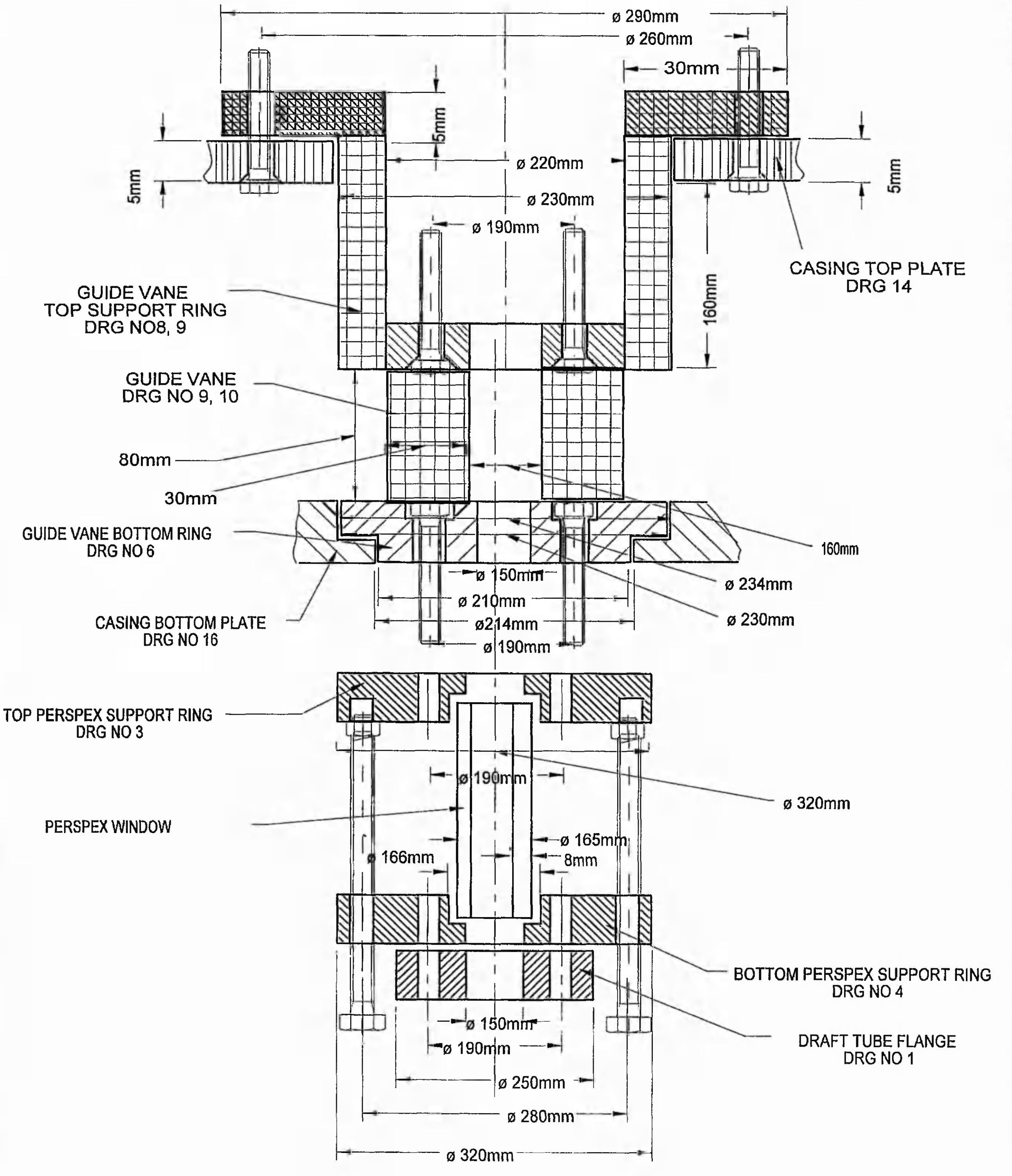
$$\tau_{max} = \frac{1}{2}\sqrt{(2.8E6)^2 + 4 \times (22.6E6)^2} = 22.64MPa$$

Since $\sigma_p > \tau_{max}$ the value of 0.02m for the shaft diameter can be safely used.

Appendix G : Design Drawings

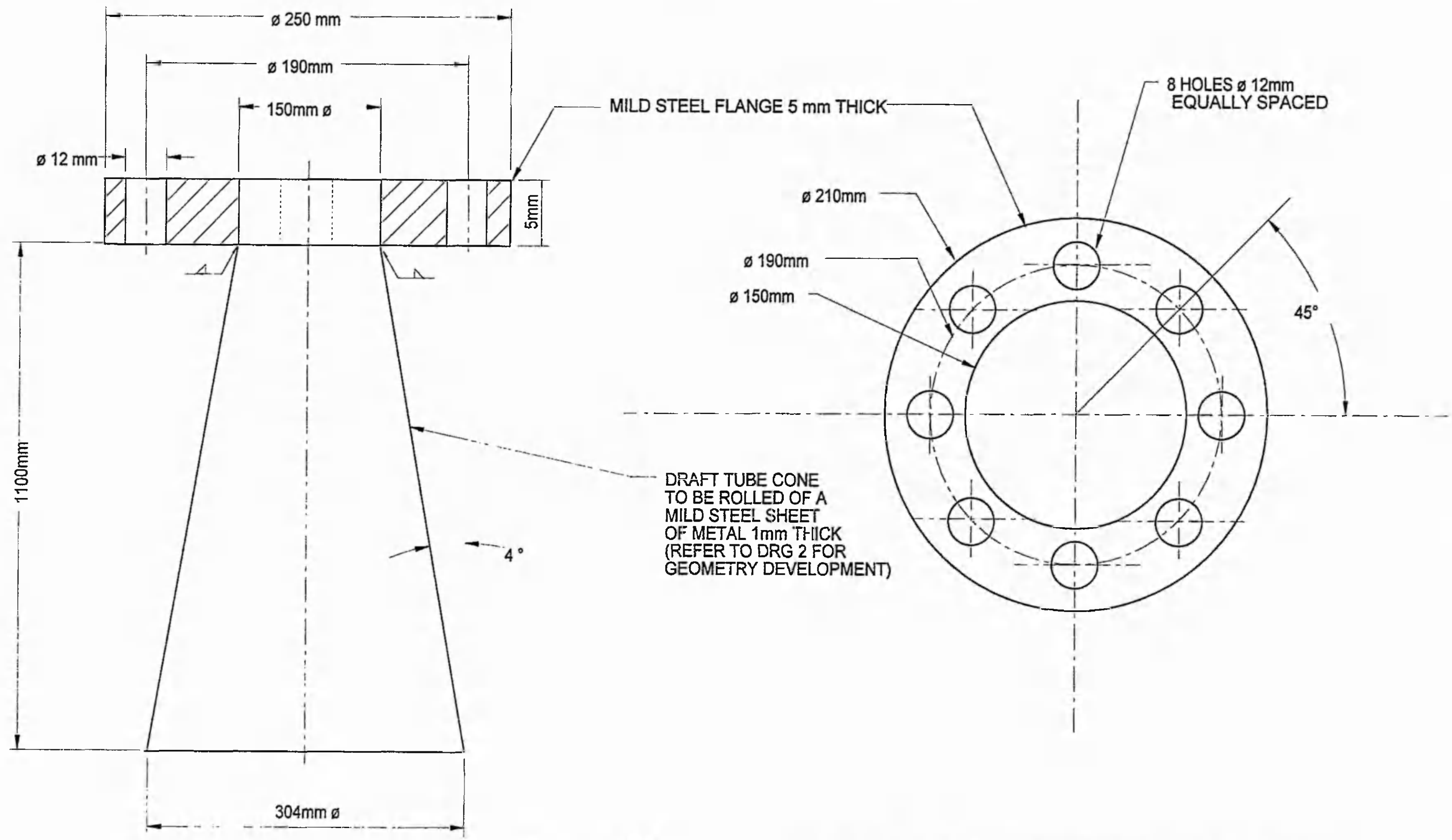
G.1 List of drawings

1. Drawing 0	Guide vane/runner window assembly.
2. Drawing 1	Draft tube/flange support.
3. Drawing 2	Draft tube geometry development.
4. Drawing 3	Runner window top flange.
5. Drawing 4	Runner window bottom flange.
6. Drawing 5	Runner window assembly.
7. Drawing 6	Guide vane details.
8. Drawing 7	Guide vane bottom support ring.
9. Drawing 8	Bottom ring bolt assembly detail.
10.Drawing 9	Guide vane top support ring.
11.Drawing 10	Guide vane ring assembly.
12.Drawing 11	Assembly details of guide vane top support ring.
13.Drawing 12	Development of a round to square adaptor for the turbine intake system.
14.Drawing 13	Round to square adaptor flange details.
15.Drawing 14	Spiral casing development.
16.Drawing 15	Spiral casing top plate.
17.Drawing 16	Spiral casing bottom plate.
18.Drawing 17	Spiral casing flange details.
19.Drawing 18	Spiral casing wall construction details.



SPECIFICATIONS	CONTRACT NO	DATE	COMPANY The Nottingham Trent University, Faculty of Engineering and Computing, Burton str. Nottingham NG1 4BU		
	DRAWN BY: G. M. DEMETRIADES				
	CHECKED BY:		TITLE GUIDE VANE / RUNNER WINDOW ASSEMBLY		
	DESIGNED BY: G. M. DEMETRIADES				
	DESIGN ACTIVITY PhD THESIS		SIZE A4	FSCM NO	DWG NO / FILE NAME 0
	CUSTOMER ELECTRICAL DEPARTMENT		SCALE NONE	DATE 13 / 8 / 95	SHEET 1 of 1

NOT TO SCALE - ALL DIMENSIONS SHOWN



		THE WELDING OF THE FLANGE TO BE WATERTIGHT			MILD STEEL	
REF	PART No	DESCRIPTION			MAT'L/SUPPL'R	No OFF
DRAWN	G M DEMETRIADES	SCALE	NONE	TITLE	PROJECTION	DRAWING No
COURSE	PhD THESIS	DATE	9/8/95	DRAFT TUBE / FLANGE SUPPORT	THIRD ANGLE	1
YEAR	GROUP	APP'D				

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Faculty of Engineering and Computing. Burton Street, Nottingham NG1 4BU.

NOT TO SCALE

THE DEVELOPMENT OF CONICAL DRAFT TUBE

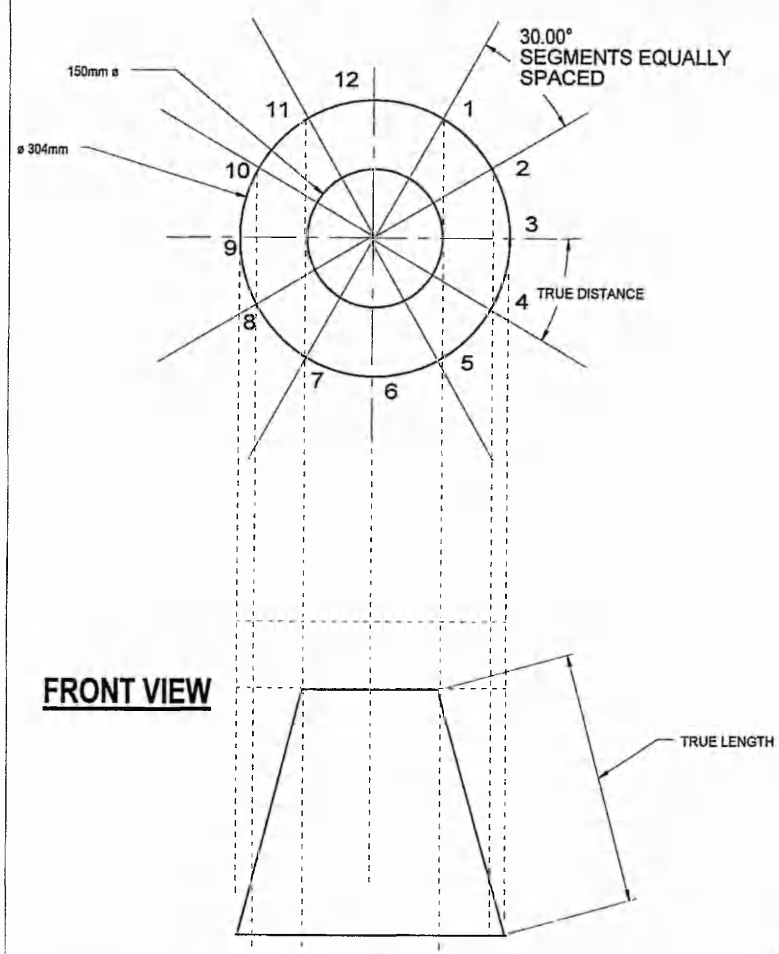
1} Divide the circle into 30 ° segmets which represent the TRUE DISTANCES. Mark each point starting from 1.

2} With a radius equal to the TRUE LENGTH swing ann arc and place point 12 anywhere on the arc.

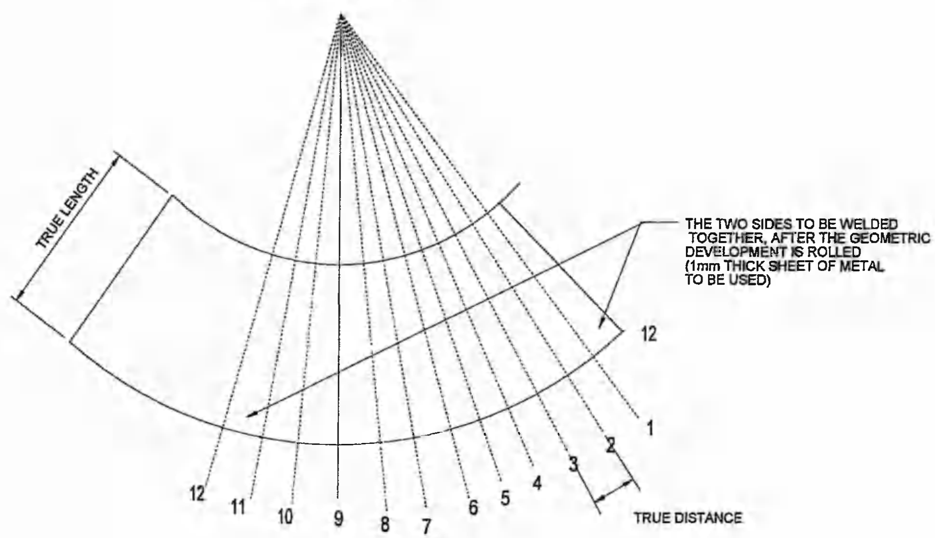
3} Measure the TRUE DISTANCE from the marked circle and transfer this onto the arc starding from point 12 for all points.

TOP VIEW

FRONT VIEW



DEVELOPMENT



REF	PART No		DESCRIPTION		MAT'L/SUPPL'R	No OFF
DRAWN	G. M. DEMETRIADES	SCALE	NONE	TITLE: DRAFT TUBE GEOMETRY DEVELOPMENT	PROJECTION	DRAWING No: 2
COURSE		DATE	10/8/95			
YEAR	GROUP	APP'D				

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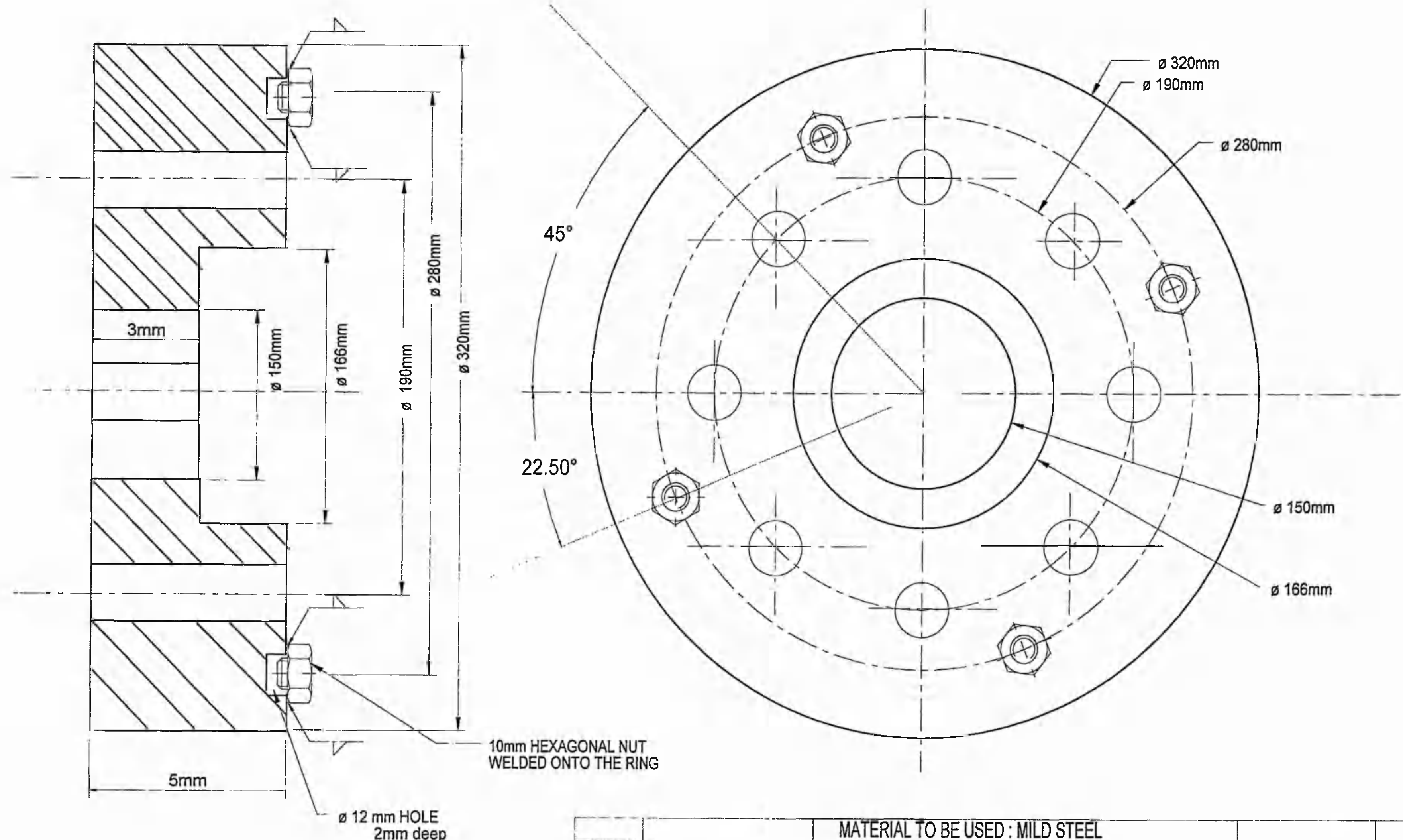

MATERIAL TO BE USED : MILD STEEL

RFF	PART No	DESCRIPTION			MAT'L/SUPPL'R	No OFF
DRAWN	G.M.DEMETRIADES	SCALE	NONE	TITLE	PROJECTION	DRAWING No
COURSE	PhD. THESIS	DATE	9 / 8 / 95	RUNNER WINDOW TOP FLANGE	THIRD ANGLE	3
YEAR	GROUP	APP'D				

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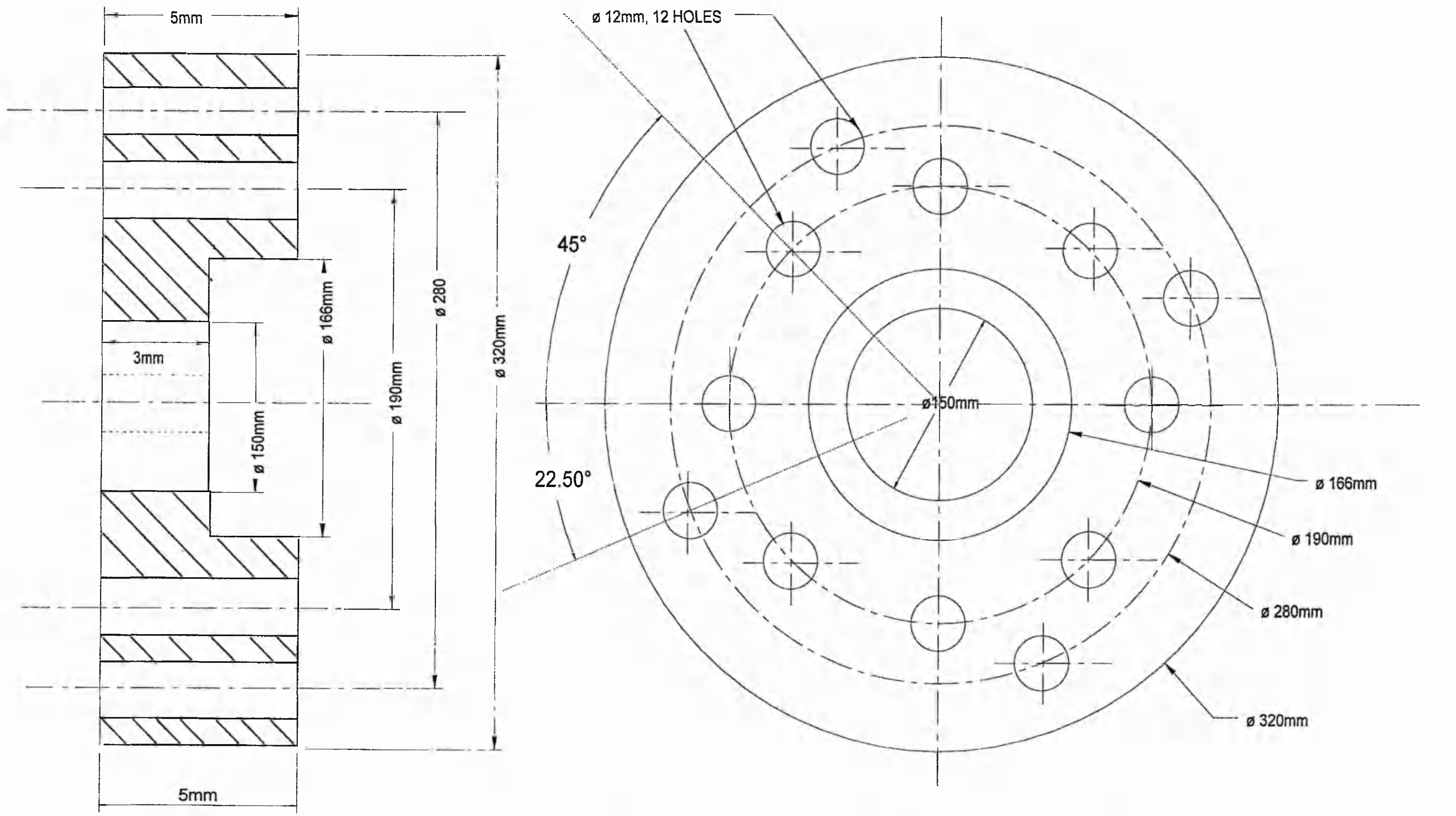

MATERIAL TO BE USED : MILD STEEL

REF	PART No	DESCRIPTION			MAT'L/SUPPL'R	No OFF
DRAWN	G.M.DEMETRIADES	SCALE	NONE	TITLE	PROJECTION	DRAWING No
COURSE	PhD. THESIS	DATE	9 / 8 / 95	RUNNER WINDOW BOTTOM FLANGE	THIRD ANGLE	4
YEAR	GROUP	APP'D				

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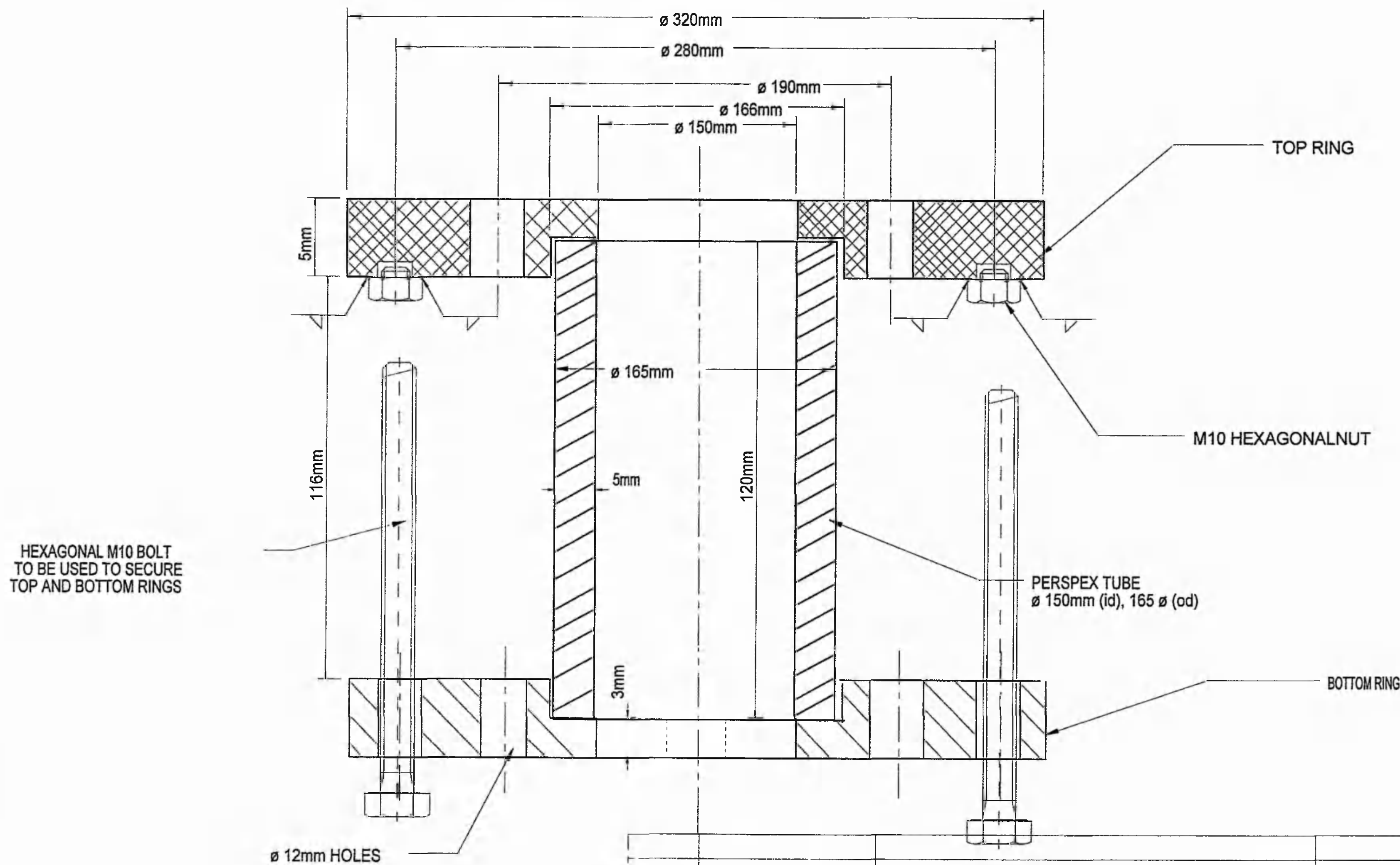

REF	PART No		DESCRIPTION SECTIONAL VIEW OF THE ASSEMBLY			MAT'L/SUPPL'R	No OFF
DRAWN	G.M. DEMETRIADES		SCALE NONE	TITLE		PROJECTION	DRAWING No
COURSE	PhD. THESIS		DATE 9/8/95	RUNNER WINDOW ASSEMBLY			5
YEAR		GROUP	APP'D				

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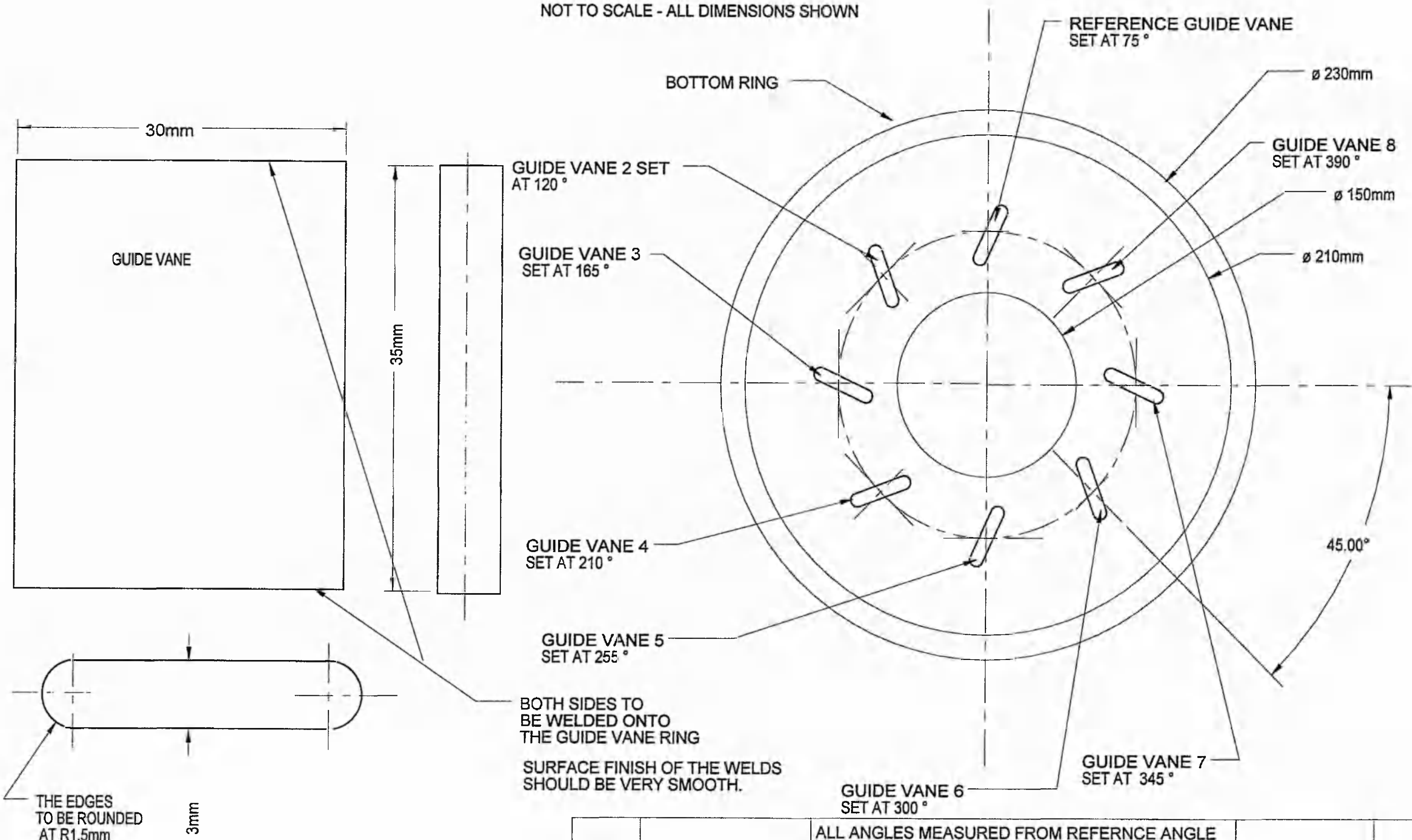

		ALL ANGLES MEASURED FROM REFERNCE ANGLE				
		GUIDE VANE ANGLES MEASURED FROM HORIZONTAL				
REF	PART No	DESCRIPTION			MAT'L/SUPPL'R	No OFF
DRAWN	G. M. DEMETRIADES	SCALE	NONE	TITLE	PROJECTION	DRAWING No
COURSE	PhD THESIS	DATE	11 / 8 / 95	GUIDE VANE DETAILS		6
YEAR	GROUP	APP'D				

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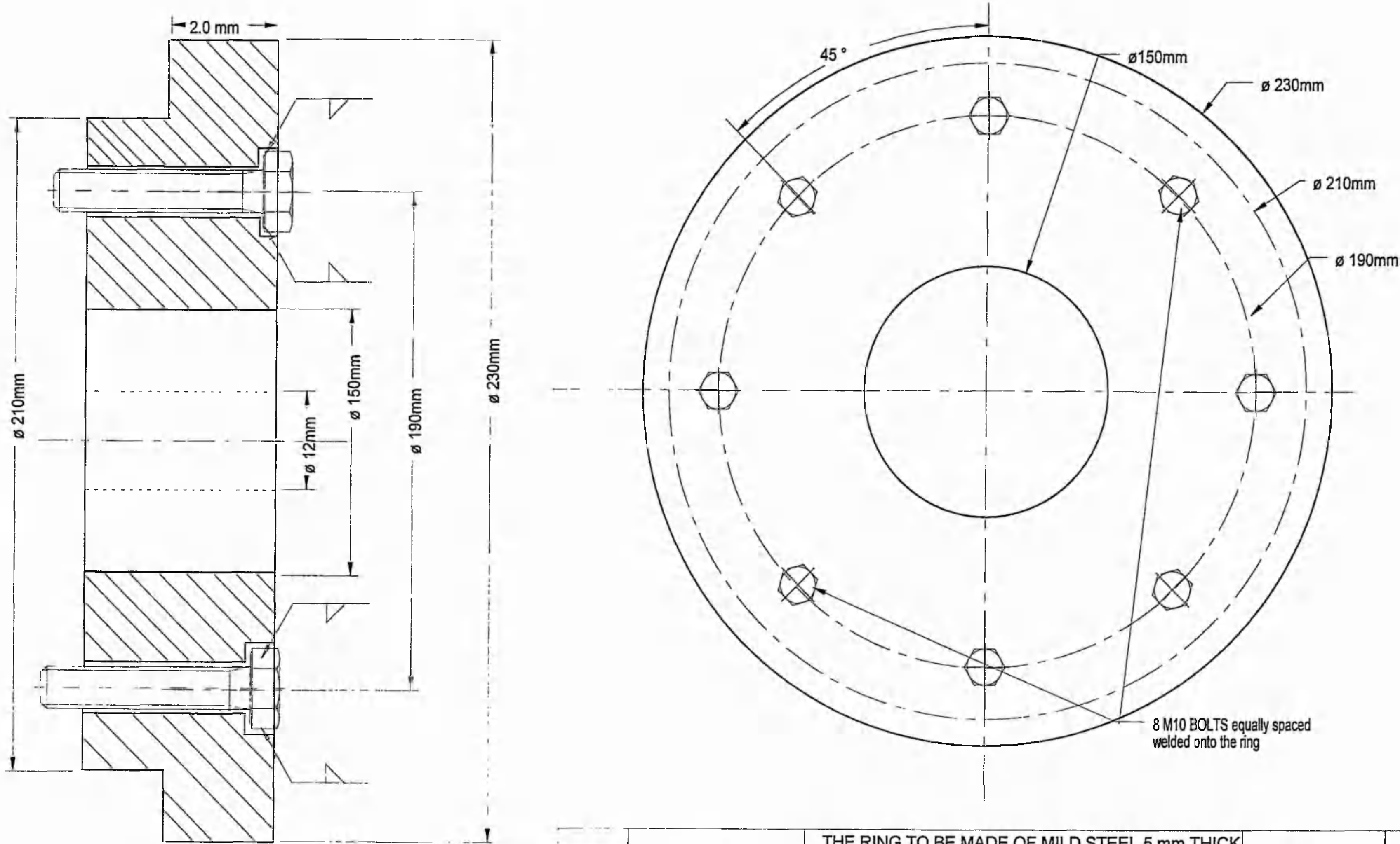
NOT TO SCALE - ALL DIMENSIONS SHOWN



		THE RING TO BE MADE OF MILD STEEL 5 mm THICK					
REF	PART No	DESCRIPTION				MAT'L/SUPPL'R	No OFF
DRAWN	G. M. DEMETRIADES	SCALE	NONE	TITLE		PROJECTION	DRAWING No
COURSE	PhD THESIS	DATE	11 / 8 / 95	GUIDE VANE BOTTOM SUPPORT RING			7
YEAR	GROUP	APP'D					

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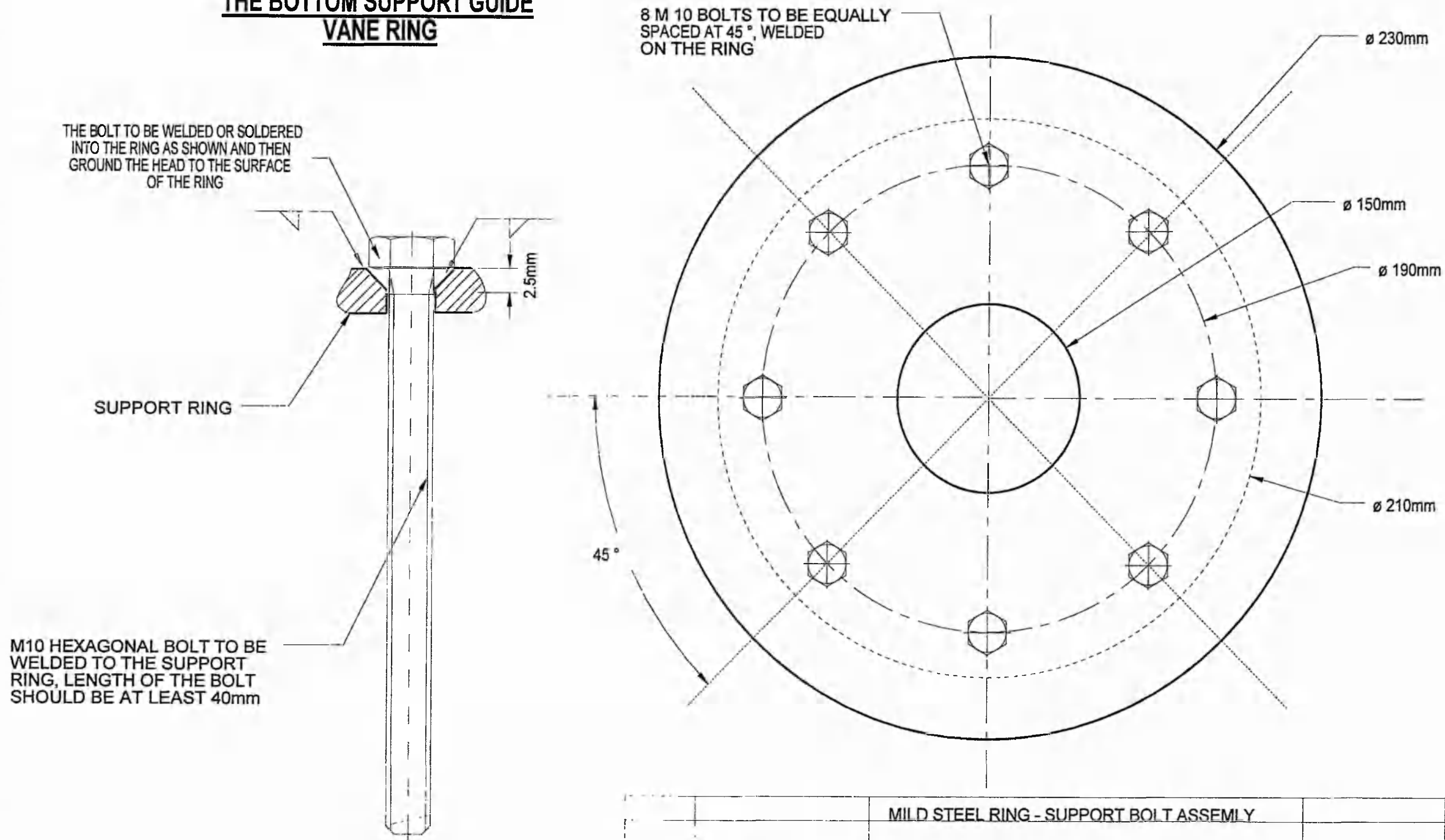
MANUFACTURING DETAIL FOR THE BOTTOM SUPPORT GUIDE VANE RING



		MILD STEEL RING - SUPPORT BOLT ASSEMLY			
REF	PART No	DESCRIPTION		MAT'L/SUPPL'R	No OFF
DRAWN	G. M. DEMETRIADES	SCALE	NONE	TITLE BOTTOM RING BOLT ASSEMBLY DETAIL	PROJECTION / DRAWING No 8
COURSE	PhD THESIS	DATE	11 / 8 / 95		
YEAR	GROUP	APP'D			

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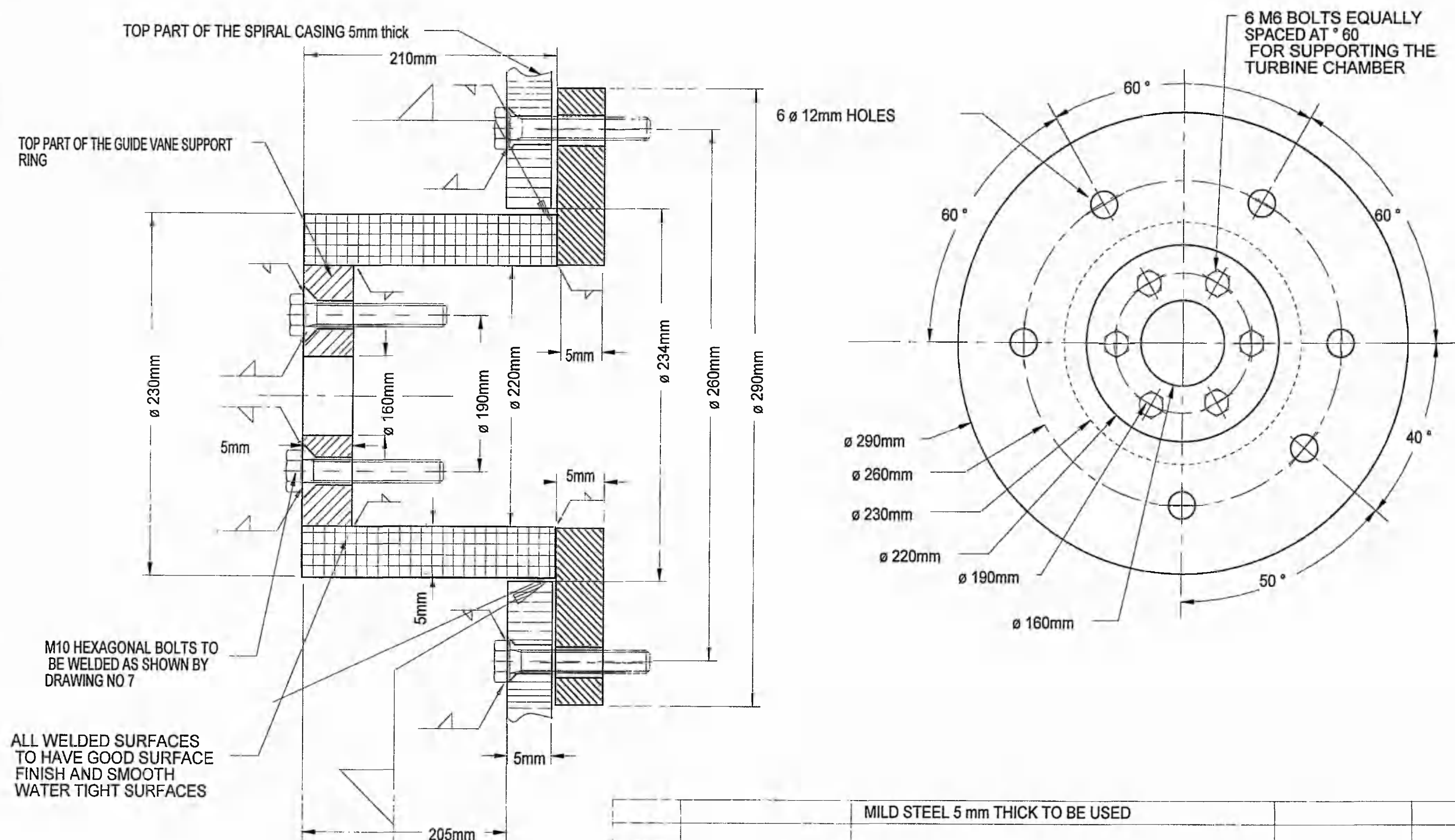
NOT TO SCALE - ALL DIMENSIONS SHOWN



		MILD STEEL 5 mm THICK TO BE USED				
REF	PART No	DESCRIPTION			MAT'L/SUPPL'R	No OFF
DRAWN	G. M. DEMETRIADES	SCALE		TITLE	PROJECTION	DRAWING No
COURSE	PhD THESIS	DATE	11 / 8 / 95	GUIDE VANE RING TOP SUPPORT RING		9
YEAR	GROUP	APP'D				

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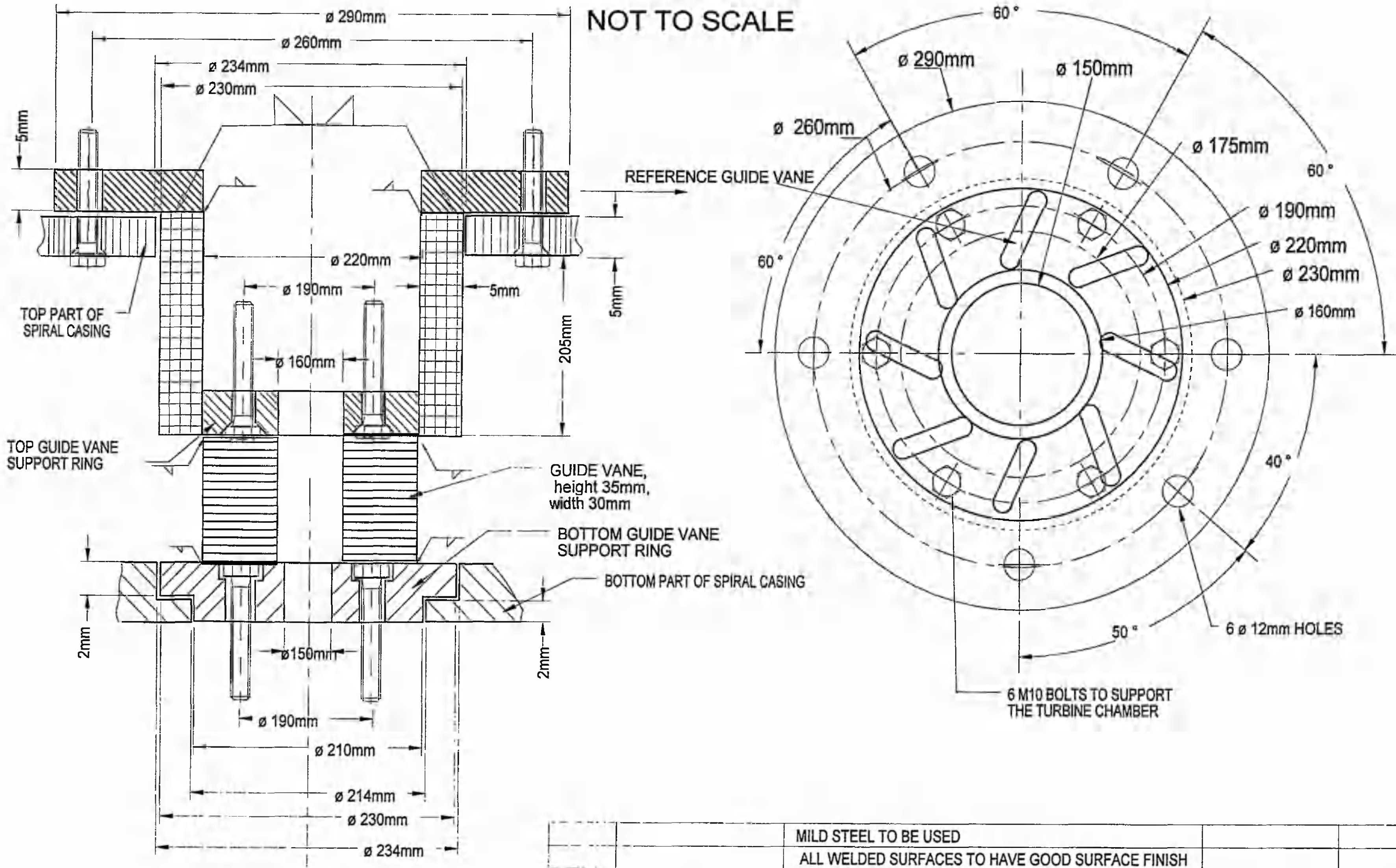

		MILD STEEL TO BE USED			
		ALL WELDED SURFACES TO HAVE GOOD SURFACE FINISH			
REF	PART No	DESCRIPTION		MAT'L/SUPPL'R	No OFF
DRAWN	G. M. DEMETRIADES	SCALE	NONE	PROJECTION	DRAWING No
COURSE	PhD THESIS	DATE	12 / 8 / 95	TITLE: GUIDE VANE RING ASSEMBLY	10
YEAR	GROUP	APP'D			

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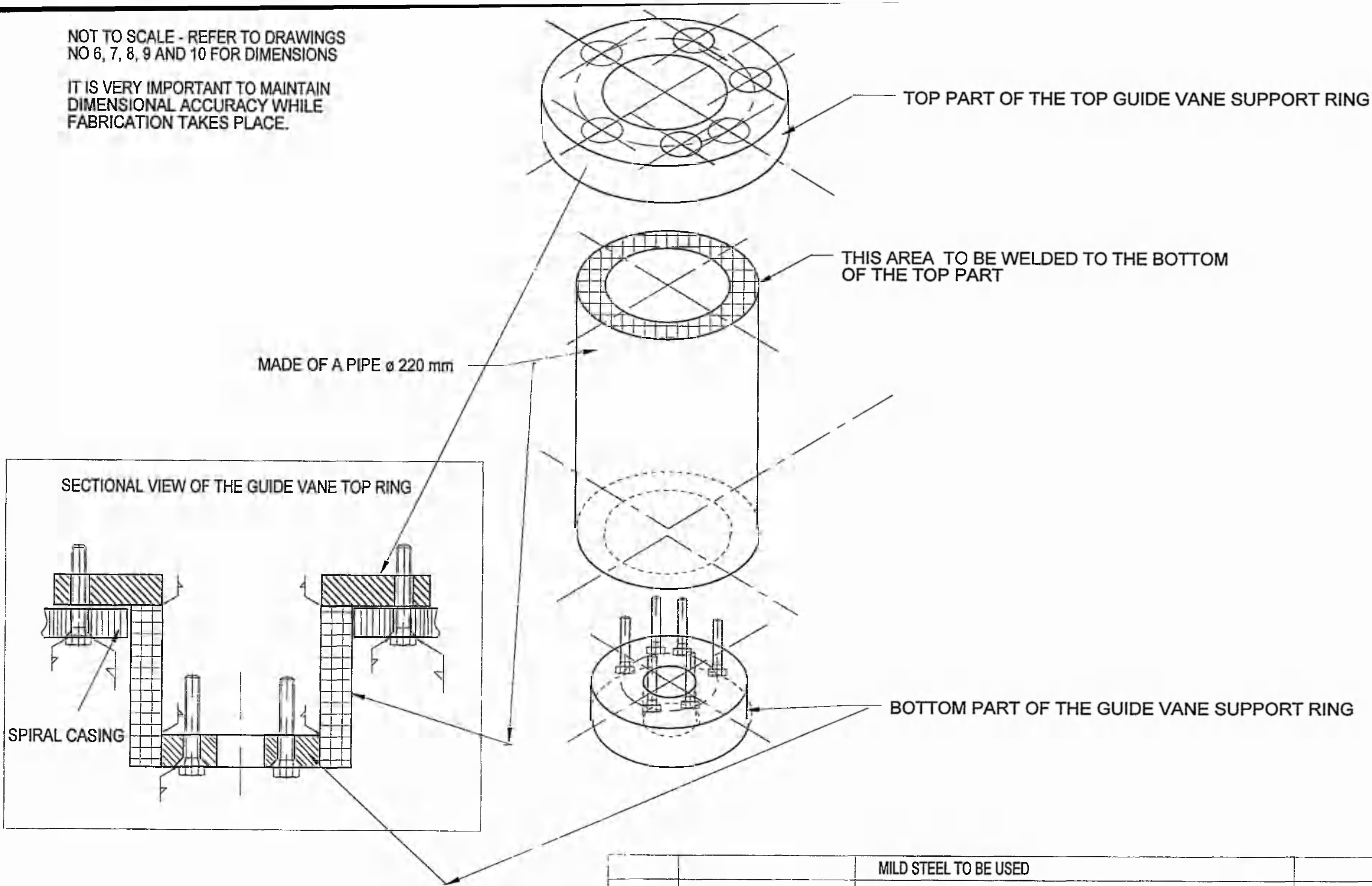
NOT TO SCALE - REFER TO DRAWINGS NO 6, 7, 8, 9 AND 10 FOR DIMENSIONS

IT IS VERY IMPORTANT TO MAINTAIN DIMENSIONAL ACCURACY WHILE FABRICATION TAKES PLACE.



		MILD STEEL TO BE USED				
REF	PART No	DESCRIPTION			MAT'L/SUPPL'R	No OFF
DRAWN	G. M. DEMETRIADES	SCALE	NONE	TITLE	PROJECTION	DRAWING No
COURSE	PhD THESIS	DATE	15 / 8 / 95	ASSEMBLY DETAILS OF THE GUIDE VANE TOP SUPPORT RING		11
YEAR	GROUP	APP'D				

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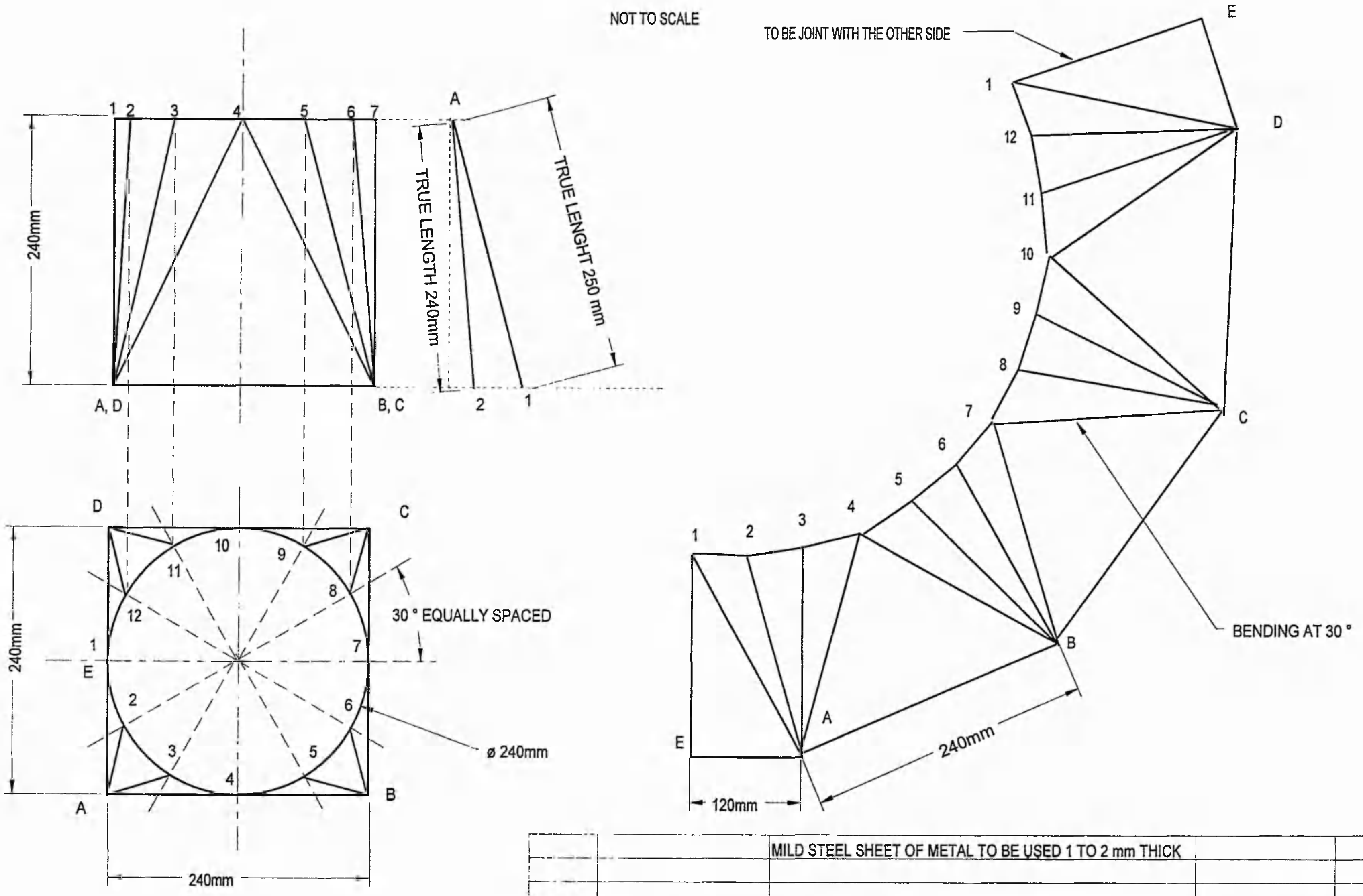


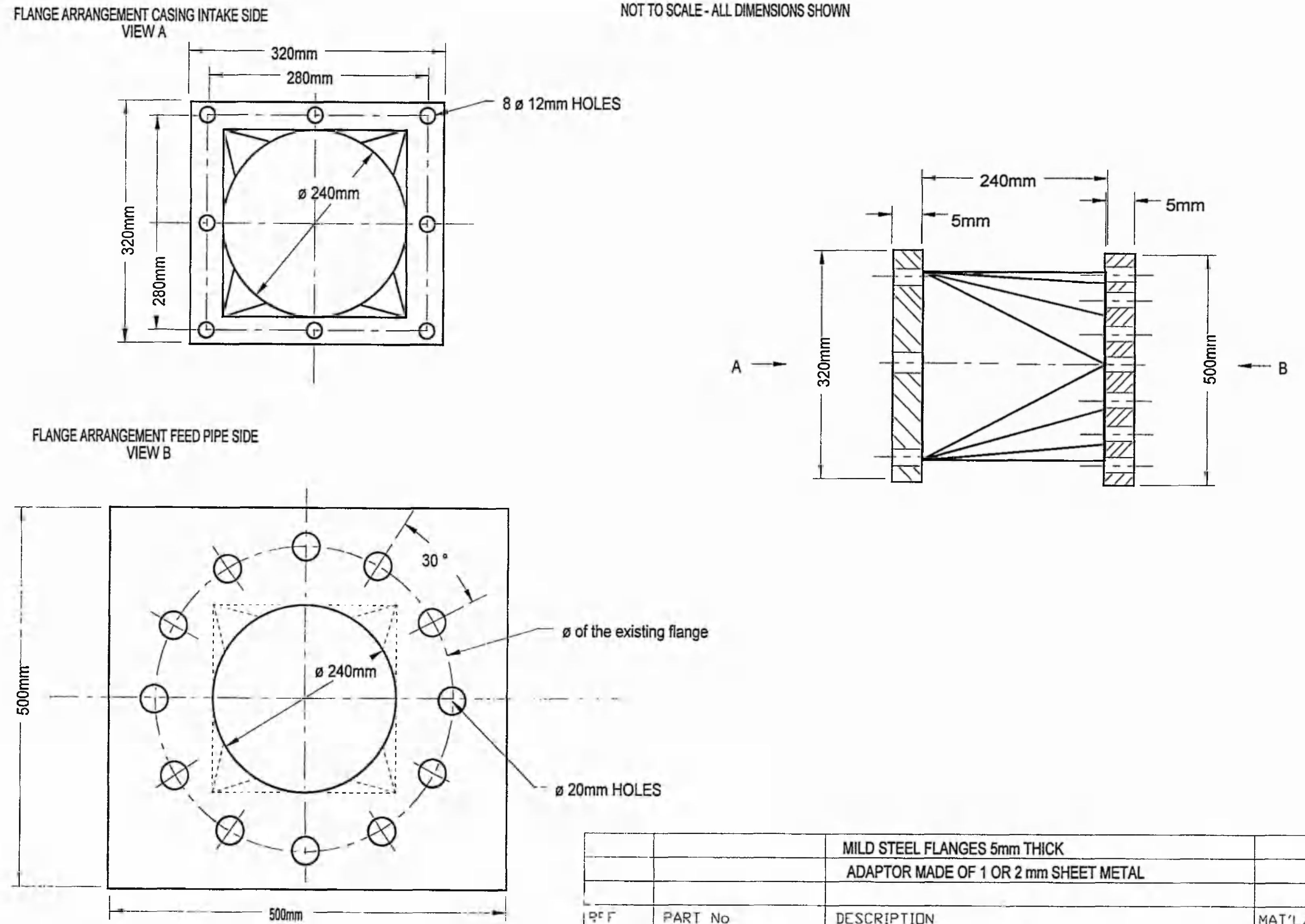
MILD STEEL SHEET OF METAL TO BE USED 1 TO 2 mm THICK

REF	PART No	DESCRIPTION		MAT'L/SUPPL'R	No OFF
DRAWN	G. M. DEMETRIADES	SCALE	NONE	PROJECTION	DRAWING No
COURSE	PhD THESIS	DATE	13 / 8 / 95		12
YEAR	GROUP	APP'D			

TITLE: DEVELOPMENT OF A ROUND TO SQUARE ADAPTOR FOR THE TURBINE INTAKE SYSTEM

The NOTTINGHAM TRENT UNIVERSITY.
Faculty of Engineering and Computing. Burton Street, Nottingham NG1 4BU.

NOT TO SCALE - ALL DIMENSIONS SHOWN



		MILD STEEL FLANGES 5mm THICK				
		ADAPTOR MADE OF 1 OR 2 mm SHEET METAL				
REF	PART No	DESCRIPTION			MAT'L/SUPPL'R	No OFF
DRAWN	G. M. DEMETRIADES	SCALE	NONE	TITLE	PROJECTION	DRAWING No
COURSE	PhD THESIS	DATE	13 / 8 /95	ROUND TO SQUARE ADAPTOR FLANGE DETAILS		13
YEAR	GROUP	APP'D				

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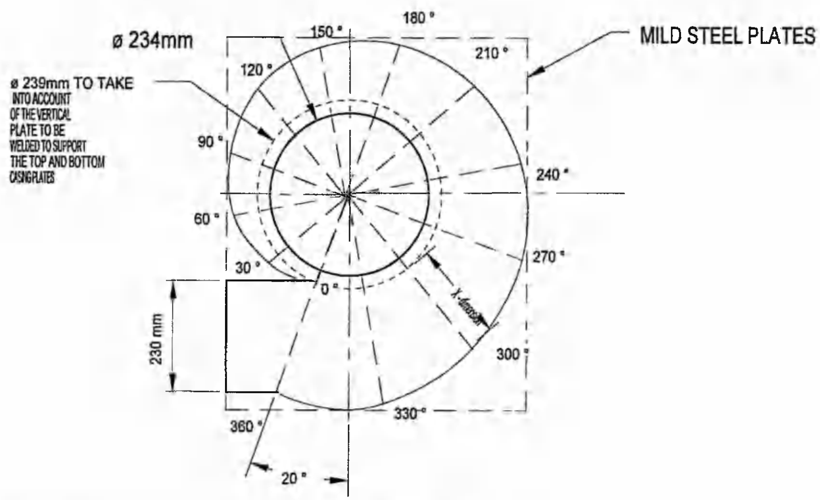
MARKING THE DIMENSIONS OF THE SPIRAL CASING - METHOD A



ANGLE	X - DIMENSION (m)
0°	0.005
30°	0.025
60°	0.045
90°	0.065
120°	0.085
150°	0.105
180°	0.125
210°	0.145
240°	0.165
270°	0.185
300°	0.205
330°	0.225
360°	0.245

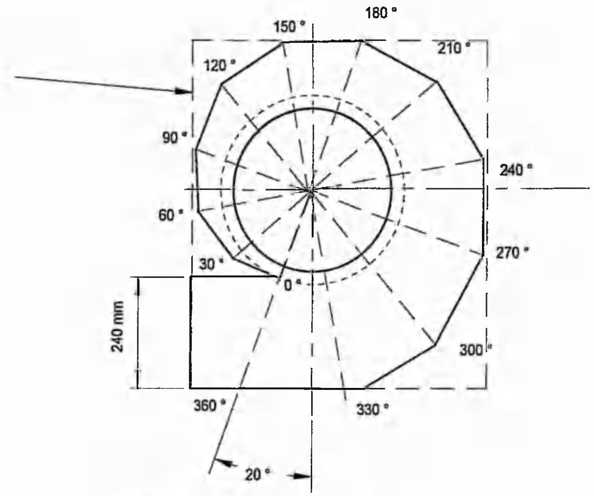
PROCEDURE TO FOLLOWED

1. MARK THE ø234mm AND THE ø239mm. THIS TAKES INTO ACCOUNT THE WALL THICKNESS OF THE CASING.

2. DIVIDE THE CIRCLES AS SHOWN AND MARK ON EACH SEGMENT THE " X - DIMENSION " FROM THE TABLE.

3. JOIN ALL THE POINTS TO FORM THE SPIRAL.

MARKING THE DIMENSIONS OF THE SPIRAL CASING - METHOD B



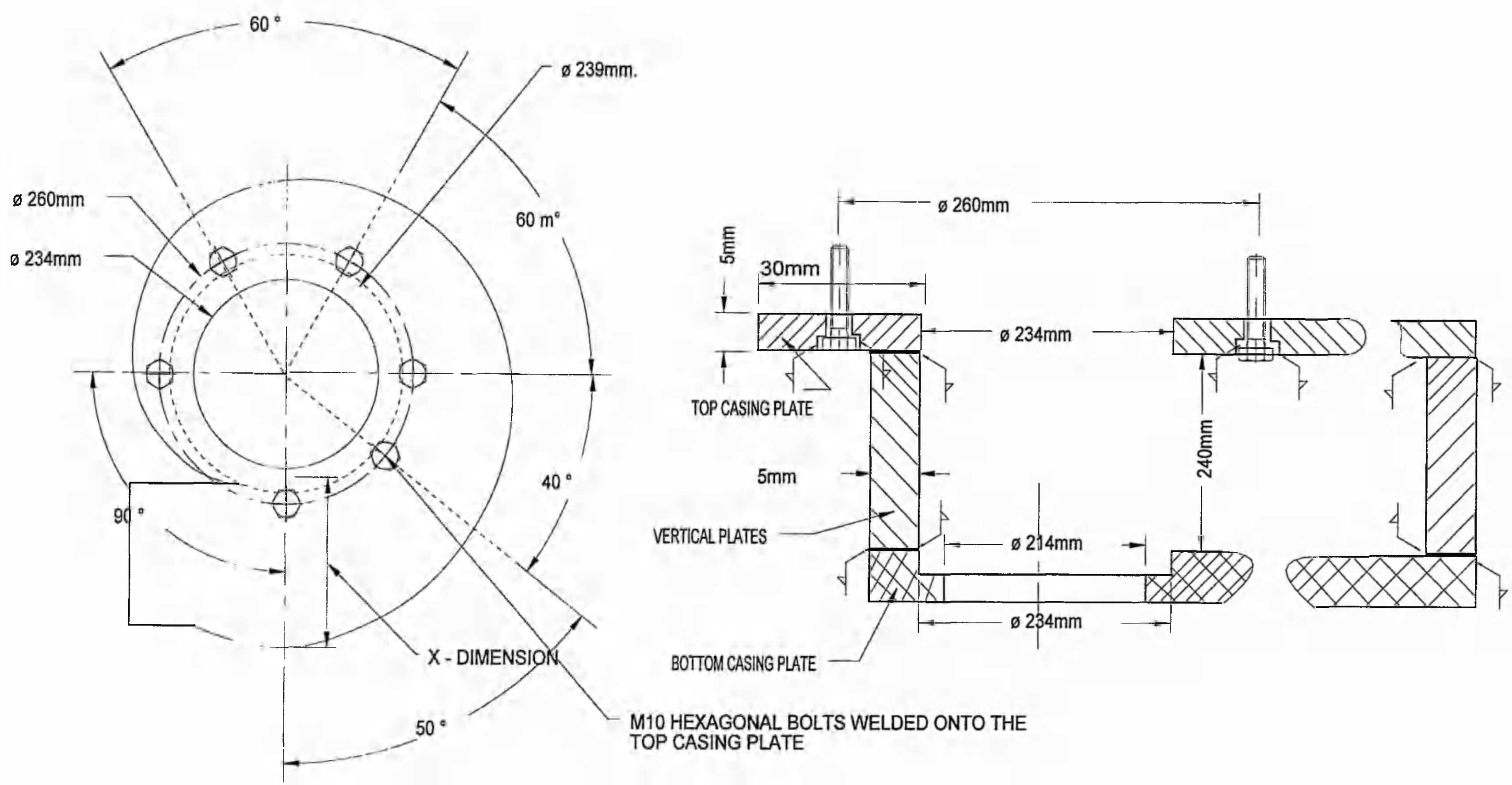
PROCEDURE TO BE FOLLOWED

1. FOLLOW STEP 1 OF METHOD A AND MARK THE RELEVANT DIMENSIONS.

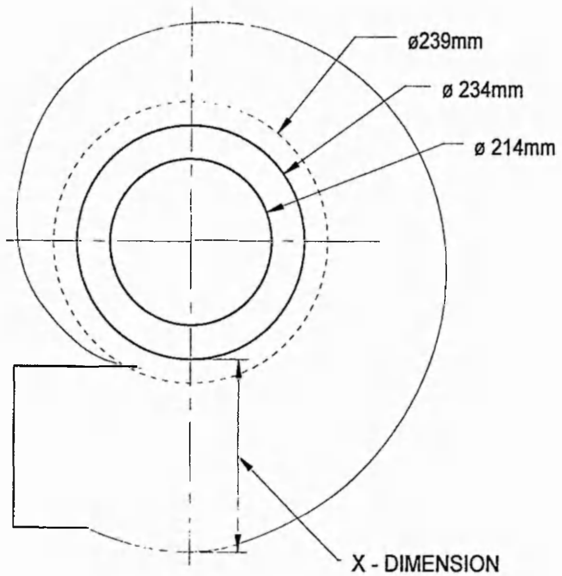
2. JOIN EACH X - DIMENSION POINT BY A STRAIGHT LINE.

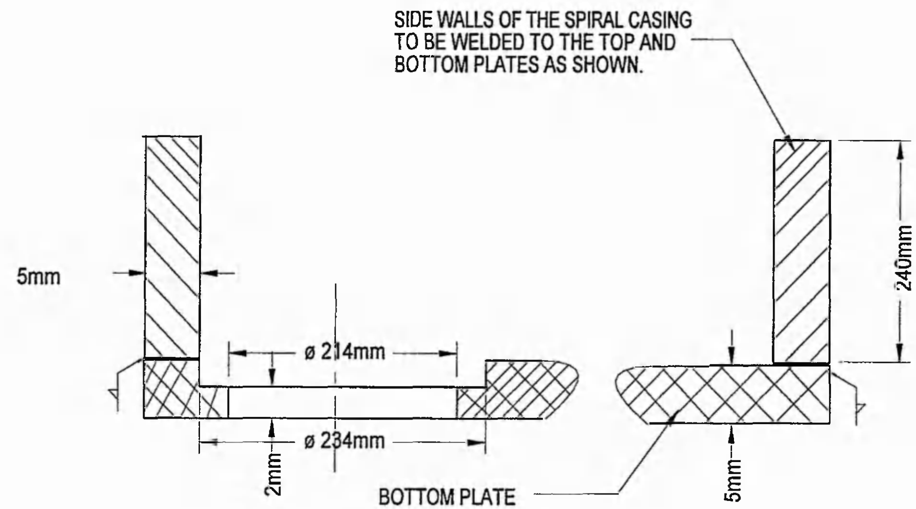
3. THE SHAPE FORMED "LOOKS" LIKE A SPIRAL AND IT IS EASIER TO BE MADE IF NO MACHINERY IS AVAILABLE TO PERFORM METHOD A.

SPECIFICATIONS CASING MATERIAL : MILD STEEL 5 mm THICK	CONTRACT NO.	DATE	COMPANY The Nottingham Trent University, Faculty of Engineering and Computing, Burton str. Nottingham, NG1 4BU
REFER TO DRAWING 18 FOR SPIRAL DVELOPMENT TO SCALE	DRAWN BY G. M. DEMETRIADES	13 / 8 / 95	
	CHECKED BY		TITLE SPIRAL CASING DEVELOPMENT
	DESIGNED BY G. M. DEMETRIADES		
	DESIGN ACTIVITY		SIZE A4 · FSCM NO. · DWG NO. / FILE NAME 14
	CUSTOMER ELECTICAL ENG. DEPARTMENT		SCALE NONE · DATE 13 / 8 / 95 · SHEET 1 of 1

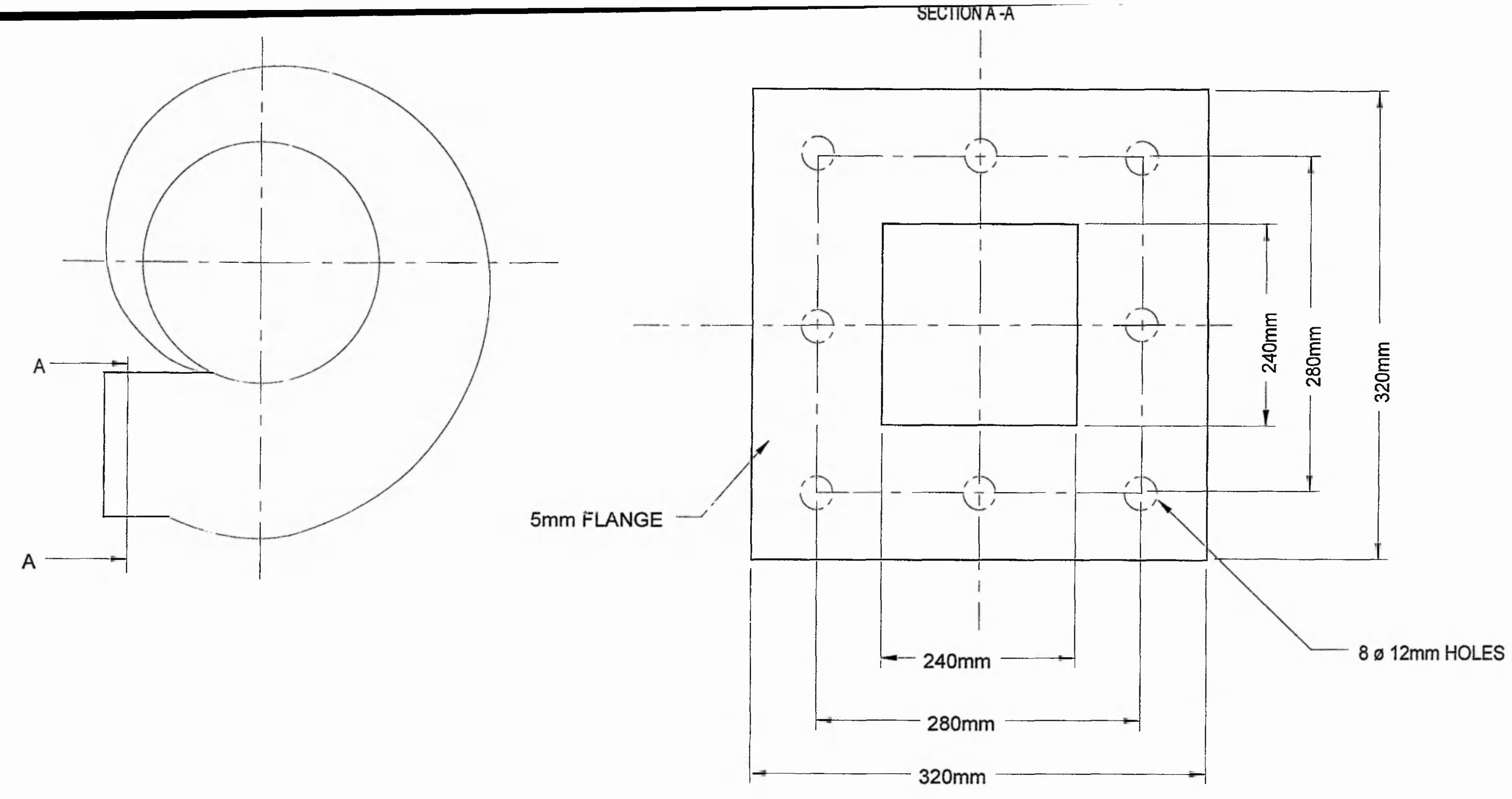


SPECIFICATIONS	CONTRACT NO.	DATE	COMPANY The Nottingham Trent University Faculty of Engineering and Computing Burton str. Nottingham, NG1 4BU		
MILD STEEL 5 mm THICK TOP CASING PLATE	DRAWN BY G. M. DEMETRIADES	13 / 8 / 95			
DIMENSIONAL ACCURACY VERY IMPORTANT	CHECKED BY		TITLE SPIRAL CASING TOP PLATE		
	DESIGNED BY G. M. DEMETRIADES				
	DESIGN ACTIVITY		SIZE A4	FSCM NO.	DWG NO. / FILE NAME 15
	CUSTOMER ELECTICAL ENG. DEPARTMENT		SCALE NONE	DATE 13 / 8 / 95	SHEET 1 of 1





SPECIFICATIONS	CONTRACT NO.	DATE	COMPANY The Nottingham Trent University Faculty of Engineering and Computing Burton str. Nottingham, NG1 4BU
DIMENSIONAL ACCURACY VERY IMPORTANT	DRAWN BY G. M. DEMETRIADES	13 / 8 / 95	
	CHECKED BY		TITLE SPIRAL CASING BOTTOM PLATE
	DESIGNED BY G. M. DEMETRIADES		
	DESIGN ACTIVITY		SIZE A4 / FSCM NO. / DWG NO. / FILE NAME 16
	CUSTOMER ELECTICAL ENG. DEPARTMENT		SCALE NONE / DATE 13 / 8 / 95 / SHEET 1 of 1



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REFER TO DRAWING 18 FOR SPIRAL DEVELOPMENT TO SCALE	CHECKED BY		TITLE SPIRAL CASING FLANGE DETAILS		
	DESIGNED BY G. M. DEMETRIADES		SIZE A4	FSCM NO.	DWG NO. / FILE NAME 17
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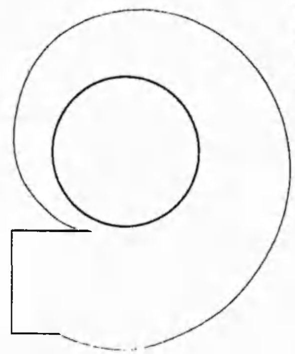
SPIRAL CASING DEVELOPED BY METHOD A



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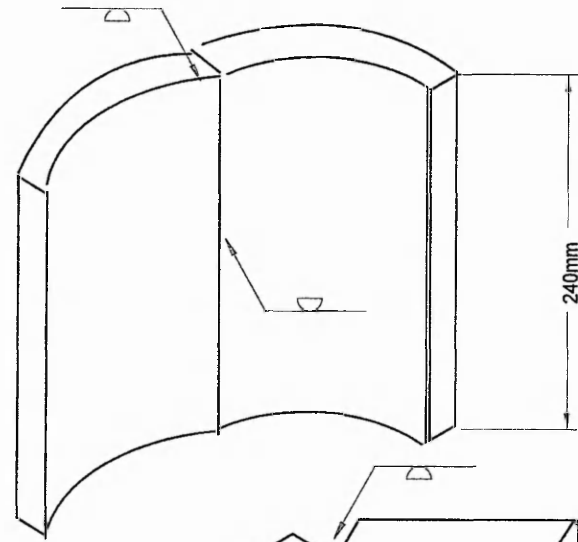

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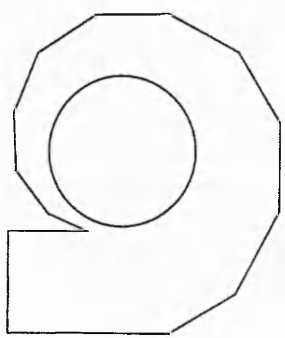
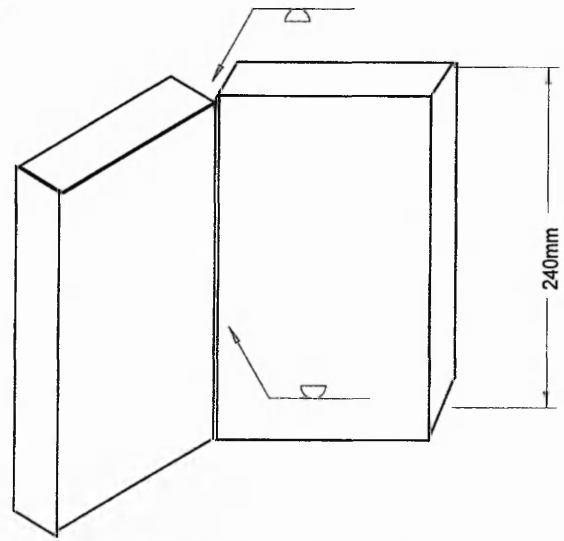


THE WALL TO BE MADE OF MILD STEEL STEEL PLATES 5mm THICK WELDED TOGETHER ALONG THE SHAPE LINE OF THE SPIRAL



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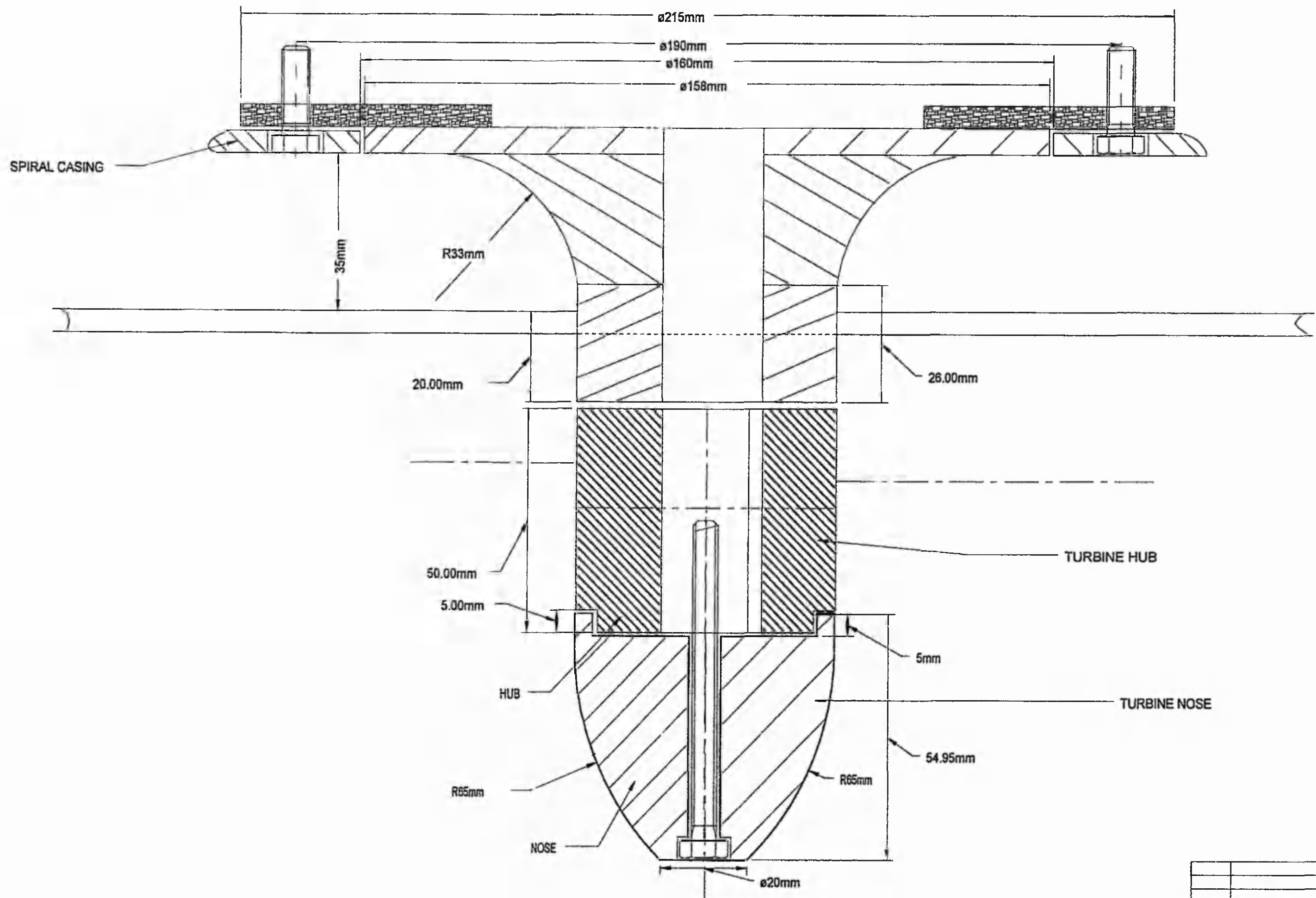
COURSE	PhD THESIS	DATE	15/8/95	SPIRAL CASING WALL CONSTRUCTION DETAILS		18
YEAR	GROUP	APP'D				

The NOTTINGHAM TRENT UNIVERSITY.
Faculty of Engineering and Computing. Burton Street, Nottingham NG1 4BU.



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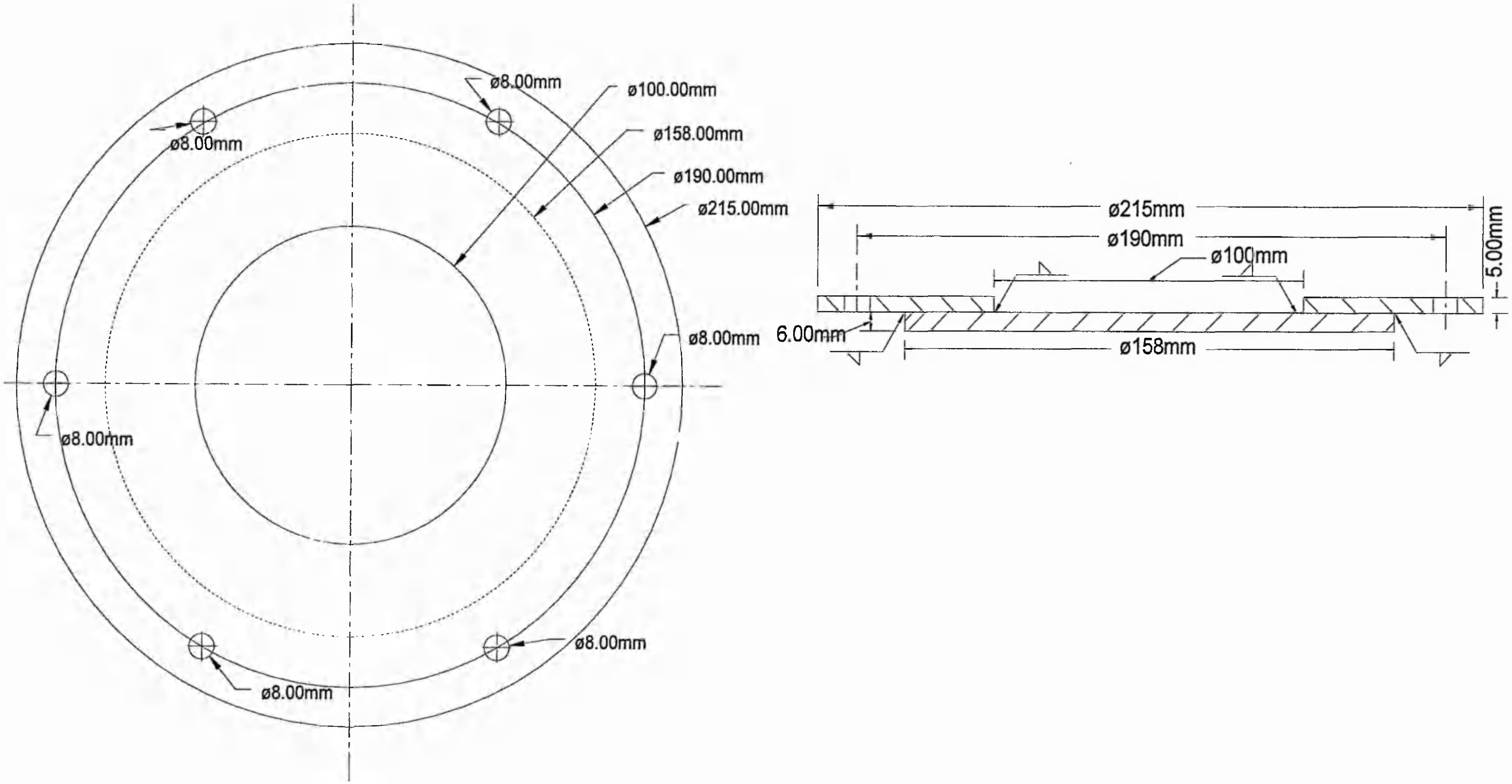

SPECIFICATIONS	CONTRACT NO.	DATE	COMPANY The Nottingham Trent University Burton Str. Nottignham NG1 4BU		
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DIMENSIONAL ACCURACY VERY IMPORTANT	CHECKED BY		TITLE TURBINE CHAMBER TOP RING		
MILD STEEL PLATES 5 mm THICK TO BE USED	DESIGNED BY G.M. DEMETRIADES		SIZE A4	FSCM NO.	DWG NO / FILE NAME 20
PLATES TO BE WELDED TOGETHER	DESIGN ACTIVITY PhD THESIS				
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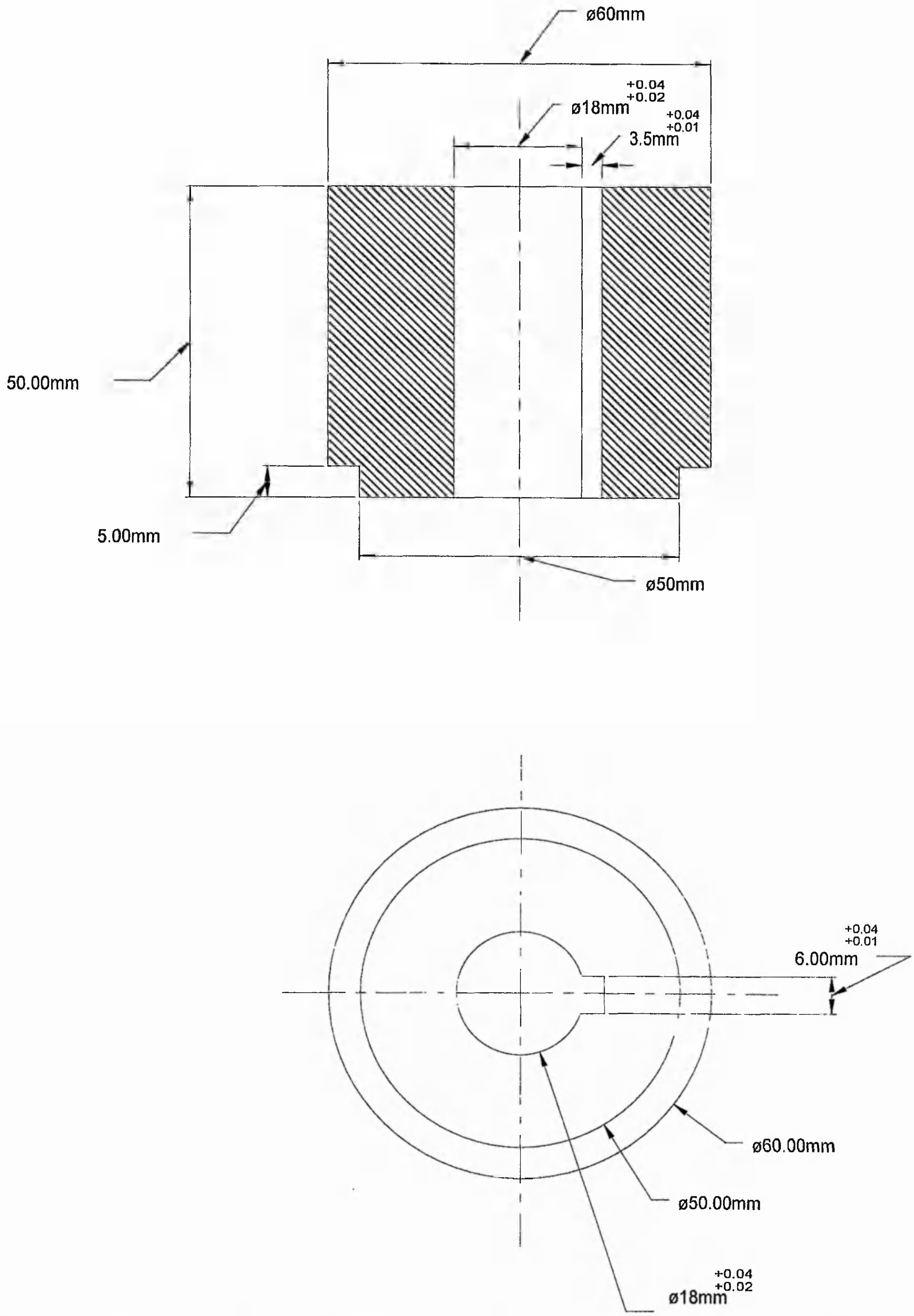

SPECIFICATIONS	CONTRACT NO	DATE	COMPANY		
MILD STEEL TO BE USED	DRAWN BY: G. M. DEMETRIADES	19 / 8 / 95	The Nottingham Trent University Burton Str. Nottingham NG1 4BU		
	CHECKED BY:		TITLE RUNNER HUB		
	DESIGNED BY: G. M. DEMETRIADES				
	DESIGN ACTIVITY PhD THESIS		SIZE A4	FSCM NO	DWG NO / FILE NAME 21
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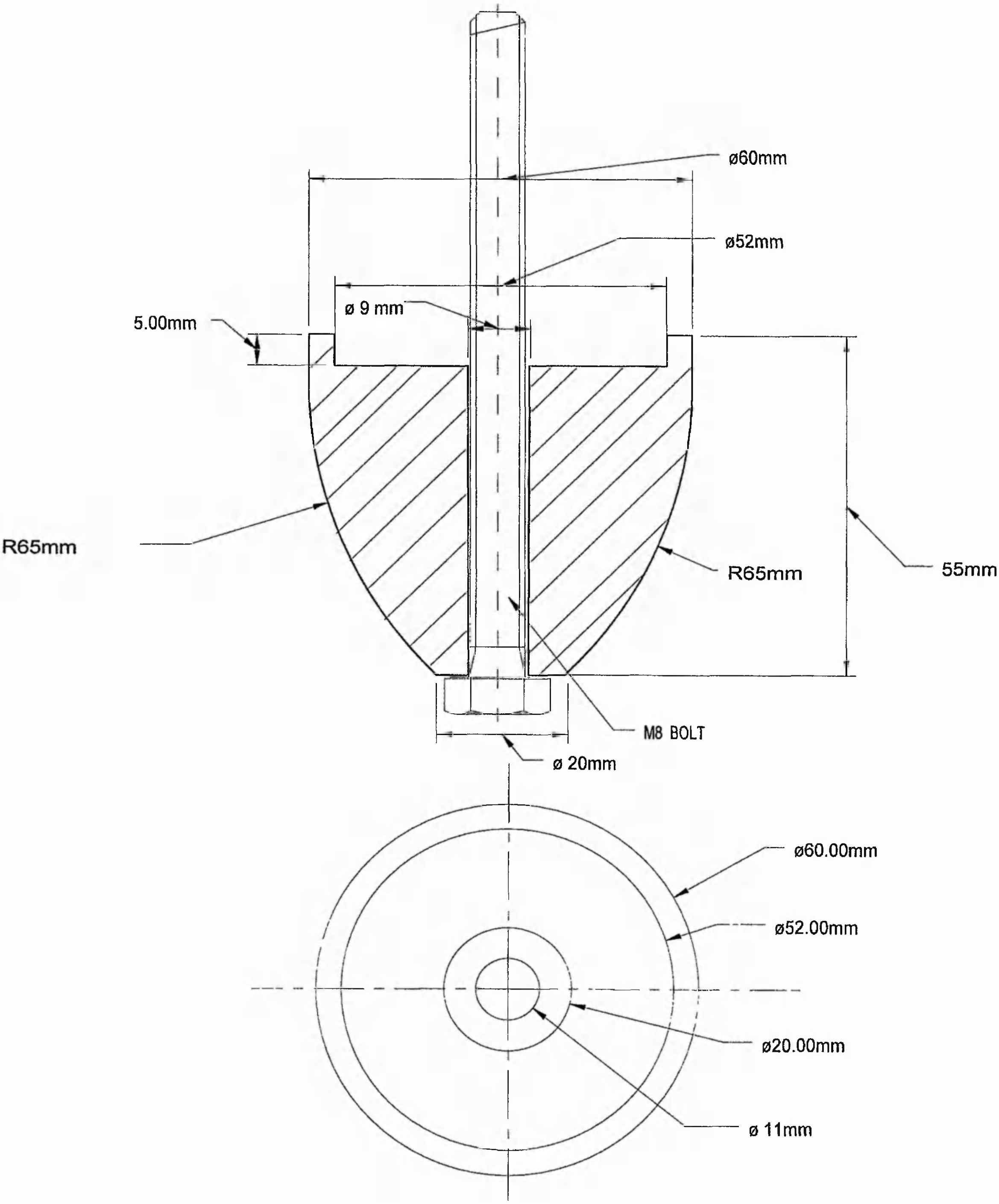

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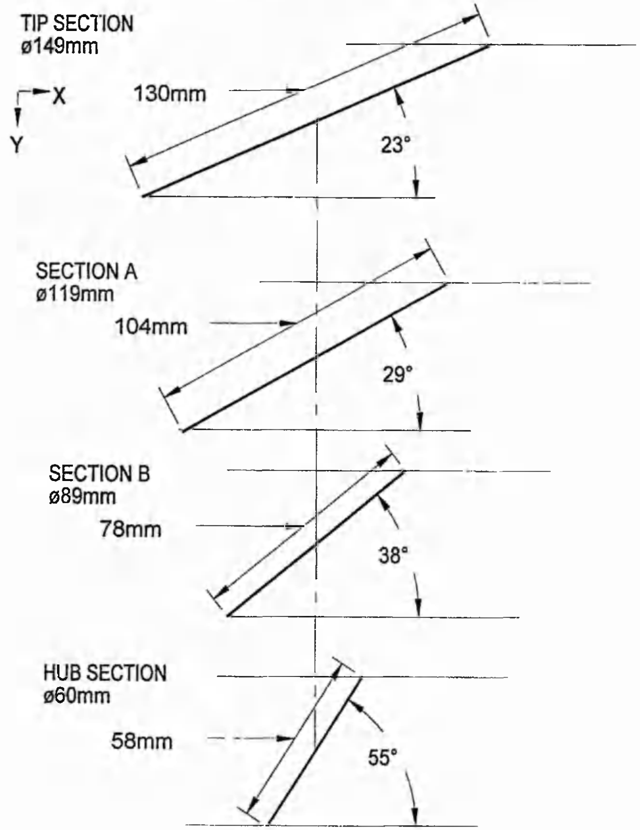
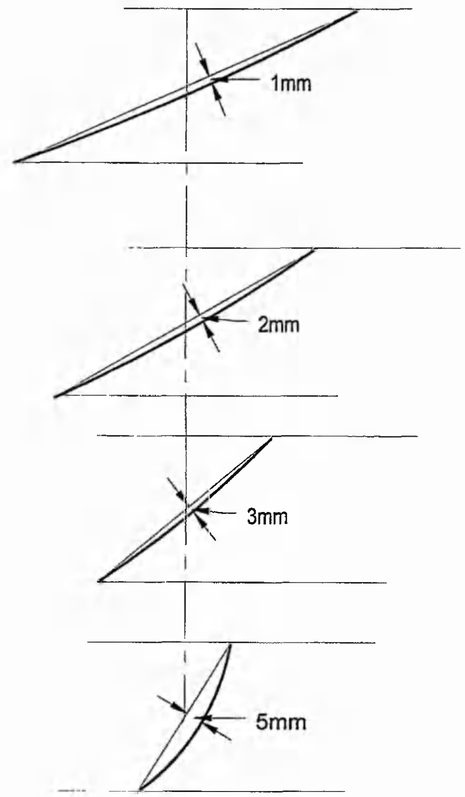
	DESIGNED BY: G. M. DEMETRIADES	19/8/95			
	DESIGN ACTIVITY PhD THESIS		SIZE A4	FSCM NO.	DWG NO / FILE NAME 22
	CUSTOMER ELECTRICAL DEPARTMENT		SCALE	DATE 19 / 8 / 95	SHEET 1 of 1

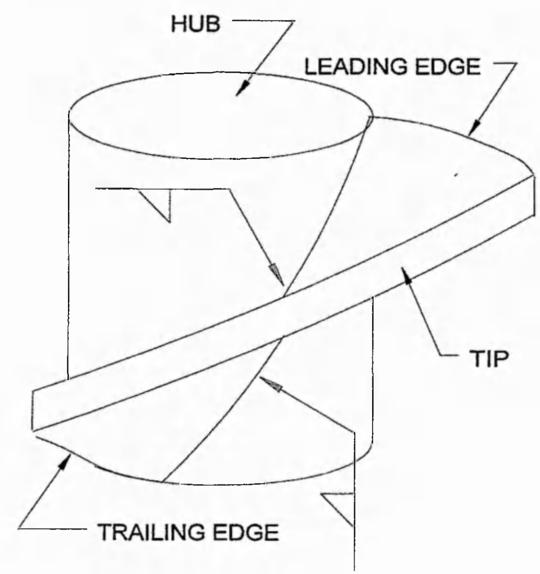
BLADE SETTING ANGLES



BLADE CAMBER POSITIONING



HUB - BLADE ASSEMBLY



SPECIFICATIONS	CONTRACT NO	DATE	COMPANY The Nottingham Trent University Burton Str. Nottingham NG1 4BU		
THE BLADES TO BE MADE OF 5mm THICK MILD STEEL PLATES	DRAWN BY G.M. DEMETRIADES	19/8/95			
CAMBER AND TWIST TO BE AS ACCURATE AS POSSIBLE	CHECKED BY		TITLE TURBINE BLADE SETTING DETAILS		
THE HUB SECTION OF THE BLADE TO BE WELDED ONTO THE HUB AS SHOWN	DESIGNED BY G.M. DEMETRIADES	JUNE 1995			
LEADING AND TRAILING EDGES TO BE LEFT AS SHOWN	DESIGN ACTIVITY		SIZE A4	FSCM NO.	DWG NO. / FILE NAME 23
ALL WELDS TO BE SMOOTH - MAINTAIN DIMENSIONAL ACCURACY	CUSTOMER ELECTRICAL DEPARTMENT		SCALE NONE	DATE 19/8/95	SHEET 1 of 1

Appendix H : Error Analysis

H.1 Systematic uncertainties in efficiency

Systematic errors are those which can not be reduced unless the equipment used for measurements are changed. The method and definitions presented here are based on the recommendations of hydraulic turbine testing standards, [BSEN 60041, 1994].

The systematic uncertainty for the turbine efficiency f_{η_T} is computed from individual systematic uncertainties by the root mean squares method from the individual systematic uncertainty in discharge f_Q specific hydraulic energy f_{gH_N} and power $f_{P_{out}}$ using

$$f_{\eta_T} = \pm\left\{\left(f_Q\right)^2 + \left(f_{P_{out}}\right)^2 + \left(f_{gH_N}\right)^2\right\}^{1/2}$$

H.1.1 Calibration of the flow measuring system

The output signal from the pressure transducer connected across the Dall tube is in a voltage form. The flow rate through the Dall tube is therefore calculated using the following equation

$$Q = \kappa\sqrt{voltage} \qquad \{H.1\}$$

Figure H.1 shows the calibration test results for determining the value of κ.

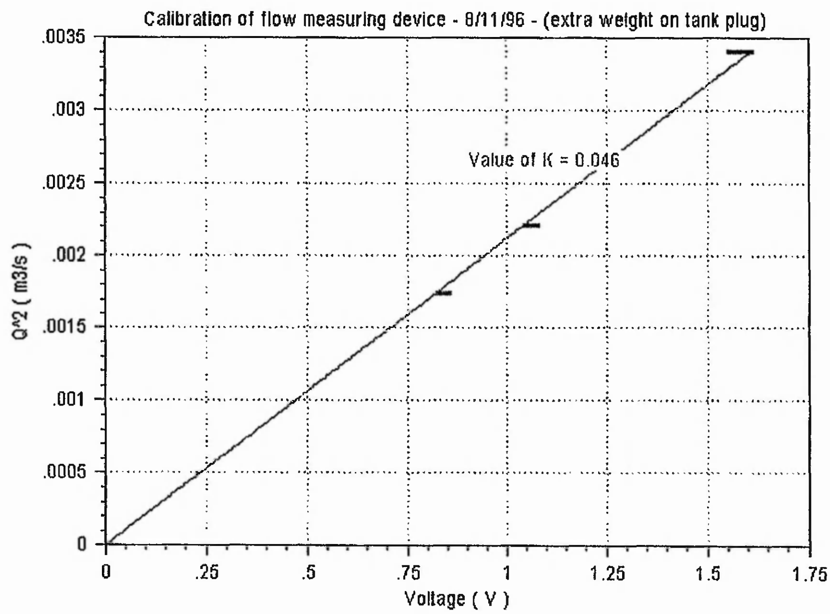


Figure H.1 : Calibration of the flow measuring system.

H.1.2 Analysis and evaluation of uncertainty

The specific hydraulic energy is given by

$$gH_N = \frac{P_1}{\rho} + \frac{V_{in}^2 - V_{out}^2}{2} + gZ$$

Let ε_x to be the absolute systematic uncertainty in quantity *x* (thus the relative systematic uncertainty is $f_x = \frac{\varepsilon_x}{x}$), the relative systematic uncertainty in the specific hydraulic energy is given by

$$f_{gH_N} = \frac{\left\{\left(\varepsilon_{P_1/\rho}\right)^2 + \left(\varepsilon_{V_{in}^2/2}\right)^2 + \left(\varepsilon_{V_{out}^2/2}\right)^2 + \left(\varepsilon_{gZ}\right)^2\right\}^{1/2}}{gH_N}$$

With reference to the test results of *runner 2*, at a head of 2m, the following readings were recorded :

$$P_1 = 0.064 \times 10^5 Pa,\ f_{P_1} = \pm 1\%$$

$$Z_1 = 1.3m,\ \varepsilon_Z = \pm 0.01m$$

$$V_{in} = 0.98m/s,\ f_{V_1} = \pm 3\%$$

$$V_{out} = 0.78m/s,\ f_{V_2} = \pm 3\%$$

V_{in} and V_{out} are inlet and exit velocities to the turbine system. Their uncertainty relies on the uncertainty of the flow measuring system which is equal to ±3%. The absolute systematic uncertainties are as follows

$$\varepsilon_{P_1/\rho} = \pm\frac{0.064 \times 10^5}{1000} \times \frac{1}{100} = \pm 0.064 \frac{J}{kg}$$

$$\varepsilon_{gZ} = \pm 9.81 \times 0.01 = \pm 0.1 \frac{J}{kg}$$

$$\varepsilon_{V_1^2/2} = \pm 0.98^2 \times \frac{3}{100} = \pm 0.029 \frac{J}{kg}$$

$$\varepsilon_{V_2^2/2} = \pm 0.78^2 \times \frac{3}{100} = \pm 0.0183 \frac{J}{kg}$$

The systematic uncertainty of the specific hydraulic energy is therefore calculated as follows :

$$_{gH_N} = \frac{\left\{(0.064)^2 + (0.029)^2 + (0.0183)^2 + (0.1)^2\right\}^{1/2}}{gH_N} = \pm 0.63\%$$

As it has already been determined from the calibration of the flow measuring system, the systematic uncertainty is ±3%. The uncertainty for the power measurement system is equal to 1%.

The overall uncertainty in efficiency is therefore given by

$$f_{\eta_T} = \pm\left\{(0.0063)^2 + (0.01)^2 + (0.03)^2\right\}^{1/2} = \pm 3.22\%$$

Appendix I : Publications Resulting From The Research Presented In The Thesis

I.1 List of Publications

1. Demetriades, G. M.; Williams, A. A. and Smith, N. P. A. 1995a. The use of propeller turbines in low head stand alone micro hydroelectric power generation units. *International Journal of Ambient Energy*. **16**(3), pp. 165-168.

2. Demetriades, G. M. and Williams, A. A. 1995b. The design of a low - cost propeller turbines for stand alone micro - hydroelectric power generation units. *In : Proceedings of the 10th Conference on Fluid Machinery*. Akatémiai Kiadó, Budapest, Hungary. pp. 116-124.

3. Demetriades, G. M., Williams, A. A. and Smith, N. P. A. 1996. A simplified propeller turbine runner design for stand alone micro - hydro power generation units. *International Journal of Ambient Energy*. **17**(3) pp. 151-162.

4. Demetriades, G. M. and Williams, A. A. 1997. A simplified propeller turbine design for direct coupling to an induction generator for village hydroelectric schemes. *In : Proceedings of the Up and Coming in Fluid Machinery Seminar*, Organized by the Power Industries Division's Fluid Machinery Committee of the Institution of Mechanical Engineers. London, November 1997.

Demetriades, G. M.; Williams, A. A. and Smith, N. P. A. 1995a.

The use of propeller turbines in low head stand alone micro hydroelectric power generation units.

International Journal of Ambient Energy. **16**(3), pp. 165-168.

THE USE OF PROPELLER TURBINES IN LOW HEAD STAND ALONE MICRO HYDRO ELECTRIC POWER GENERATION UNITS.

G. M. Demetriades, A. A. Williams, N. P. A. Smith

Micro-hydro Research Group, Electrical and Electronic Engineering Department,

The Nottingham Trent University, Nottingham, UK

1 SYNOPSIS.

The mountainous regions of developing countries offer a great potential for small scale hydroelectric schemes, running as stand alone units. Such schemes with power output less than 100 kW are usually referred to as **micro - hydro power generation units**. For low - **head sites** (available head less than 10m), there is a vast number of suitable sites in countries with less mountainous areas and high rainfall, or extensive irrigation canals.

The present paper introduces the design features of an appropriate **propeller turbine** design. The turbine will be directly coupled to an **induction generator**. The design requirements, materials selection and manufacturing processes are analysed with respect to experiences from pilot projects within the UK and abroad.

2 INTRODUCTION.

The term micro-hydro refers to all the equipment and civil works necessary to exploit a hydropower resource of up to 100 kW. To be considered as a practical proposition, micro-hydro must be competitive with other options such as central grid connections, diesel and petrol generator sets. The following factors need to be taken into account when choosing a suitable turbine for a micro-hydro installation :

* ***Cost***
* ***Reliability***
* ***Efficiency***
* ***Operating speed***

Turbines are devices for converting the energy of water into mechanical energy. The energy of water can be in the form of potential energy or in the form of kinetic energy. Thus, classification of turbines may be either of ***impulse*** or ***reaction*** type.

The essential feature of an ***impulse turbine*** is that there is no change of static pressure as the water passes through the runner. All the pressure energy of the flow is converted into kinetic energy in the form of a jet. In ***reaction turbines***, there is a reduction in the static pressure as the fluid goes through the runner.

In addition to the above classification, turbines may be classified according to the predominant direction of the water through the runner. In a ***radial flow machine*** the flow path is wholly or mainly in the plane of rotation. If, however, the flow direction is parallel to the axis of rotation, then the machine is said to be an ***axial flow*** one. If the flow is partly axial and partly radial the term ***mixed flow machine*** is used.

Large hydraulic turbines have achieved very high hydraulic efficiencies. Unfortunately, it is not advisable to scale down these turbines for micro-hydro applications for many reasons. Principally, the cost per kilowatt does not remain constant with decreasing size. It becomes necessary at some point to simplify the machine's design, often at the expense of hydraulic efficiency, to keep the costs competitive with other energy sources.

Micro-hydro systems comprise four main components : civil works, turbine, control system, generator and transmission.

Civil works. The civil works include a weir to direct the water of a stream into a canal, a settling tank with a trash rack, penstock and power house.

Turbine. The choice of a turbine to be installed on a given site depends on several considerations relating to power demand and its characteristics: the head, the range of water flow over the whole year, the efficiency of the turbine required at any point within the range of flows. Particularly important in developing countries, is the lack of turbine manufacturing capabilities, the cost of setting up such an industry and the provision of spare parts and servicing. In Sri Lanka for example, most micro-hydro sites in the tea estates have high heads (greater than 30m). For such heads, Pelton wheels have been used for a number of years now. Peltric (Pelton Turbine Induction Generator combination) units require only small flow rates to run as opposed to axial or Francis turbines, and these come with a number of jets so as to cope with the fluctuations of water flow all year round or power demand between day and night time. The simplicity of the design and installation, make it possible to use Peltric units anywhere in a remote community. Moreover the entire Peltric technology is available in the country itself and it is locally made at affordable cost. All the Peltric installations in Sri Lanka have been made by local industries. The buckets are made individually by casting (bronze) and then assembled to the main wheel using bolts and nuts. This technique makes it easier to change individual buckets which might be damaged after a certain operating period. The casings and nozzles of the Peltric system are also made locally and assembled by local workshops.

The policy of ITDG (Intermediate Technology Developing Group) is to use the skills and facilities of small local industries found scattered round the remote areas. The benefits are numerous. Small communities are given jobs which provide a reasonable income. Designs tend to be simple. The available means of manufacturing can only cope with straightforward designs. This is also less costly. More people are also trained in the making of turbines which means there will be more people capable of servicing and maintaining these units in the future. Moreover, if the turbines are made locally, repairs can take place locally, thus the time that a unit is out of operation is kept to a minimum, Smith [1].

For heads lower than 30m a few Francis turbines have been used, which have had to be imported from abroad. According to ITDG office in Colombo/Sri Lanka, there is an increasing interest by owners of low head sites in utilising power at heads below 10 m. At the moment Francis or crossflow turbines are used which tend to be bulky and heavy. In addition, since they operate at much lower speeds, they require a speed up mechanism to drive a generator, whereas peltric and axial flow turbines can be connected directly to a generator.

Induction generators. Standard electrical motors are used as generators without a great difficulty. Capacitors are all that are needed to convert motors into generators. In contrast to conventional synchronous generators, motors are very robust, easy to obtain and are cheaper when used for small schemes (up to 30 kW) than synchronous machines, Smith (et al) [2].

Controller. The induction generator controller (IGC) is used to control the induction generator. An electronic load controller manages the flow of electricity. The generator produces the same amount of electricity all the time. The controller directs electricity which is temporarily in excess of demand into heaters called ballast load or even to battery chargers. Like all other components of a micro-hydro unit installed by ITDG, the controller is made locally. ITDG runs courses to train local people in the manufacturing and installation of the IGC's with induction motors run as generators, Smith [3].

3 TURBINE SELECTION.

The selection of the best turbine for any particular hydro site depends on the site characteristics, the dominant factors being the ***head available*** and ***power required***. Figure 1 shows a range of turbines with their operating heads. Selection also depends on the ***speed*** at which the machine is to operate.

The design of a turbine generator unit that would ideally suit the remote village communities, must satisfy many basic requirements.

Low cost manufacture is of first priority. It should be self contained, light enough to be carried by 2 or 3 men and able to withstand rough handling while being transported. The next important factor is the ease of installation. This is important because skilled labour is not available in remote areas and sending technical support from urban areas will be prohibitively expensive. It is best if the machine can be installed by the villagers after a demonstration and requires minimum maintenance.

In general, reaction turbines rotate faster than impulse turbines given the same head and flow conditions. These high speeds have the very important implication that reaction turbines can often be directly coupled to a generator without any speed increasing drive system i.e. belt drives or gearboxes.

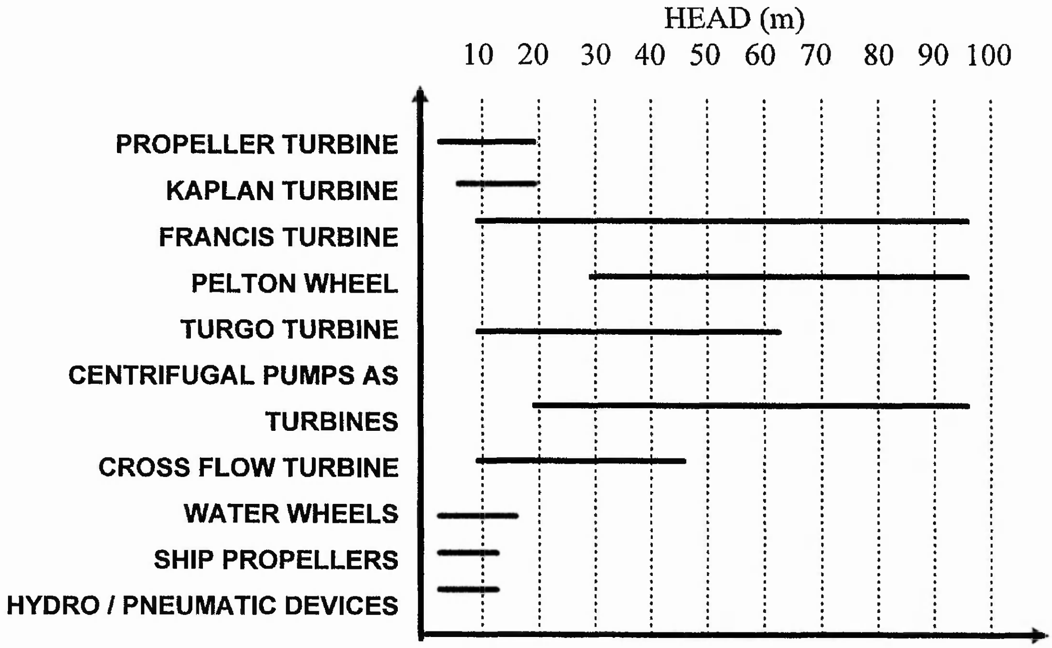


Figure 1: Range of operating heads of various turbine types.

Water wheels are easy to manufacture but they have large dimensions and run at very low speeds. Cross flow turbines are relatively easy to make but they are bulky and require a speed increasing drive system since they operate at low speeds.

Ship propellers are relatively expensive to buy and are not readily available in developing countries. In addition, a very careful consideration is required to correctly match the propeller to the sites specifications. Hydro/pneumatic devices (see Bellamy [4] and Gould [5]) though considered to be very efficient for low head applications, still are difficult to make and maintain in the environments concerned. Either of these have not been widely used.

Propeller turbines give the best performance when operated at such low heads, but their design and manufacture is relatively complicated. Though all reaction turbines are subjected to the danger of cavitation and tend to have poor part flow efficiency characteristics, still their high speeds make them ideal for low heads since they can be compact and directly coupled to an induction generator.

4 GENERAL DESIGN CRITERIA.

To satisfy the needs of the users and the variation in materials and skills in different countries, the design should be very flexible and basic so that it can be easily adapted. This means that there is not such a thing as the best design. The design should be a compromise between what is good hydraulically and what is easy and cheap to make. The general criteria for such a design are :

1) Low cost with minimum payback period - competitive with other energy options.

2) Easy to manufacture in developing countries with limited workshop facilities.

3) Use of materials available in the country.

4) Easy to transport and install.

5) Easy to operate by unskilled people.

6) Efficiency above 65%.

7) Minimum maintenance requirements - high corrosion and silt resistance.

8) Cavitation free operation.

5 DESIGN LAYOUTS.

Basically there are two kinds of layouts : ***open flume** type* and ***tube turbines*** (figures 2 and 3 respectively).

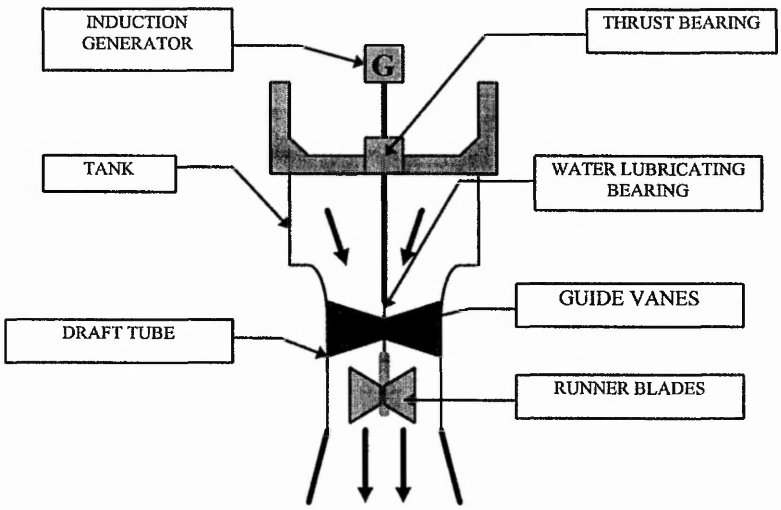


Figure 2: Open flume design layout.

The ***open flume type,*** generally consists of a big tank at the end of a channel with the turbine installed at the bottom of the tank. In this type of layout no penstock is required, but a significant amount of civil works is needed. The ***tube turbine*** layout consists of a small section of a pipe in which the turbine sits. A small penstock is used and since both turbine and generator come as a compact unit, installation is easier.

For a cheap and easy to maintain design it is preferable to install the generator out of reach of water and have it directly driven by the turbine shaft. The open flume type is advantageous over the tube type in this respect since a seal is not required between the generator and the turbine.

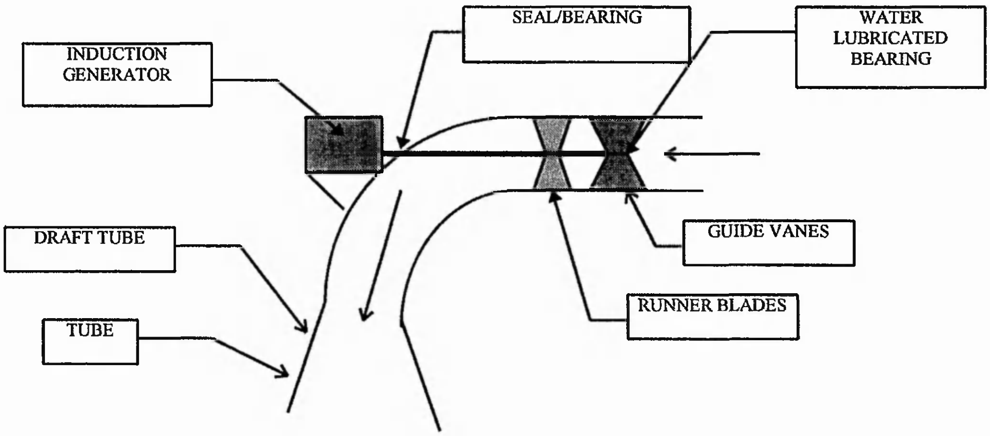


Figure 3: Tube design layout.

6 BLADES PROFILES - MANUFACTURING.

Low head axial flow turbines have been designed with a variety of blade numbers. 4 blades (Heitz [6]; Viani [7]) is common but many designs go up to 8 (Ranatunga [8], Susanto [9]). Nechleba [10] suggests 3 blades for a head of 5 m. According to Stepanoff [11], "good performance is obtained with 3 or 4 blades. With 2 blades, the ratio vane chord length to vane spacing is too low for good efficiency, and with 5 blades both the efficiency and the head capacity drop." However, by increasing the number of blades, the design becomes heavier, more complicated and the free area of flow is restricted to a considerable extent. It is therefore very important to investigate the effects of the number of blades on the performance of axial low head water turbine before finalising any design procedures.

Experiences from Sri Lanka suggest that it is difficult to fabricate blades by bending and twisting thin sheets of metal and the accuracy of the desired shapes is never guaranteed. A milling machine can be used instead to make blades of any shape at very high accuracy. However, there are various drawbacks mainly due to the limited availability in developing countries of such sophisticated machine tools.

Another way of making the hub-blades assembly is casting. Both the hub and the blades can be cast to a very high degree of accuracy and if mass production is required, this seems to be the best technique to use. The runner can be made as a whole casting or the hub can be cast separately from the blades and then assembled using bolts and nuts. The advantages of making the blades separately from the hub are :

* ***Simplicity in the design of the mould.***

* ***The blades will be interchangeable.***

* ***The design of the hub can be such that the blades can be bolted at different locations. Therefore, the blades will be adjustable to cope with the variations of flow between high and low rainfall.***

The casting of bronze is widely used in developing countries, another reason in favour of using this technique. In addition, by polishing, a very smooth surface finish may be obtained and thus minimise friction losses and produce a smooth flow.

According to Turton [12], the blades should be of long chord and low camber at the tip and relatively thick and cambered at the root. Wallis [13] suggests that "a constant thickness plate rounded at the leading and trailing edges can be bent around the camber line. Although not as efficient as an Aerofoil section, cambered plate blades are simple and cheap to make." In addition, Eckert [14] proved that for low duty fans, the performance of a cambered circular plate was similar to that of an Aerofoil.

Camber plates are usually assumed to behave in a similar way to orthodox aerofoils with the same camber, provided the inlet flow is approximately tangential to the leading edge of the blade. When the angle of incidence exceeds $\pm 3^{\circ}$, local flow separations at the leading edge will reduce efficiency. The intention here is to try to achieve the maximum best performance using a simple and cheap method.

7 GUIDE VANES.

There are two possible guide vane layouts : ***radial*** and ***axial***.

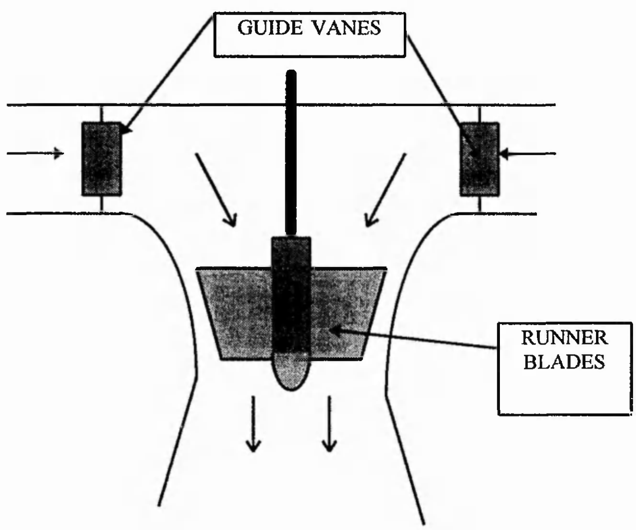


Figure 4: Radial guide vanes.

A simple approach, which is most feasible with the radial arrangement, is to incorporate the guide vanes into the concrete base of the intake.

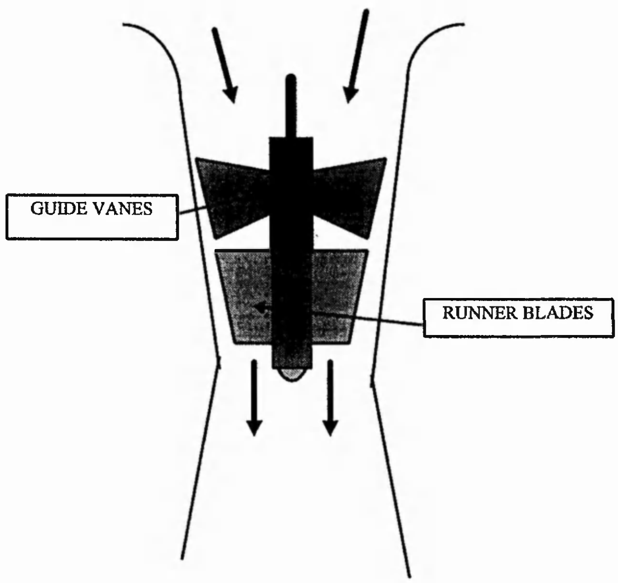


Figure 5: Axial guide vanes.

8 INSTALLATION-TESTING OF A LOW HEAD AXIAL FLOW TURBINE IN SRI LANKA.

The blades of this particular design were made by twisting and bending flat sheets of metal to the desired shape. The blades were then welded to the hub. Bending and twisting pieces of metal of that size to the desired shape was a difficult task to achieve with the limited workshop facilities in the area. In addition welding the blades on the runner affected the rotordynamic efficiency of the runner. and at high speeds, excessive vibrations resulted which cause rapid wear of bearings.

Test results. During the tests carried out, the maximum efficiency obtained was 33%. The inaccuracies of the blade angle formation and the 90° sharp intake at the bottom of the feed tank located just above the guide vanes were thought to be the major causes of inefficiency. Sharp edges cause an enormous whirling to the fluid going past them thus making the flow very disturbed and unsteady. This result shows that the guide vane design did not produce the desired effect The conditions designed for, did not match the real operating conditions. The performance of the unit was therefore lower than expected. The surface finish of the blades was very rough, another factor that contributed towards the disturbance of the flow.

In addition, the designed flow rate of the turbine was $50^{l}/_{s}$ where as the actual flow rates did not even reach the value of $40\,^{l}/_{s}$.

9 GENERAL CONCLUSIONS.

a) Prefabrication ensures easy assembly at minimum time. Since electrical power is not available in remote areas, it is essential that everything is made beforehand in a workshop. Small sections are easier to handle and carry through rough terrain and mountainous areas.

b) Accuracy of construction. At least once before the turbine is taken on site, all the sections have to be assembled and each part clearly marked to facilitate assembly. If there's a need for any modifications, these are easier to carry out in a workshop rather than on site.

c) Availability of water. The design should be such as to cope with fluctuations in the flow.

d) Stream flow. Always try to cause the least degree of disturbance in the flow. Avoid sharp intakes and use smooth curved polished surfaces.

d) Simplicity of civil works.

e) Trash racks, settling tanks and vent pipes are very important. It is often the case that small stones, leaves and plastic bags flow down the penstock. As a result, flows are restricted and sometimes due to the rapid increase of pressure, there is a danger of pipe bursts.

f) The cost of micro-hydro units should be minimum. The majority of the expenses is covered by charity organisations such as The Rotary Club. However, it is often the case that the village communities themselves contribute a considerable amount to the total cost of a project.

g) Minimum maintenance requirements - durability - reliability - cheap spare parts locally available. Since every village is responsible for maintenance and service of its own plant, the maintenance procedures have to be as simple as possible. In the case that spare parts are required, these should be available locally in order to minimise the time the community is out of power and to make sure that the costs are kept low.

h) The participation of the whole village community in a micro-hydro project is of great importance. This involves the measurements of available head, flow rates and the carrying out the majority of the civil works.

10 FURTHER WORK.

a) Further testing of existing axial flow micro-hydro units:

* A turbine made in Vietnam.

* A turbine already installed at Pedly Wood Conversation Trust.

* A turbine already installed at the River Wandle in London.

b) Carry out a detailed analysis of the design of turbine blades made of thin sheets of metal bent and twisted at the desired shape.

c) Study the effects on turbine design and efficiency of :

* Blade number.

* Hub to tip ratio.

* Coefficient of lift (C_L) and coefficient of drag (C_D).

* Blade spacing and chord length.

d) Design, manufacture and test a prototype axial flow low head turbine.

The first prototype in mind, will be developed using a flat constant thickness plate, bend and twisted to the desired angles and then bolted to the hub. The intention is to come up with a design whose blades will be interchangeable in order to be able to test a number of blade designs at different angles with the same hub/tip ratio. If the blades are to be welded, care should be taken to maintain an "even" amount of welding round the weld in order to minimise the effects of excessive vibration due to unbalanced loads.

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Demetriades, G. M. and Williams, A. A. 1995b.

The design of a low - cost propeller turbines for stand alone micro - hydroelectric power generation units.

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Akatémiai Kiadó, Budapest, Hungary. pp. 116-124.

THE DESIGN OF LOW - COST PROPELLER TURBINES FOR STAND ALONE MICRO - HYDRO ELECTRIC POWER GENERATION UNITS

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SUMMARY

The mountainous regions of developing countries offer a great potential for small scale hydroelectric schemes running as stand alone units, i.e. not connected to the grid. Such schemes with power output less than 100 kW are usually referred to as **micro - hydro** power generation units.

The present paper introduces a simplified approach to the design of an appropriate **propeller turbine** to be directly coupled to an induction generator for micro - hydro power generation. Emphasis is given to the hydraulic design of the runner which has blades made of constant thickness metal sheets, based on a circular arc profile. The use of such a shape, is ideal for low - cost manufacturing in developing countries as it utilizes as much of the local skills and materials available as possible.

1. INTRODUCTION

Micro - hydro is an indigenous, renewable source of energy, the development of which is generally not associated with adverse environmental consequences. It does not disrupt the environment like massive hydro projects which demand the construction of large reservoirs, and, quite often, the resettlement of large number of people. Small hydroelectric power generation is especially suited for decentralized village electrification in the mountainous regions of developing countries, where large installations or grid extensions are not practical, i.e. too remote, too expensive, or too time consuming.

The setting up of isolated stand alone micro - hydroelectric power generation units, especially where the extension of the national grid is not feasible, has long being acknowledged as an appropriate way of providing power to a significant number of remote rural communities. A general applicable modus operandi for the implementation of appropriate hydroelectric power schemes is being developed.

Micro - hydro can provide energy through direct mechanical power or by linking a turbine to an ac generator to produce electrical power. It is now recognized as a reliable, least - cost option for providing energy to rural communities where favourable physical conditions exist. Micro - hydro schemes can meet the power requirements of many small scale industrial processes such as milling of flour, rice hulling, coffee processing, sugarcane crushing, sawmills, bakeries and small workshops. In addition, micro - hydroelectric power can replace kerosene for lighting and reduce the dependence on wood for cooking.

2. TURBINE SELECTION.

Pelton turbines are widely used for **high** and **medium head** micro - hydroelectric schemes, Smith et al [1], whereas, **crossflow turbines** are suitable for **medium heads,** Hothersall [2]. In addition, **centrifugal pumps run as turbines** can also be used for medium head schemes, Williams [3]. Apart from power generation, some versions of these turbines have been designed so that they can be manufactured in developing countries and thus use as much local skills and materials as possible, while providing a reliable, easily maintained system suited for the needs of a remote community, Meier [4].

For **low head sites** (available head less than 5m), there is a vast number of suitable sites in countries with less mountainous areas and high rainfall, or extensive irrigation canals. The high initial cost to import already existing propeller turbine units and the inability of the local communities to cope with their maintenance requirements, makes them unsuitable for isolated schemes.

The type of turbine to be installed on a given site depends on several considerations relating to power demand and its characteristics: the head, the range of water flow over the whole year, the efficiency of the turbine required at any point within the range of flows. Particularly important in developing countries, is the lack of turbine manufacturing capabilities, the cost of setting up such an industry and the provision of spare parts and servicing. Figure 1 shows a range of turbines with their operating heads.

As a general rule the selection of the most suitable turbine for any particular hydro site depends on the site characteristics, the dominant factors being the **head available** and **power required,** Holland [5] . Selection also depends on the **speed** at which the machine is to operate.

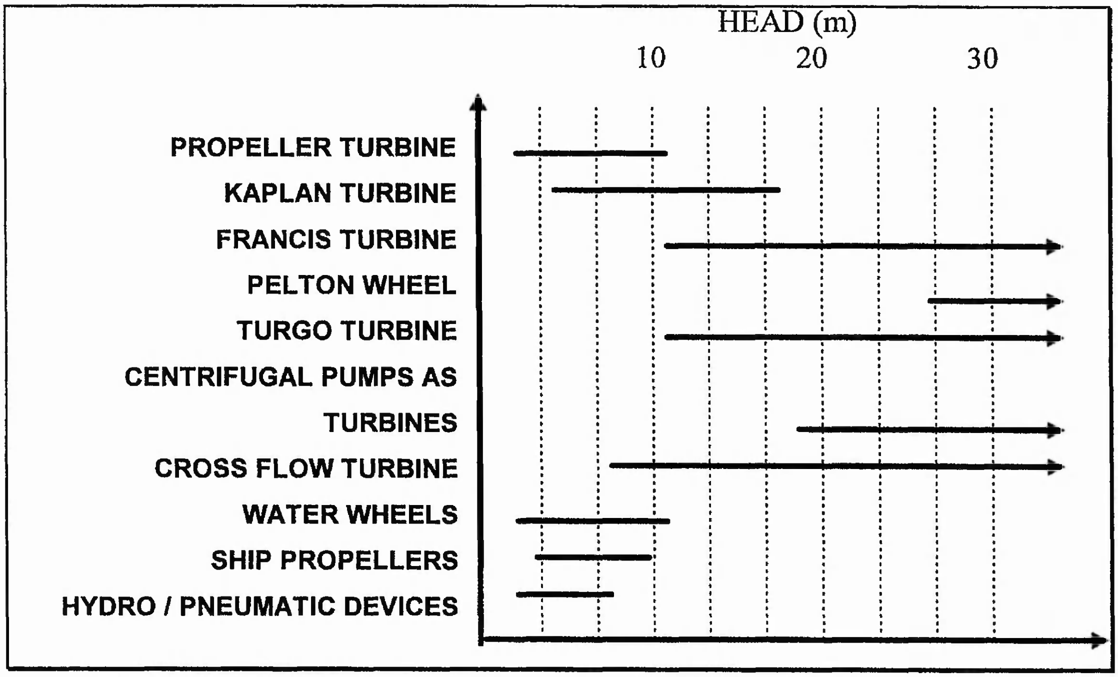

Figure 1 : Range of operating heads of various turbine types.

In general, reaction turbines rotate faster than impulse turbines given the same head and flow conditions. These high speeds have the very important implication that reaction turbines can often be directly coupled to an ac generator without any speed increasing drive system i.e. belt drives or gearboxes.

Water wheels are easy to manufacture but they have large dimensions and run at very low speeds. Crossflow turbines are relatively easy to make but for low head sites, they are bulky and require a speed increasing drive system since they operate at low speeds. Ship propellers are relatively expensive to buy and are not readily available in developing countries. In addition, a very careful consideration is required to match the propeller to the sites specifications correctly, Huetter [6].

Hydro / pneumatic devices are difficult to make and maintain in developing countries. Pilot project designs have proved to require materials and manufacturing techniques not easily found in developing countries, Gould [7], at costs well above the other alternatives available.

Propeller turbines give the best performance when operated at such low heads, but their design and manufacture is relatively complicated. Though all reaction turbines are subjected to the danger of cavitation and tend to have poor part flow efficiency characteristics, still their high speeds make them ideal for low heads since they can be compact and directly coupled to an ac generator, Demetriades[8].

3. TURBINE LAYOUT.

In order to decide which turbine layout to use, the following factors need to be taken into account :

- Seal requirements - problems with silt and sand present in the water.
- Prefabrication - ease of transportation.
- Assembly procedure on site - accuracy of shaft and bearing alignment on site.
- Shaft length.
- Civil works required.
- Use of thrust bearings.

The above factors are not only related to the complexity of design and cost of the turbine. Moreover, maintenance and life expectancy do rely on these as well. Basically there are two kinds of layouts : **open flume type** and **tube turbines** (figures 2 and 3 respectively).

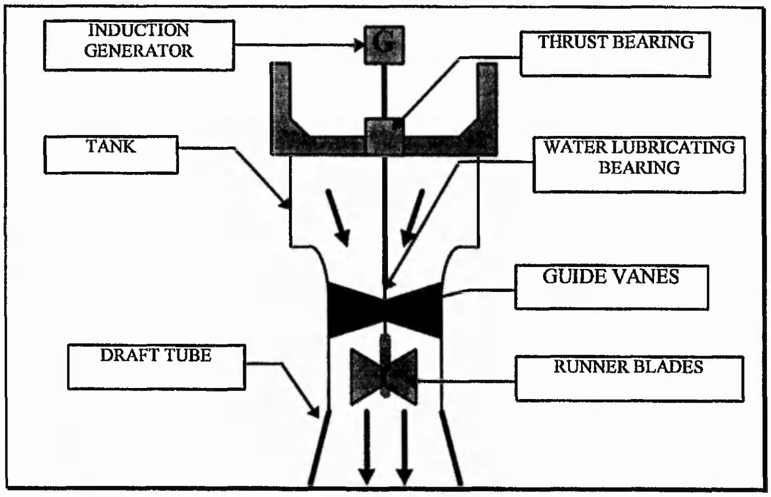

Figure 2: Open flume design layout.

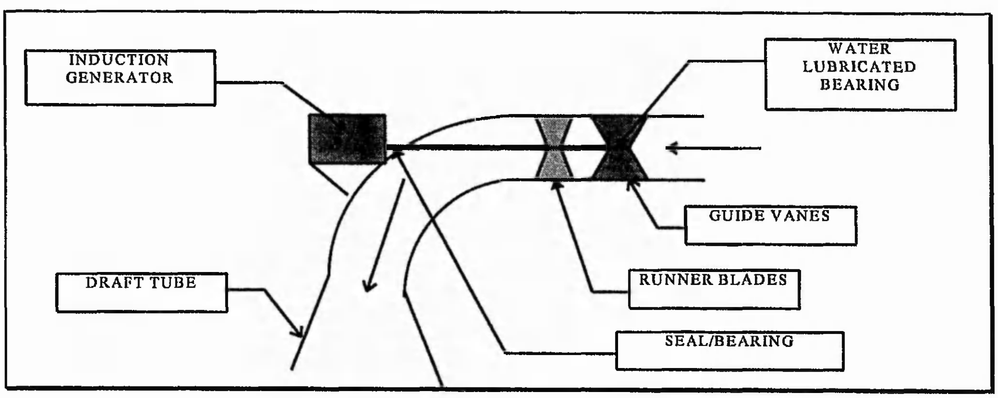

Figure 3: Tube design layout.

The **open flume** type, generally consists of a big tank at the end of a channel with the turbine installed at the bottom of the tank. In this type of layout no penstock is required, but a significant amount of civil works is needed. The **tube turbine** layout consists of a small section of a pipe in which the turbine sits. A small penstock is used and since both turbine and generator come as a compact unit, installation is easier. For a cheap and easy to maintain design it is preferable to install the generator out of reach of water and have it directly driven by the turbine shaft. The open flume type is advantageous over the tube type in this respect since a seal is not required between the generator and the turbine.

4. RUNNER DESIGN.

The primary purpose of a water turbine is power generation. It is therefore essential during the design process to determine and assess all the parameters involved in order to maximize the work done by the flow of water through the runner. For a given flowrate and head, the process of designing a propeller turbine is based on the procedures followed for any axial flow turbomachine and involves the following parameters :

- Turbine design speed,
- Turbine specific speed,
- Runner outside diameter,
- Hub to Tip ratio,
- Number of blades used,
- The meridional velocity of flow,
- The blade inlet and outlet angles,
- The fluid inlet and outlet angles,
- Chord to spacing ratio,
- The blade setting angle,
- The shape of the blade to be used,
- The coefficient of lift and drag of the blade in the cascade,
- Incidence angle,
- Deviation angle,
- Camber angle,
- Solidity ratio,

4.1. Physical dimensions of the runner.

The physical dimensions of the runner are directly related to the specific speed of the turbine. The design speed of the turbine is determined by the type of the ac generator used. Synchronous generators usually run at 1500 and 3000 rpm. However, for low head micro - hydroelectric schemes, with power outputs of a few kW's, induction motors are generally used as generators, Smith, [9], which can be run at 750, 1000, 1500 or even 3000 rpm. The specific speed of the turbine is given by :

$$\boxed{Ns = \frac{N\sqrt{Q}}{H^{3/4}},\ (\mathrm{min}^{-1})} \qquad \{1\}$$

The values of Z, (D_H / D_T) and K_{u1} can be obtained from table 1, as adopted from Bohl [10]. In addition, the tip peripheral velocity of the runner is given by :

$$\boxed{u_T = k_{u1}\sqrt{2\,g\,H},\ (m/s)} \qquad \{2\}$$

The peripheral velocity is also given by :

$$\boxed{u = \omega\, r,\ (m/s)} \qquad \{3\}$$

Combining equations {2} and {3}, the value of D_T is :

$$\boxed{D_T = \frac{84.6\ k_{u1}\sqrt{H}}{N},\ (m)} \qquad \{4\}$$

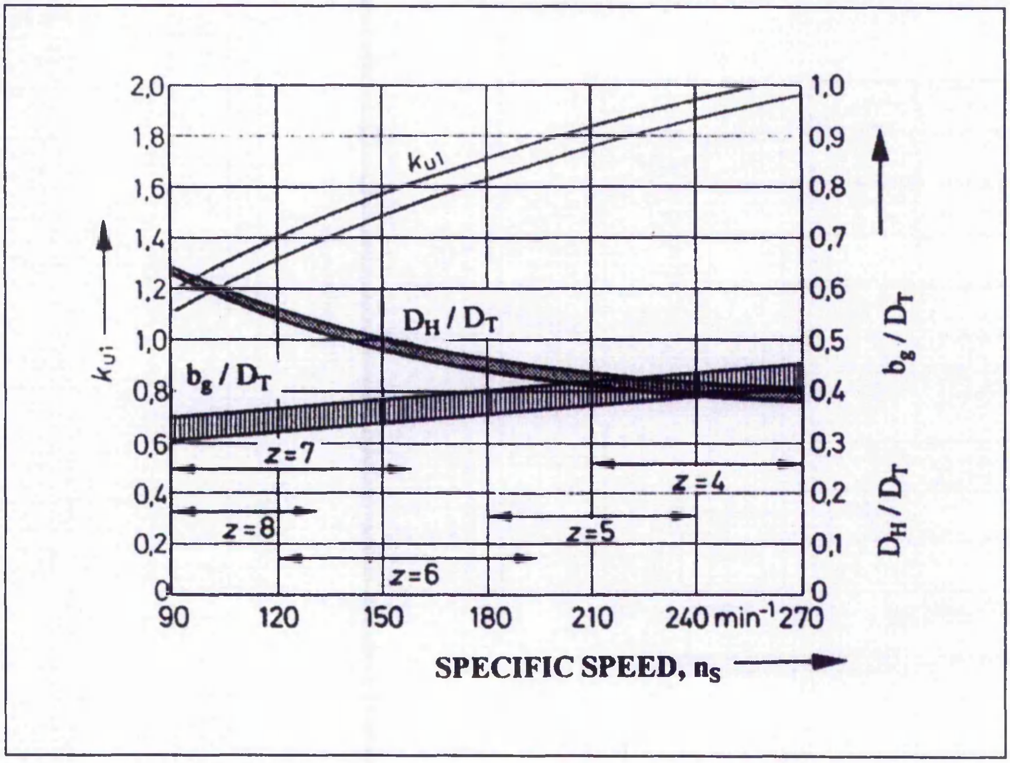

Table 1. Runner dimensions, (adopted from Bohl [10]).

Following recommendations on the design of turbomachines by Turton [11] it is assumed that V_m is constant at the inlet and outlet of the runner. For maximum efficiency, the whirl component of the fluid at exit (V_{u2}) is assumed to be zero, Sayers [12]. In addition, being an axial flow machine, changes take place at the same radius between inlet and outlet, therefore,

$$\boxed{u = u_1 = u_2 = \omega\, r, \ (m\ /\ s)} \quad \{5\}$$

Having determined all the above, the velocity diagrams with regard to the fluid path through the runner can be drawn as shown in figure 4.

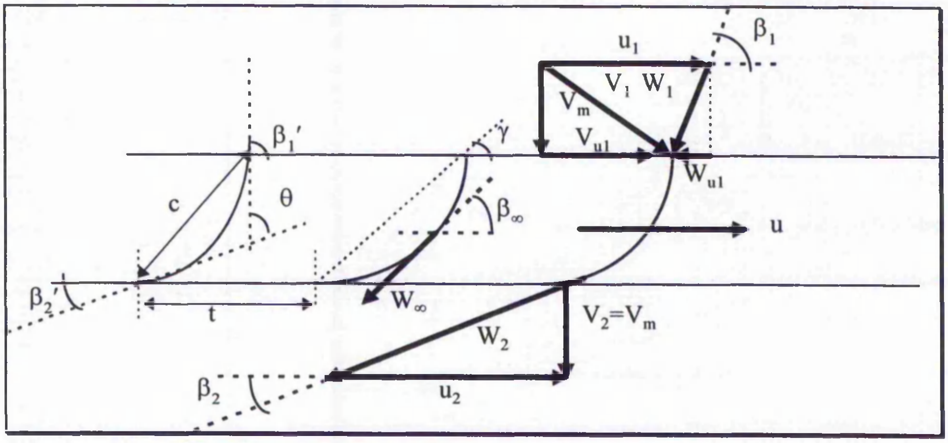

Figure 4. Velocity diagrams at the inlet and outlet of the turbine cascade.

4.2 Runner blade design.

For the design of axial flow hydraulic turbine blades, aerofoil shape profiles are commonly used. Water turbine blades tend to be of long chord at the tip and low camber and relatively thick and cambered at the root. However, such shapes require manufacturing techniques and skills that are not easily found in developing countries. The use of constant thickness sheet of metal, bent and twisted at the desired shape is the ideal alternative to conventional aerofoil blade design with respect to fabrication simplicity and manufacturing costs.

From experiments conducted on both blades made of constant thickness sheets of metal and aerofoil sections, Eckert, [13], showed that the same efficiency has been attained by both shapes, provided that the curvature i.e. camber and twist are preserved. The results from these tests are valid for Reynolds numbers within the range of 100,000 to 500,000.

Figure 5 shows the lift and drag forces on one blade in the cascade. L acts in a direction perpendicular to the average relative velocity, and is given by :

$$\boxed{L = \frac{1}{2}\rho\, C_L\, W_\infty^{\,2}\ [\text{blade area}],\ (N)} \qquad \{6\}$$

The drag force, acts in the same direction as W_∞ and is given by :

$$\boxed{D = \frac{1}{2}\rho\, C_D\, W_\infty^{\,2}\ [\text{blade area}],(N)} \qquad \{7\}$$

Both L and D can be resolved into axial and tangential components and from the analysis of these forces and the change of angular momentum in the axial direction, the coefficient of lift for an ideal blade, i.e. with zero drag force, can be determined. This resultant lift coefficient is given by :

$$\boxed{C_L = 2\left(\frac{t}{c}\right)[\cot\beta_1 - \cot\beta_2]\sin\beta_\infty} \qquad \{8\}$$

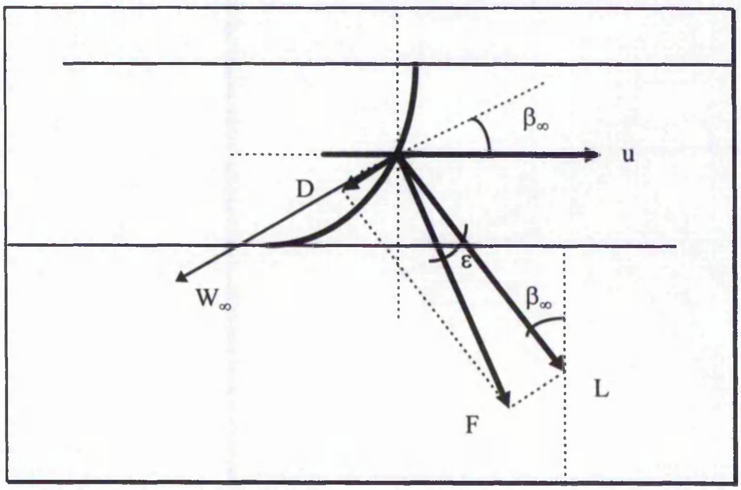

Figure 5. Force analysis of a blade in a cascade.

From the velocity diagrams shown in figure 4, it is possible now to determine the value of C_L in terms of relative velocities and solidity ratio, as shown below :

$$\boxed{C_L = 2\left(\frac{t}{c}\right)\frac{\Delta W_u}{W_\infty}} \qquad \{9\}$$

Typical solidity ratio values given by Sigloch [14], suggest that these should lie between 0.6 and 1.2. The value of 1.2 represents the highest limit with cavitation free operation.

The values of C_L can be as low as 0.2 at the tip whereas at the hub approach the value of 1. The simplest profile to be used concerning fabrication is the circular arc with maximum camber (f) at 50 %c. What is left now, is to determine the value of the blade inlet and outlet angles, β_1' and β_2', which define the blade camber, given by :

$$\boxed{\vartheta = \beta_1' - \beta_2', (^\circ)} \qquad \{10\}$$

4.3 Blade geometry.

From what has been outlined so far, the flow deflection is equal to β_1 - β_2 and thus the circular arc with the above inlet and outlet angles has to be realized at infinite number of blades. However according to Weinig [15], an excessive blade bending should be selected in such a way that $\theta > (\beta_1 - \beta_2)$.Weinig [15] has also showed that the ratio $(\beta_1 - \beta_2) / (\beta_1' - \beta_2')$ relates to the blade setting angle and the solidity ratio, and values of this ratio are selected for every individual geometry according to the fluid and blade angles. Assuming that incidence and deviation angles are of the same value, the geometry of the blade will be given by :

$$\boxed{R = \frac{c}{2\sin(\frac{\vartheta}{2})}, (m)} \qquad \{11\}$$

and the maximum camber located at 50 % of c, given by :

$$\boxed{f = \frac{c^2}{8R}, (m)} \qquad \{12\}$$

Figure 6 shows the geometry of a circular arc blade profile, based on the method analysed by Eck [16].

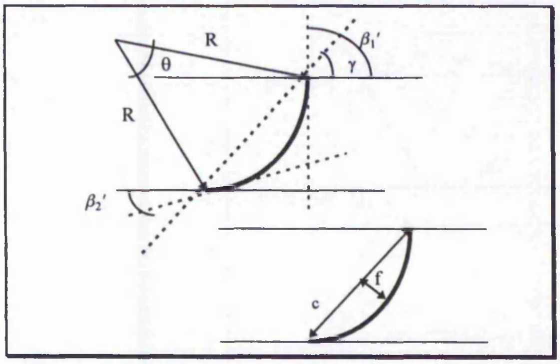

Figure 6. Blade geometry of circular arc profile.

5. CONCLUSIONS.

A simplified method for the hydraulic design of propeller turbine runners has been presented. The important feature of this design methodology, is the use of simple geometry circular arc blades made of constant thickness sheets of meal, as opposed to conventional design approaches where aerofoil profiles are used. The blades will be given a certain camber and twist following the conventional way of designing hydraulic turbines.

The design method described, is based on design methodologies applied for the design of axial flow fans, pumps and water turbines. With the current research project, preliminary design studies may be conducted to effectively guide designers in developing countries for the development of low - head micro - hydroelectric power generation schemes for rural electrification.

The aims of this design methodology are to :

- Provide the basis for further investigation on the use of simple shape runners blades for low head propeller turbines.
- Encourage the design and manufacture of low head propeller turbines by local workshops in developing countries.
- Utilize as much of local skills and materials available in developing countries and thus promote the development of low - cost, low - head micro- hydroelectric power generation schemes.

6 NOMENCLATURE

c	Chord length,	(m)
c / t	Chord to spacing ratio	
C_D	Drag coefficient,	
C_L	Lift coefficient,	
D	Runner diameter,	(m)
	Drag force,	(N)
D_H / D_T	Hub to tip ratio	
F	Resultant force between lift and drag,	(N)
f	Maximum camber	(m)
g	Acceleration of gravity,	(m/s^2)
H	Head	(m)
k_{u1}	Coefficient of peripheral velocity at tip,	
L	Lift force,	(N)
N	Turbine speed,	(rpm)
N_s	Turbine specific speed,	($min.^{-1}$)
Q	Flow rate,	(m^3 / s)
R	Blade radius of curvature	(m)
t	Blade spacing,	(m)
u	Peripheral velocity,	(m / s)
V	Absolute velocity,	(m / s)
W	Relative velocity,	(m / s)
Z	Number of blades,	

Greek letters

β	Fluid angle,	(°)
β'	Blade angle,	(°)
γ	Blade setting angle,	(°)
δ	Deviation angle, ($\beta'_2 - \beta_2$)	(°)
ε	Gliding angle,	(°)
θ	Blade camber angle, ($\beta'_1 - \beta'_2$)	(°)
i	Incidence angle, ($\beta_1 - \beta'_1$)	(°)
ρ	Density of water,	(1000 kg / m^3)
σ	Solidity ratio,	(c / t)
ω	Rotational speed,	(rpm)

Subscripts

∞	Average direction of flow,
H	Hub section,
T	Tip section,
m	Meridional direction of flow,
u	Peripheral,
1	Inlet,
2	Outlet,

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A simplified propeller turbine runner design for stand alone micro - hydro power generation units.

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A SIMPLIFIED PROPELLER TURBINE RUNNER DESIGN FOR STAND ALONE MICRO - HYDRO POWER GENERATION UNITS

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1. SYNOPSIS.

In most developing countries, the vast majority of potential micro - hydro power generation sites, i.e. with power outputs up to 100 kW, are found in areas with high rainfall or extensive irrigation works with small canal drops. These sites, where the available head does not exceed 5 m, are usually referred to as low head sites.

The present paper introduces a simplified design of a propeller turbine suitable for direct coupling to an induction generator. The use of such a unit, is a promising technology for setting up low - head power generation schemes for village electrification in developing countries. Emphasis is given to the hydraulic design of the runner blades which are made of constant thickness sheets of metal. The use of such a shape is ideal for low cost manufacturing in developing countries as it enables local skills and materials to be used.

2. INTRODUCTION.

Due to their relatively high specific speeds, propeller turbines can be compact and operate at relatively high speeds. Other hydraulic machines available for utilizing hydropower from low head sites, such as water wheels and cross flow turbines, tend to be bulky, run at very low speeds and therefore require speed increasing mechanisms, such as gearboxes and belt drive systems, in order to drive a generator (1).

A directly coupled turbine - generator unit, offers several advantages over belt drive systems and gearboxes, such as improved system efficiency, increased turbine and generator bearing lifetime, reduced costs, lower maintenance requirements and simplicity of installation. In addition, the cost and reliability can be further improved when induction motors run as generators are used instead of synchronous generators (2).

Induction machines are designed for continuous operation and require less maintenance than synchronous generators, they are robust and for powers less than 20 kW are less expensive and more easily available.

The successful development and use of low cost, locally made turbine units directly coupled to induction generators can be seen by the increasing number of Peltric (Pelton Turbine Induction Generator) micro - hydro systems (3).

3. TURBINE - GENERATOR SPEEDS.

Synchronous generators run at a fixed speed which is known as synchronous speed. This may be calculated from :

$$N_{synch.} = \frac{120}{p} f, \qquad \{1\}$$

Induction motors run at speeds slightly lower than synchronous speed, due to slip, s. Typical values for slip are between 0.02 and 0.05 at full load. The motor speed is given by :

$$N_m = \frac{120}{p} f(1-s), \qquad \{2\}$$

and slip is found from :

$$s = \frac{N_{synch.} - N_m}{N_{synch.}}, \qquad \{3\}$$

It is evident from the above, that for a 4 pole induction motor running at 50 Hz with a rated speed 1445 revs. $min.^{-1}$, the synchronous speed will be 1500 revs. $min.^{-1}$ and the speed that will have to be run as a generator would be 1555 revs. $min.^{-1}$, as the slip becomes negative when it runs as a generator (2). Slip is approximately the same for both motor and generator mode. Figure 1 shows the torque - speed characteristics for induction motors and generators.

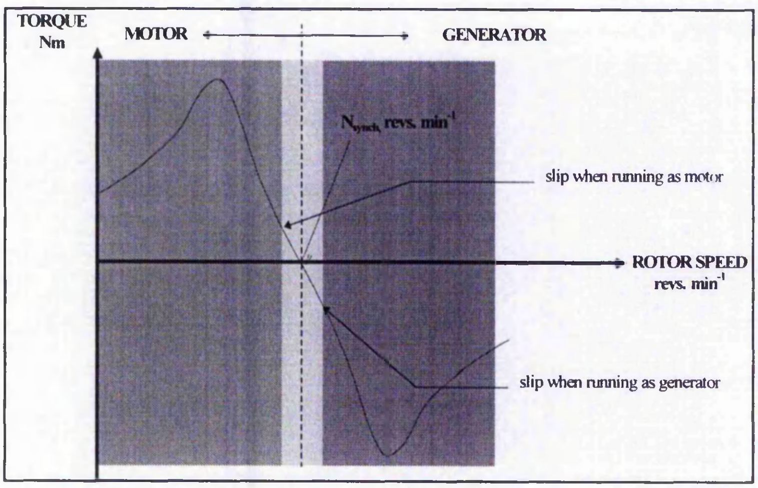

Figure 1 : Torque - speed characteristics of induction machines.

4. THE USE OF FLAT METAL SHEETS FOR THE RUNNER BLADES.

For the design of axial flow hydraulic turbine blades, aerofoil shape profiles are commonly used (4). However, such shapes require manufacturing techniques and skills that are not easily found in developing countries. The use of constant thickness sheets of metal, bent and twisted to the desired shape is the ideal alternative to conventional aerofoil blade design with respect to fabrication simplicity and manufacturing costs (5). From experiments conducted on both blades made of constant thickness sheets of metal and aerofoil sections, it has been shown that the same efficiency has been attained by both shapes (6), provided that the curvature i.e. camber and twist are preserved.

The use of casting, that is widely used in the countries of South East Asia, is currently being investigated to ensure a more accurate method of manufacturing of the turbine units. However, the wide diversity of skills, standards and materials suggest that fabrication of the turbine runner with constant thickness sheets of metal is the best option, with respect to simplicity, ease of manufacturing and cost effectiveness. However, the accuracy of manufacture is an essential factor in order to avoid blade shape imperfections that would result to cavitation.

5. RUNNER DESIGN.

The physical dimensions of the runner are directly related to the dimensionless specific speed of the turbine (7) given by :

$$N_s = \frac{N\sqrt{Q}}{(gH)^{3/4}}, \qquad \{4\}$$

The theoretical spouting velocity of water at the tip of the runner is $\sqrt{2gH}$, and according to (8), the tip velocity of the runner is given by

$$u_T = k_{u1}\sqrt{2\,g\,H}\,, \qquad \{5\}$$

In addition, the tip rotational speed of the runner is

$$u_T = \omega\,\frac{D_T}{2}\,, \qquad \{6\}$$

With reference to (8) and (9), combining equations {5} and {6}, and since $\omega = 2\,\pi\,N$, the value of D_T is given by :

$$D_T = \frac{k_{u\,1}\sqrt{2\,g\,H}}{\pi\,N}, \qquad \{7\}$$

Figure 2 shows the lift and drag forces on one blade in the cascade. The lift force acts in a direction perpendicular to the average relative velocity, and is given by :

$$L = \frac{1}{2}\rho\,C_L\,W_\infty^2\,A, \qquad \{8\}$$

and the drag force, acts in the same direction as W_∞ and is given by :

$$D = \frac{1}{2}\rho\, C_D W_\infty^2\, A, \qquad \{9\}$$

Both L and D can be resolved into axial and tangential components and from the analysis of these forces and the change of angular momentum in the axial direction, the coefficient of lift for an ideal blade, i.e. with zero drag force, can be determined, and is given by :

$$C_L = 2\left(\frac{t}{c}\right)\left[\cot\beta_1 - \cot\beta_2\right]\sin\beta_\infty \qquad \{10\}$$

For the above design approach, it is assumed that V_m is constant at the inlet and outlet of the runner and that for maximum efficiency, the whirl component of fluid flow at exit (V_{u2}) is zero (4). Typical solidity ratio values given by (10) suggest that these should lie between 0.6 and 1.2. The value of 1.2 represents the highest limit for cavitation free operation. The simplest profile for fabrication is the circular arc with maximum camber at 50% of the chord length. From the velocity diagrams shown in figure 3, it is possible now to determine the value of C_L in terms of relative velocities and solidity ratio, as shown below :

$$C_L = 2\left(\frac{t}{c}\right)\frac{|W_{u1} - W_{u2}|}{W_\infty} \qquad \{11\}$$

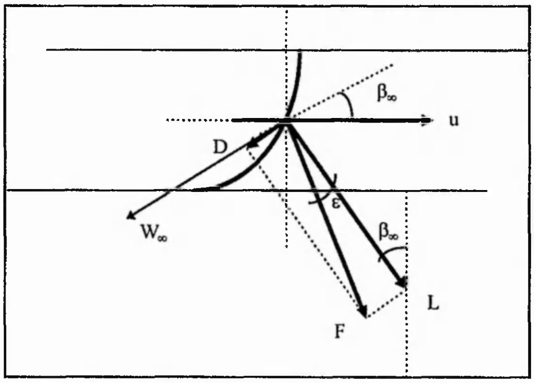

Figure 2 : Lift and drag force analysis of a blade in a cascade.

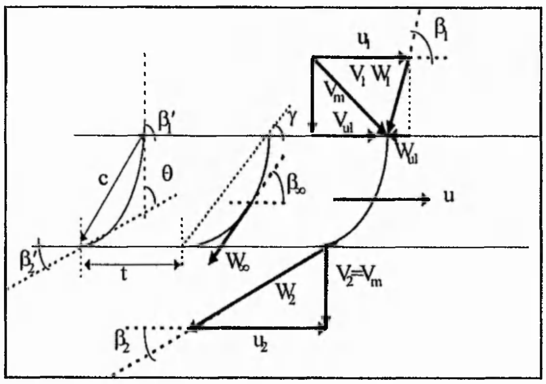

Figure 3 : Velocity diagrams at the inlet and outlet of the turbine cascade.

6. PROTOTYPE DEVELOPMENT.

A prototype propeller turbine is currently being constructed by the Department of Manufacturing Engineering of Nottingham Trent University. Emphasis is given to the construction process, materials and equipment used so that fabrication conditions resemble those commonly found in workshops in developing countries. The design characteristics of this prototype are shown in table 1. A speed of 1560 revs. min.$^{-1}$ was chosen as the turbine is intended to be directly coupled to a 4 pole, 50 Hz induction motor with rated speed 1445 revs. min.$^{-1}$. According to (8) and (10), the number of guide vanes selected for machines of up to 250 mm of tip diameter is 8.

Operating head : 2 m	Design speed : 1560 revs. min.$^{-1}$
Design flow rate :72 l/s	Tip diameter : 150 mm
Number of blades : 4	Hub diameter : 60 mm
Number of guide vanes : 8	Turbine overall efficiency : 72% (assumed)
Power Input : 1.4 kW	Power Output : 1 kW

Table 1 : Prototype turbine design characteristics.

The values of chord length and maximum camber are the ones that will dominate the development of the blade. Tables 2 and 3 show calculation results for the geometrical description of the above prototype propeller turbine for the hub and tip sections respectively.

t/c	0.800	0.900	1.000	1.200	1.300	1.400	1.500	1.600
c, (m)	0.059	0.052	0.047	0.039	0.036	0.034	0.031	0.029
C_L	0.812	0.913	1.015	1.218	1.319	1.420	1.522	1.623
μ	0.640	0.580	0.550	0.480	0.450	0.420	0.380	0.360
θ, ($\beta'_2-\beta'_1$), (deg.)	37.530	41.413	43.672	50.041	53.377	57.189	63.209	66.721
R	0.091	0.074	0.063	0.046	0.040	0.035	0.030	0.027
f	0.005	0.005	0.004	0.004	0.004	0.004	0.004	0.004
f/c	0.080	0.088	0.093	0.106	0.112	0.120	0.131	0.137

Table 2 : Geometrical definition of the propeller turbine for hub section.

t/c	0.800	0.900	1.000	1.200	1.300	1.400	1.500	1.600
c, (m)	0.147	0.131	0.117	0.098	0.090	0.084	0.078	0.073
C_L	0.153	0.172	0.191	0.229	0.249	0.268	0.287	0.306
μ	0.500	0.450	0.400	0.325	0.280	0.255	0.225	0.215
θ, ($\beta'_2-\beta'_1$), (deg.)	4.252	4.724	5.314	6.541	7.592	8.336	9.448	9.887
R	1.979	1.584	1.267	0.858	0.682	0.577	0.475	0.426
f	0.001	0.001	0.001	0.001	0.001	0.002	0.002	0.002
f/c	0.009	0.010	0.012	0.014	0.017	0.018	0.021	0.022

Table 3 : Geometrical definition of the propeller turbine for tip section.

Due to the fact that the blades will be welded to the runner hub, the designer must give the highest possible value of blade chord length at the hub in order to assist the welding processes and avoid excessive thermal stress concentration. The camber line and setting angle for hub, tip and two intermediate sections of the blades, are shown in figure 4.

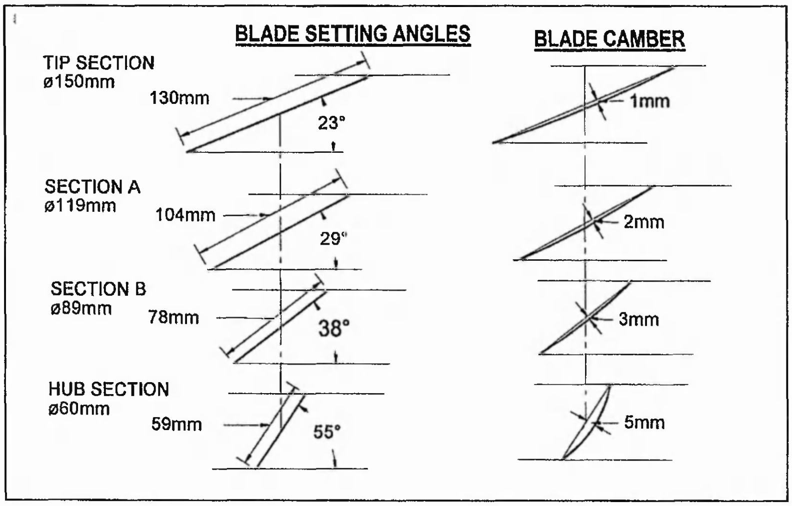

Figure 4 : Camber line and blade setting angles.

A careful selection of chord length will ensure the formation of a blade shape that can be easily cut out of metal plates, figure 5. Moreover, the amount of bending that the blade will have to be subjected to before being attached to the runner can be minimal if the values of maximum camber are kept low. Figure 6 gives a three - dimensional description of the blade within the cascade.

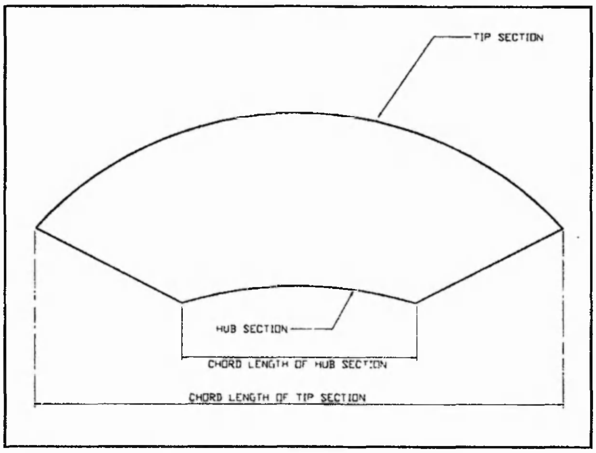

Figure 5 : Blade development before any bending or twisting.

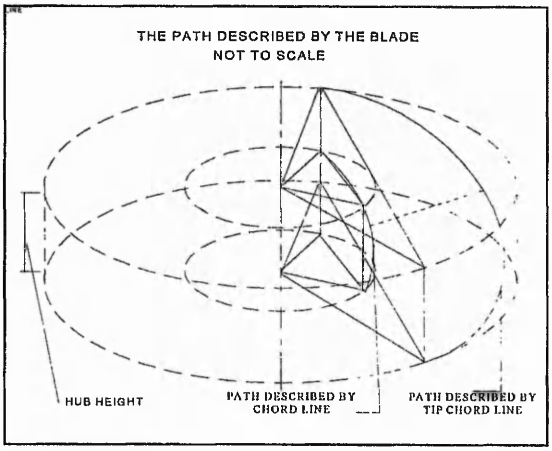

Figure 6 : Three - Dimensional description of the blade in the cascade.

7. FURTHER WORK.

A number of propeller turbine runners is currently being developed for 3 and 4 m of head. The tip diameter and speed of these runners are of the same value as the ones introduced in this paper. The reason for keeping the speed and overall dimension of the runner constant is for the standardization of the designs. All three prototype propeller turbines will be tested in order to derive the performance characteristics at design and off design operating conditions. The test results will then be used to develop an application chart, which will form the basis for the design of propeller turbines for any given site.

8. CONCLUSIONS.

A simplified method for the hydraulic design of propeller turbine runners has been presented. The important feature of this design methodology, is the use of simple geometry circular arc blades made of constant thickness sheets of metal, as opposed to conventional design approaches where aerofoil profiles are used. The design will utilize as much of the local skills and materials as possible while providing a reliable, robust and easily maintained system for the needs of remote communities. The range of flow rates and heads for which a particular runner can be used will be determined, and therefore, form the basis of a standard methodology for the design of propeller turbines. A more cost effective propeller turbine technology will be developed by paying attention to production costs. The involvement of local workshops in the manufacturing of these units serves a number of purposes. First of all, work is provided for the local communities, labour and transportation costs are kept to a minimum and maintenance requirements can be quickly attended to and therefore eliminate the risks of the schemes being shut down for prolonged periods.

9. NOMENCLATURE

A	surface area of the blade,	(m^2)
C_D	Drag coefficient,	
C_L	Lift coefficient,	
D	Diameter,	(m)
	Drag force,	(N)
F	Force between L and D,	(N)
f	Frequency,	(Hz)
H	Head,	(m)
k_{u1}	Coefficient of peripheral velocity at tip,	
L	Lift force,	(N)
N	Turbine rotational speed,	(s^{-1})
N_s	Dimensionless turbine specific speed,	
$N_{synch.}$	Synchronous speed,	(s^{-1})
N_m	Motor speed,	(s^{-1})
p	number of poles of motor,	
Q	Flow rate,	(m^3 / s)
R	Blade radius of curvature,	(m)
s	Slip,	
t	Blade spacing,	(m)
u	Peripheral velocity of blade,	(m / s)
V	Absolute velocity,	(m / s)
W	Relative velocity,	(m / s)

Greek letters

β	Fluid angle,	(°)
β'	Blade angle,	(°)
γ	Blade setting angle,	(°)
δ	Deviation angle, ($\beta'_2 - \beta_2$)	(°)
ε	Gliding angle,	(°)
θ	Blade camber angle,	(°)
i	Incidence angle, ($\beta_1 - \beta'_1$)	(°)
μ	$[\beta_1 - \beta_2] / [\beta'_1 - \beta'_2]$	
ρ	Density of fluid,	(kg/m^3)
σ	Solidity ratio,	(c / t)
ω	Rotational speed,	(s^{-1})

Subscripts

∞	Average direction of flow,
H	Hub section,
T	Tip section,
m	Meridional direction of flow,
u	Peripheral,
1	Inlet,
2	Outlet,

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A Simplified Propeller Turbine Design For Direct Coupling To An Induction Generator For Village Hydroelectric Schemes

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SYNOPSIS

The present paper introduces a propeller turbine design suitable for direct coupling to an induction generator. The use of such a unit, is a promising technology for setting up low-head power generation schemes for village electrification in developing countries. Emphasis is given to the hydraulic design of the runner blades which are made of constant thickness sheets of metal. The use of such a shape is ideal for low cost manufacturing in developing countries as it enables local skills and materials to be used. A prototype propeller turbine has been built and tested at Nottingham Trent University.

1. INTRODUCTION

Lack of electricity is a serious constraint to development in most of the rural communities in developing countries [1]. The extension of the national grid to remote communities in developing countries is prohibitively expensive. The potential sites for small scale, stand - alone, i.e. not connected to the grid, hydroelectric schemes, coincide quite closely with the locality of scattered rural villages and permits power to be generated near the end users at low transmission and distribution costs [2]. Such schemes with power output less than 100 kW are usually referred to as *micro hydro* schemes.

Pelton wheels are used for high head sites (above 30m), whereas cross flow turbines are most common for medium head sites (5 to 30m). Although there is a large number of people who live in the hilly and mountainous areas, population density is greater in low lying areas where rivers have large flow rates but with more gentle head drops (less than 5m). In a recent report to the Overseas Development Agency (ODA), it is suggested that it is in these areas that micro hydro has the greatest potential [3]. Potential low head sites are also found in countries with extensive irrigation canals with small drops, such as Egypt, India Indonesia and Pakistan. When a low head scheme is part of irrigation works, capital costs are generally lower because much of the civil works are already in place.

Propeller turbines are ideal for low head micro hydro electric schemes. Due to their relatively high specific speeds, propeller turbines can be compact and operate at relatively high speeds. Other hydraulic machines available for utilizing hydropower from low head sites, such as water wheels and cross flow turbines, tend to be bulky, run at very low speeds and therefore require speed increasing mechanisms, such as gearboxes and belt drive systems, in order to drive a generator.

A directly coupled turbine - generator unit, offers several advantages over belt drive systems and gearboxes, such as improved system efficiency, increased turbine and generator bearing lifetime, reduced costs, lower maintenance requirements and simplicity of installation. In addition, the cost and reliability can be further improved when induction motors run as generators are used instead of synchronous generators [4]. Induction machines are designed for continuous operation and require less maintenance than synchronous generators, they are robust and for powers less than 20 kW are less expensive and more easily available.

The successful development and use of low cost, locally made turbine units directly coupled to induction generators can be seen by the increasing number of Peltric (Pelton Turbine Induction Generator) micro - hydro systems [5]. Peltric units are now common in Nepal and Sri Lanka and are also being manufactured in Colombia, Peru and Indonesia. Many of these units are fitted with Induction Generator Controllers (IGCs) which keep the voltage and frequency within close limits for stand alone operation. IGCs are being manufactured by trained engineers in the same countries where the Peltric units are made. This reduces the cost of the controller and enables repairs to be carried out, when necessary, by local people.

2. TURBINE SPEED AND LAYOUT

The design speed of the turbine is determined by the type of the generator used and the operating speed which it was designed for. Induction motors are usually available in 2-pole, 4-pole and 6-pole configurations. When run as generators typical generating speeds for 50 Hz supply are 3100 r/min, 1550 r/min and 1030 r/min respectively. The prototype low head propeller turbine was designed to run at either 1030 r/min or 1550 r/min. The use of 2 pole machines is not suitable due to the high operating speeds for direct coupling.

The layout of the prototype unit is shown in Figure 1. Power is extracted upstream the overhung runner. The major advantage of this arrangement is the direct coupling of the propeller runner onto the generator shaft which requires a small extension that was successfully tried with Peltric units by Katmandu Metal Industries in Nepal. The generator's bearings are used to take the axial and radial runner loads. Less components, such as bearings and bearing housings are used, less workshop time is spent for mounting the generator-runner assembly and therefore the overall cost of the unit is reduced.

The radial guide vane assembly is bolted into the spiral intake casing which is made of sheet metal plates shown by Figure 2. While the turbine was fabricated, every effort was made to use basic equipment and tools that are available in workshops in developing countries. The use of a lathe was limited to the turning of the turbine hub and runner blade tip diameter. The unit is made in different sections that would allow ease of transportation and installation over rough terrain in rural areas. The assembly of the unit takes no more than two hours for one person using only an M10 spanner.

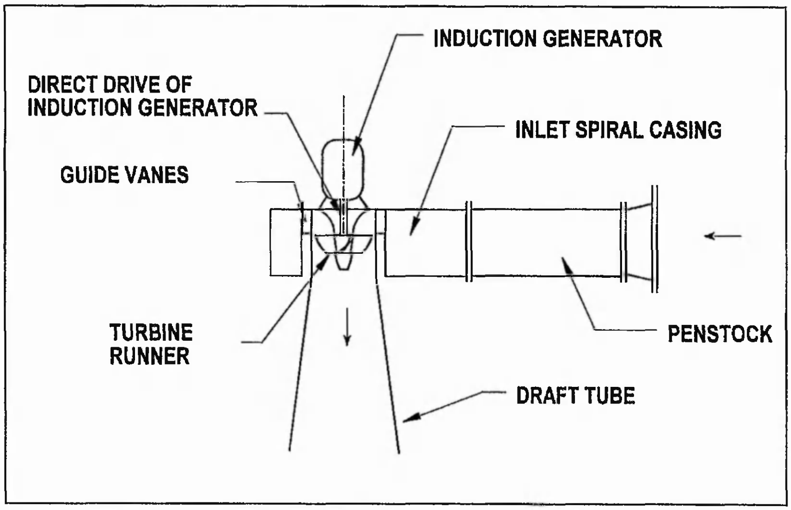


Figure 1 : Propeller Turbine / Induction Generator Unit Assembly

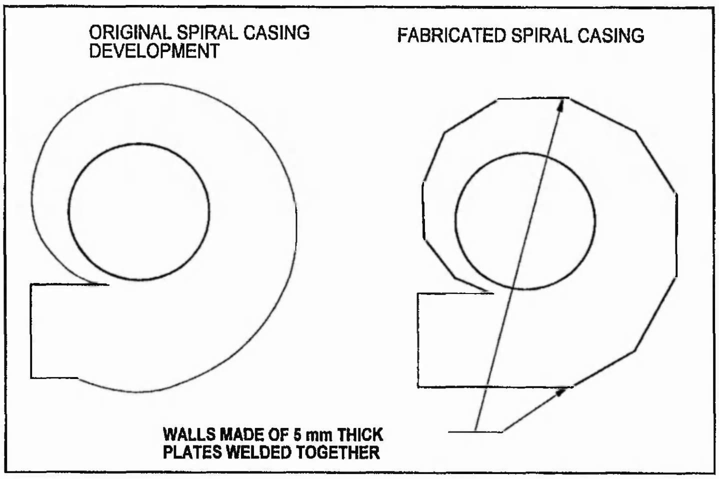


Figure 2 : Turbine Intake Casing

3. RUNNER DESIGN AND FABRICATION

Conventional axial flow turbine runners have aerofoil shape blades. However, due to the geometrical complexity of these shapes, specific milling operations and skills are required for their production that are not commonly available in small workshops in developing countries. And even if the skills and machinery are available, turbine runner blades made of aerofoil shaped profiles would cost a lot more compared with fabricated blades made of flat sheets of metal, bent and twisted to the desired angles. Experiments conducted on axial flow fans, which have Reynolds numbers similar to small water turbines, have shown that the same efficiency could be attained by blades made of constant thickness sheets of metal as for aerofoil sections [6], provided the geometry was preserved. Figure 3 shows the design methodology used for the hydraulic design of the first prototype propeller turbine. Details of this methodology is given by [7].

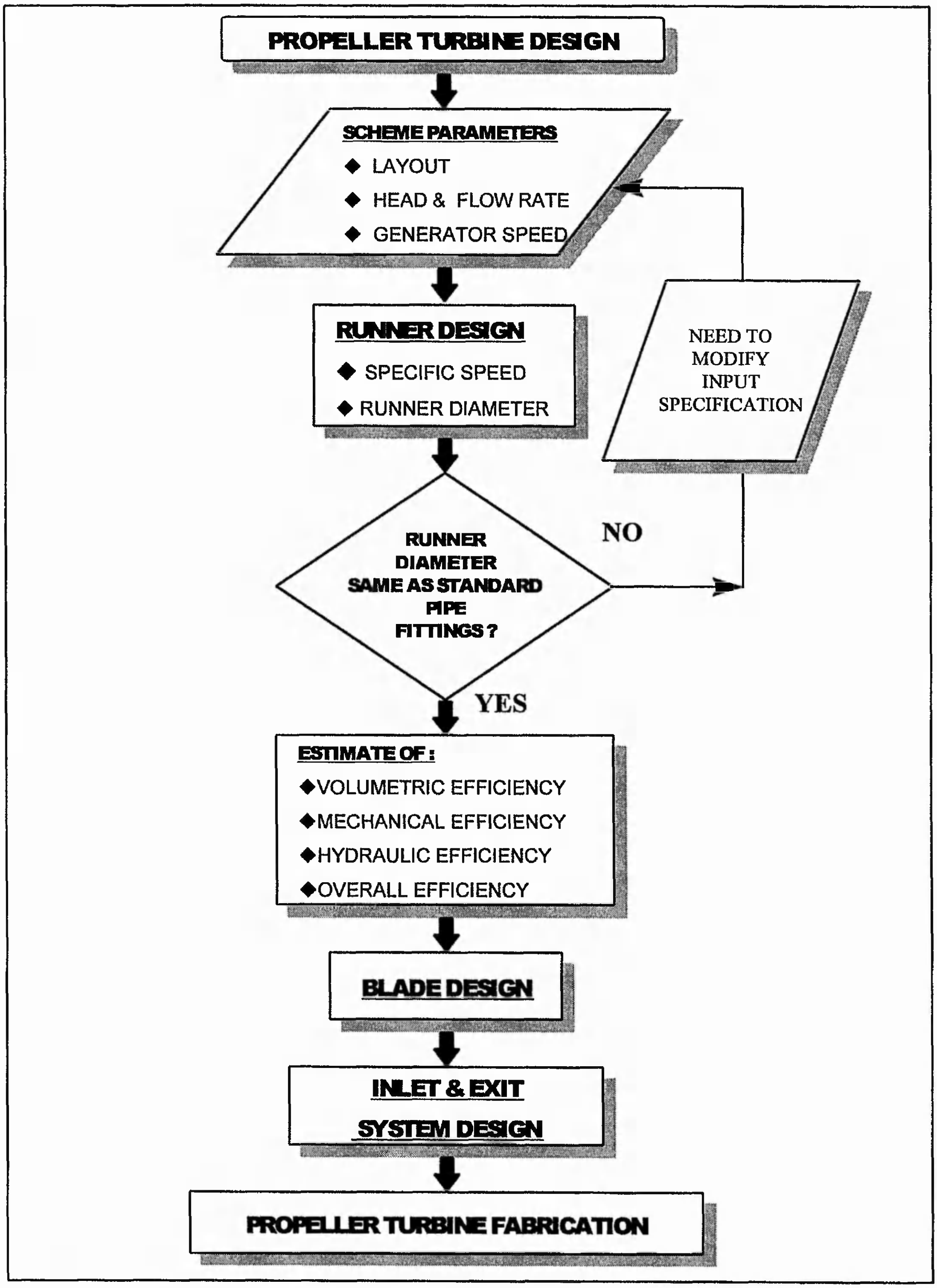


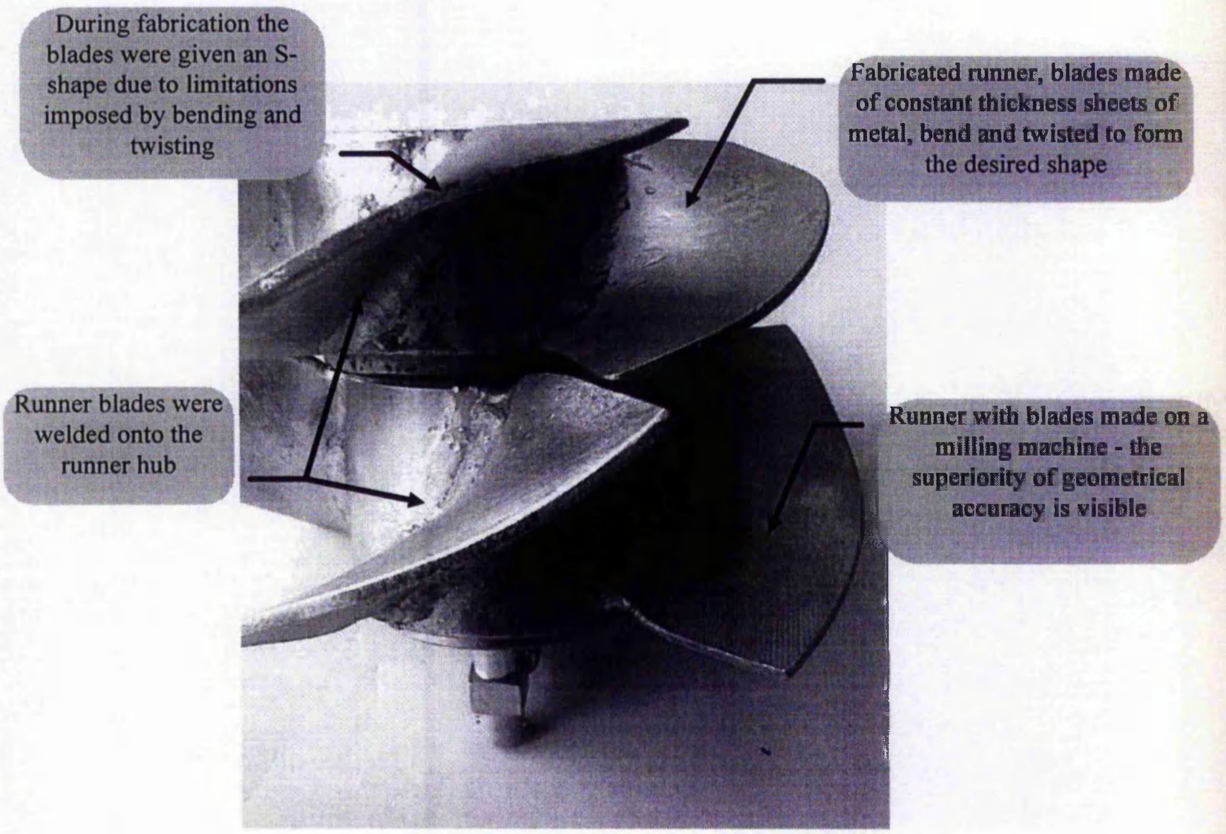
Figure 3 : Hydraulic design of the first prototype propeller turbine

The design method has allowed the use of dimensions that would allow standard pipes and fittings to be incorporated into the design and therefore reduce manufacturing and material costs. The present prototype casing central part can therefore be made from standard 150 mm (6 inch) diameter steel pipe. The design characteristics of the first prototype are shown in Table 1.

Number of runner blades : 4	Runner tip diameter : 149 mm
Number of guide vanes : 8	Runner hub diameter : 60 mm
Design Head : 2 m	Design Flow rate : 0.072 m^3/s

Table 1 : Design Characteristics of the prototype propeller turbine

The profile used for fabrication simplicity is the circular arc. Each 3mm constant thickness blade was welded onto the turbine hub and then given a gradual twist with the aid of a wrench at the tip section. Bending and twisting flat plates to form the desired blade shape has proved to be a challenge. As it can be seen from Picture 1 the blades of the runner have a slightly S-shaped profile and only the mid section of the blades was fabricated to the geometrical accuracy produced by the hydraulic design. A second runner has been manufactured, for comparison, with blades made by a milling machine and then attached onto the runner hub as shown by Picture 1.



Picture 1 : The prototype propeller turbine runners

4. EXPERIMENTAL WORK AND TEST RESULTS

The prototype design was adapted to meet the requirements for testing. For example, a vehicle alternator was used for loading the turbine in order to allow testing to be carried out over a wide range of operating speeds. The three phase output from the alternator was fed through an AC/DC converter into a 1kW resistive load bank (25V DC light bulbs were used). Moreover, a perspex tube was used in place of the steel pipe for the runner housing so that cavitation could be observed. The current test rig set up (Figure 4) complies with the requirements set by International Standards for the testing of hydraulic machinery [8]. Table 2 and Table 3 show selected operating characteristics for the fabricated runner and the runner with blades made on a milling machine respectively.

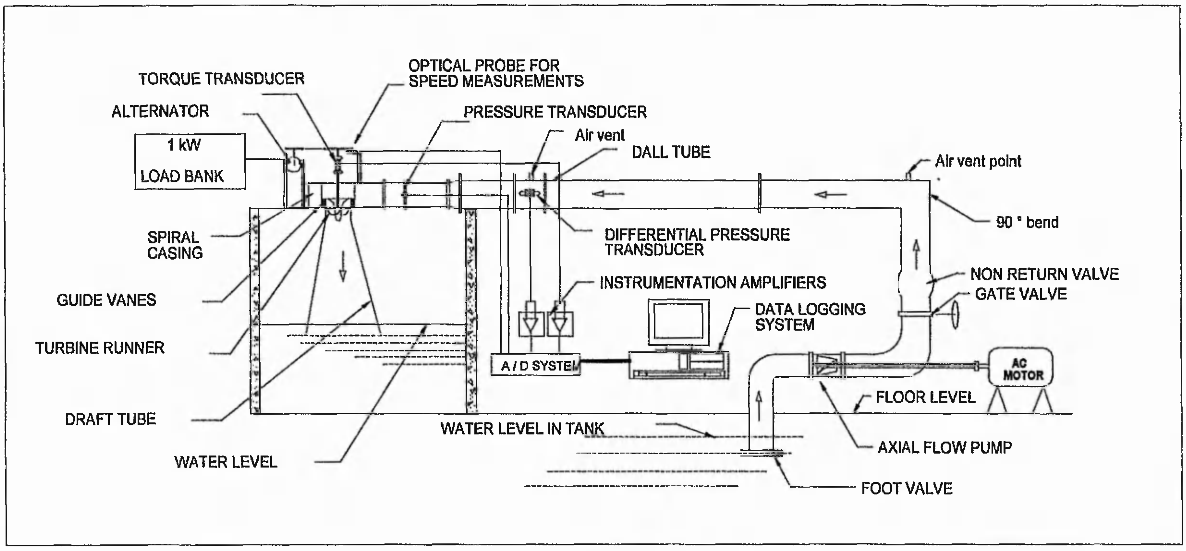


Figure 4 : Test rig layout

Operating Head	2 m	2.5 m
Flow rate	0.048 m^3/s	0.052 m^3/s
Best efficiency speed	1041 r/min	1050 r/min
Turbine best efficiency:	39 %	43 %
Power Input	0.94 kW	1.28 kW
Shaft Power Output	0.37 kW	0.55 kW

Table 2 : Operating characteristics of fabricated runner

Operating Head	2 m	2.6 m
Flow rate	0.051 m^3/s	0.058 m^3/s
Best efficiency speed	1054 r/min	1076 r/min
Turbine best efficiency:	48 %	54.7 %
Power Input	1.0 kW	1.48 kW
Shaft Power Output	0.48 kW	0.81 kW

Table 3 : Operating characteristics of fabricated runner

Figure 5 shows a comparison of the performance characteristics between the fabricated runner and the one with blades made on a milling machine.

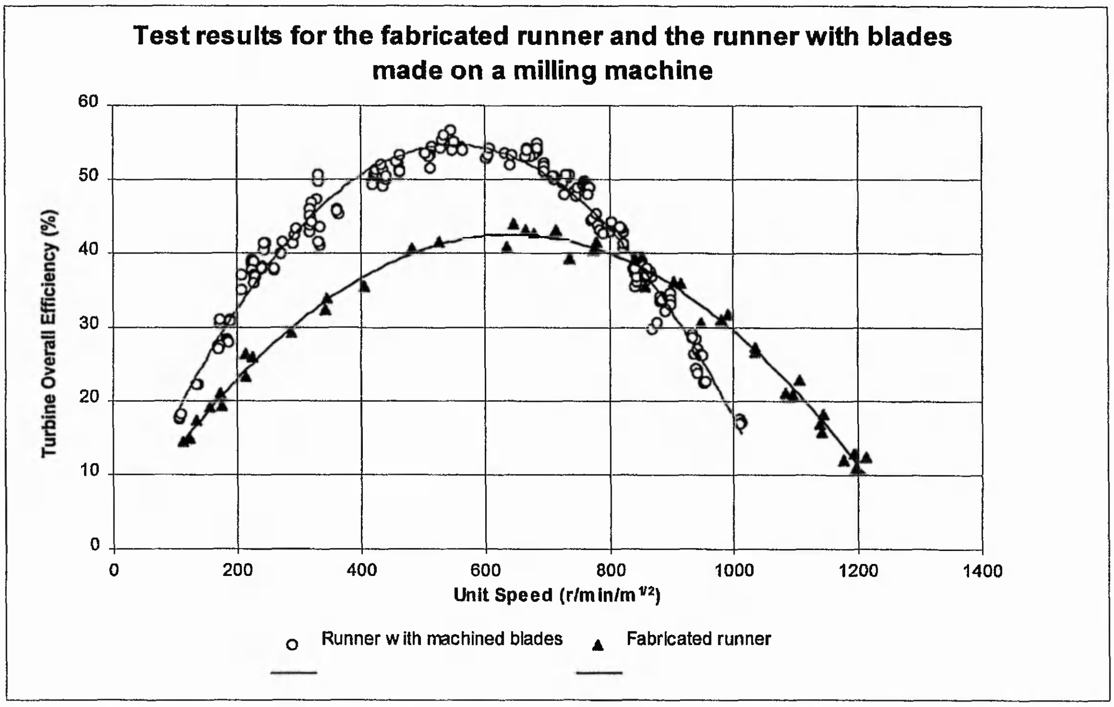


Figure 5 : Efficiency versus unit speed test results

5. CONCLUSIONS

The turbine overall efficiency of 54.7 %, although low in comparison with large axial flow turbines designs, is acceptable for village hydroelectric schemes. Similar low head propeller turbine designs have either poor performance [9] or run at much lower speeds [10].

A simple design of a low head propeller turbine has been proved. The important feature of this prototype is the use of simple geometry circular arc blades made of constant thickness sheets of metal, as opposed to conventional design approaches where aerofoil profiles are used. The design also utilizes as much of the local skills and materials as possible while providing a low cost, reliable, robust and easily maintained system for the needs of remote communities. The results presented by Figure 5 suggest that the turbine performance is sensitive to the geometrical accuracy of the runner blades. Moreover, it is obvious that the fabricated runner has a flatter performance characteristic curve and that the best efficiency point occurs at speeds slightly higher than that of the runner made with machined blades. Bending and twisting sheets of metal to form turbine blades has proved to be difficult but not impossible. From the above prototype propeller turbine a number of lessons can be drawn. In order to facilitate fabrication, the amount of twist in the runner blades has to be kept to a minimum. This can be achieved by designing the turbine for a lower specific speed and therefore using a slightly bigger diameter i.e. 200 mm (8 inch).

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