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VARIABLE PITCH FOIL VERTICAL AXIS TIDAL TURBINE

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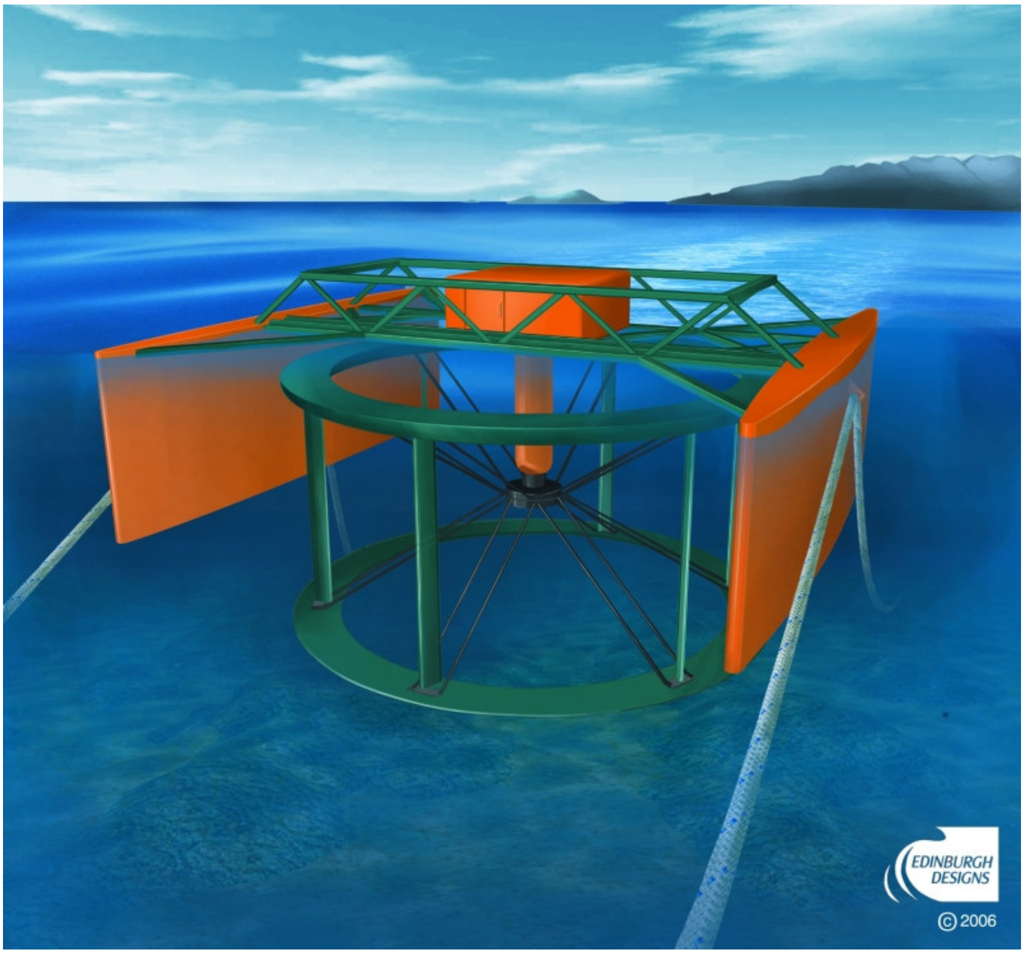
Project T/06/00234/00/00

Variable Pitch Foil Vertical Axis Tidal Turbine

Final Report



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EXECUTIVE SUMMARY

Background

There is continued interest in tidal energy generation as one of the components needed in the UK portfolio of renewable energy generation schemes. As the understanding and assessment of UK tidal energy resources progresses, it has become apparent that tidal regimes and site characteristics can vary considerably from one location to another. This, together with a survey of the state of the art in tidal energy technology, suggests that tidal energy converter schemes capable of economical operation over a wide range of site conditions would be inherently attractive. As a step towards this goal, Edinburgh Designs present in this report the results of an 11 month investigation into the technical and economical merits of a floating, variable pitch, Vertical Axis Tidal Turbine (VATT) scheme.

Technical Rationale

Tidal energy converter concepts based on conventional, fixed-pitch vertical axis tidal turbines have not shown great promise so far, owing to relatively poor efficiency, uneven loads and cavitation-limited operation. These issues can be resolved by introducing individual, active pitch control of each turbine foil. In addition, the floating vertical axis configuration offers significant advantages:

- Power train components and control system located above water, allowing for easy access and reducing maintenance costs and infrastructure.
- Rectangular energy capture cross-section, making the use of high rated powers permissible even in shallow water sites
- Versatility: the configuration is suitable for moored , pile or barrier installation

Objectives of the Study

The aim of this study, as set out in the proposal for a floating variable pitch VATT, is to investigate the following points:

Technical analysis:

- Impact of variable pitch on machine performance and energy output
- Sensitivity of expected gains in energy production to less-than-optimal design parameters
- Optimum pitch control strategy and optimum foil profiles for the task
- Structural loading and mooring configuration

Economic Analysis:

- Tidal resource investigation and definition of a baseline site for the proposed device
- Investigation of annual energy capture
- Build and operation costs for baseline site
- Optimisation of unit cost

Performance Modelling

Two models are developed to simulate the variable pitch VATT:

- A semi-analytical, multiple streamtube, double actuator disk model that is used to predict Cp power coefficients and hydrodynamic loads and torques on the rotor.
- A CFD model, using the CFX5 software, the aim of which is to validate the output of the semi-analytical model and in particular investigate the impact of local flow and wake turbulence.

Semi-Analytical Model

The output of the semi-analytical model confirms the importance of active variable pitch, which allows power coefficients in excess of 0.45 to be achieved. This efficiency figure is similar to that of a pitch-controlled horizontal axis design. Furthermore, the figures indicate that performance on a particular site may benefit from using a combination of pitch control and rotor speed control.

Variable pitch, together with the use of thicker foil sections, helps keeping cavitation on the blades to a minimum. As it may not be possible to eliminate it completely, some protection may be necessary at the top end of the blades but it is not expected to cause cost or reliability issues.

In terms of loads it is found, as expected, that a larger number of blades yields a smoother torque output. While this has little bearing on the design of small units, we anticipate that production-sized units (500kW), where loads are much larger, would use a 6-bladed rotor.

CFD Model

A CFD model of a dynamic 3-bladed, 45kW VATT with fixed or variable pitch has been implemented. Only the fixed pitch results could be incorporated in this report due to long simulation run times. Despite this, the results achieve an encouraging degree of agreement with the semi-analytical model output in terms of both magnitude and dynamics and justify the intensive use of the semi-analytical model for parametric studies.

Structural Analysis

This section investigates the natural dynamic characteristics of a floating vertical axis turbine structure and provides the basis for designing a simple but effective mooring system with optimum point of application and inclination of mooring forces. The steady-state analysis shows that the floating turbine structure is stable over a wide range of conditions. Heel angles up to 14 Degrees are tolerable without compromising stability. Locating the point of application of mooring force 0.5m below the deck gives the best stability against pitching. In further work, it is recommended that the effect of varying rotor torque on the structure be examined, as this induces yaw effects that cannot be represented by the current model.

Site Optimisation and Prediction of Energy Output

Tidal data for the Cullivoe Ness site, Shetland, are used as input to determine the achievable annual energy output on this site with variable pitch VATT technology. An energy model developed for this purpose also gives statistical summary of site specific tidal including the annual velocity distribution of the site and a tidal rose. An optimisation exercise is then carried out on a production size VATT with 500kW rated power.

It is found that on this site, a judicious combination of variable speed and active pitch control results in a 28% increase in annual energy capture over a fixed pitch VATT.

Two rotor configurations are examined: 3-bladed and 6-bladed. The 6-bladed configuration is preferable not only in terms of shaft torque fluctuation, but also since it results in lower blade loading and deflection.

Unit Cost of Energy

The unit cost of energy is determined for a production-sized 500kW VATT unit with fixed and variable pitch at two site locations:

- A baseline site with a 3m/s rated stream velocity, for which unit cost data were available for a horizontal axis tidal turbine scheme (MCT design, 1MW).
- The Cullivoe Ness site, for which detail tidal data are available

The results are summarized in the table below for a range of discount rates:

	Unit costs of energy (p/kWh)			
Discount rate	5%	10%	15%	8%
ED (500kW) Fixed Pitch, baseline site	9.1	11.9	15.2	10.7
ED (500kW) Variable Pitch, baseline site	7.2	9.4	12.1	8.5
ED (500kW) Variable Pitch, Cullivoe Ness site	10.3	13.6	17.3	12.2
MCT (1MW), Baseline Site	7.3	9.69	12.75	8.63

The economic benefit resulting from active pitch control is clear from the figures. While not competitive with existing energy sources at this point in time, the unit cost price for a 500kW variable pitch VATT unit is similar to that found for the MCT 1MW scheme. Generally, we find that for a given rated power and a given site, the pitch-controlled VATT may yield cheaper unit costs than the horizontal axis MCT design.

Conclusions

Overall, the objectives of this study, as listed above, have been met. In particular, we list the following conclusions:

1) Active pitch control on a vertical axis tidal turbine provides a net performance and economic benefit over the standard fixed pitch alternative. This is supported by parametric studies carried out with both a semi-analytical model and a CFD simulation.

2) Steady-state structural loads are well understood and a simple, robust mooring configuration has been studied for the device. This area, however, would benefit from further work to determine how transient loads affect the dynamic stability of the floating structure.

3) For a single production-sized unit, electricity unit cost comparable to that forecasted for a state-of-the-art horizontal axis tidal turbine

seem achievable with the variable pitch VATT concept (8-9p/kWhr on the baseline site).

4) Based on available data (2001), the results of the economic study suggest that for shallow sites (less than 25m depth, lower peak current velocities) the variable pitch VATT device may have an economic advantage over the MCT horizontal axis device.

A logical next step would be the development and testing of a small-scale prototype to explore practical design aspects, provide detailed input data for accurate costing of production units, and act as a technology demonstrator.

1 CHAPTER ONE – INTRODUCTION

1.1 Background

Tidal power will be an important ingredient of the UK's future renewable energy mix, and the available economic resource has been estimated in the range of 3-15% of total electricity demand (Craig 2001[1], Binnie and Veatch 2001[2], 2005). Assuming even the lower of these figures, the commercial prospects for tidal stream power in the UK are significant.

At current 'green' electricity prices the annual market for tidal electricity would be worth around £600m, and assuming an installed cost of £1000/kW, the market for UK tidal stream installations about £4bn. Estimates of the total worldwide resource are at an early stage but it is likely to be at least 100 times that of the UK.

Tidal energy is particularly valuable due to its predictable nature: the daily output may be forecast many years in advance, so that tidal generation can be scheduled into the UK supply system on both a day-to-day and long-term basis. Moreover due to the tidal variation at various sites, careful planning of farms across the country may provide a near baseload.

Significant R&D effort is still required, however. The first generations of horizontal-axis turbines are intended for water depths of 20-50m, but a significant resource lies outside this range (Black and Veatch, 2005). For deeper water moored structures may be required, while in shallower channels the horizontal-axis device does not make optimum use of the channel cross sectional area. The vertical axis configuration therefore presents a possible solution. A key design

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[1] *World energy Council Survey of Energy Resources 2001,* James Craig, AEA Technology (see www.worldenergy.org)

[2] DTI Report ETSU T/06/00209/REP, *The Commercial Prospects for Tidal Stream Power,* Binnie Black & Veatch/ IT Power, 2001.

choice is whether to use fixed or variable-pitch blades. A further choice for variable pitch blades is whether to use a passive or active control system. Variable pitch control can increase performance coefficients, and offer supplementary benefits such as the ability to feather all blades in the event of a system malfunction, but represents an increased technological risk.

1.2 Horizontal versus Vertical Axis Tidal Turbines

There has been a long-running debate on the relative merits of horizontal and vertical-axis wind turbines. A superficial analysis of the field would suggest that the horizontal-axis configuration has been the most successful. There is thus a strong and not unjustified reasoning that with this being the prevalent technology for wind energy, it may represent the best solution for tidal current energy. Whilst there are clearly many parallels between wind and tidal current energy, there are also notable differences, and because of these differences it can be argued that it is worth re-appraising the arguments of horizontal- versus vertical-axis turbines.

Notable differences include water being over 800 times as dense as air and the flow speeds of interest being an order of magnitude lower – while a wind turbine might have a cut-in wind speed of 5m/s and a rated wind speed of 13 m/s, a tidal turbine is likely to have cut-in and rated speeds of perhaps 0.7 and 2.5m/s respectively. These flow velocities applicable to wind turbines result in rotational speeds where the centrifugal forces are significant and promote a Troposkien blade shape on a vertical-axis turbine. With a lower flow velocity and a consequently lower rotational speed, the level of centrifugal stress (which scales with the square of the rotational speed, ω^2) is not likely to be a key design driver. Again, with wind, a ground-based vertical-axis turbine is undesirable owing to the atmospheric boundary layer, while with a floating vertical-axis tidal turbine this problem is removed; in this instance the swept area of the turbine would be almost completely in the region of maximum flow velocities.

Having suggested that the disadvantages of the vertical-axis turbine when applied to wind may not transfer to tidal currents, we might turn to the potential advantages. These are generally cited as there being no need to have a tower, and there being no need for a yaw mechanism. When applied to tidal currents the avoidance of a tower is likely to be a minimal advantage; a horizontal-axis turbine could be (one concept is) a floating device. With regards to the yaw mechanism, many horizontal-axis tidal current turbines rely upon the fact that many tidal current sites have flood and ebb tides that are approximately 180° apart. For sites where this is not the case, the vertical-axis concept offers a relative advantage.

Cavitation is an issue that affects tidal current turbine design, but does not affect wind, and so it should be considered whether the horizontal or vertical-axis turbine is better suited to avoiding cavitation. For a given foil geometry, cavitation will be a function of depth and velocity squared. The relevant depth will be the minimum water depth on the blade, which for a floating turbine might be near to zero. The velocity will be the maximum foil velocity, which is proportional to the rotor tip speed ratio. For cavitation performance, vertical-axis turbines have an advantage in that the maximum performance coefficient is reached at lower tip speed ratios. This is a consequence of the double disk intersection; i.e. a turbine blade intersects with the flow twice every revolution. Moreover, hydrodynamic loading is relatively even over the span of the foils compared to horizontal axis turbine where the outer tips of the blades do most of the work.

- The advantages of the variable pitch vertical axis turbine discussed above is as summarized below:
- Energy capture is insensitive to flow direction.
- There is more even hydrodynamic loading

- Lower rotational speed and lower swept depth range, hence reduced cavitation
- Efficiency can be made comparable to best horizontal axis designs
- Rectangular capture cross-section, hence can be used even in shallow waters

1.3 Design Issues

The above argument suggests that the vertical axis machine has enviable advantages; however it is not without challenges. The use of variable fixed pitch blades is a crucial design choice, which affects efficiency. Cavitation is also a problem as with all underwater lift surfaces. This section looks at the possible advantages of pitch control and its effect on cavitation.

1.3.1 Fixed versus Variable Pitch Control

The vertical axis tidal turbine comes from its wind counterpart, of which the fixed pitch Darrieus type is a common implementation. This fixed pitch turbine is easy to install and maintain, requires a simpler control system and can self-adapt to the inflow. However it has good performance only when the blade solidity is low and tip speed is very high. Furthermore it can hardly self-start. This is because in the starting phase, the blades experience a large range of attack angle (incidence angle), as they move both upstream and downstream during a single revolution, with the lift vector changing from one side of the section to the other. Consequently they will be subject to the high drag losses associated with stall, leading in turn to a reduction in power extraction efficiency and causing significant fatigue loading especially when tip speed is low.

The solution to these issues is to keep the rotor blades below stall at all times by controlling their pitch at all points of the rotation cycle. Using this technique, the efficiency of the vertical axis tidal turbine can be markedly increased. In addition, pitch control leads to a smoothing of the blade hydrodynamic loading during the rotor revolution, resulting in reduced variation in output torque; energy extraction is more evenly shared between the blade upstream and downstream strokes. Again this is beneficial in terms of increased fatigue life for the blades and other major components. Several attempts have been made at incorporating variable pitch features in vertical axis tidal turbines.

Salter (2005) suggested an algorithm for controlling the blade pitch angle to maximise the power output from the turbine. This is based upon achieving a two-thirds reduction in the momentum of the flow passing through the turbine, as specified by the Betz limit for optimal performance of wind turbines. A performance coefficient of 0.51 is reported. This remains steady through a range of flow speeds and tip speed ratios, provided that stall is avoided.

Another implementation that has actually been tested in a marine environment is The Kobold turbine[3]. Installed in the Strait of Messina, Italy it is such that blades are partially free to pitch under the action of hydrodynamic and inertial forces so as to reduce the angle of attack and hence the ability of the blade to stall. Allowed angular swinging of the blades is limited by the presence of two mechanical endstops. This control scheme is purely passive.

A comprehensive review of different passive control mechanisms proposed for wind turbines can be found in Pawsey (2002). A common feature of almost all the passive pitch control mechanisms reviewed was the provision of restoring moment acting independently on each blade. This moment may be produced by elastic means or by inertial loads on the blade, or a combination of both.

[3] Fiorentino, A, *Tidal Stream Plant at the Straits of Messina*, Ponte di Archimede SpA, Italy, 1998

1.3.2 Cavitation

Cavitation is a problem with all underwater lift surfaces. It occurs when the blade surface pressure falls below the vapour pressure of water, causing bubbles to form, and then quickly collapse; the problem is encountered near the water surface, where static pressure is lowest. Rotor performance can be degraded but a more significant effect is blade erosion, and potential fatigue damage. For each water depth and speed there is a safe cavitation-free envelope for a hydrofoil. It is therefore possible to alter the foil pitch so that it is in this safe region. In other words the blade pitch angle must be altered such that the blade remains below the critical cavitation pressure in all flow regimes.

Initial analysis of 4 symmetrical NACA 4-digit foils suggested that NACA0018 had the best characteristics for a vertical axis tidal current turbine. Its cavitation resistance is significantly better than NACA0015 and marginally worse than NACA0021. It has lower drag coefficients compared to NACA0021 though. However for structural reasons the NACA0021 or NACA0024 can be used with minimal detriment to performance.

1.4 Aims and Objectives of Project

The overall aim of this project is to assess the technical and economical viability of a vertical-axis tidal turbine concept capable of efficient operation in a wide range of current velocities to maximise energy production and minimise maintenance costs. The main objective is to establish the optimum configuration of the device for cost-effective power production. The specific objectives are:

- Creation of a design tool to access the performance of a vertical axis turbine.
- Comparison of energy capture gain provided by variable pitch control with the added cost of a pitch control system.

- Estimation of cost of energy produced by the tidal rotor device and comparison with energy cost of other devices.
- Conduct a design study to investigate the choice of material and construction techniques for a tidal rotor device.
- Comparison of the size, structure, loads and mechanical detail of the device with a generic horizontal axis machine and obtain comparative costs of transmitted power.

1.5 Structure of the Report

The report is structured into eight chapters. A brief description of what to expect in each chapter is as below.

- Chapter 1 gives a background of the study.
- Chapter 2 provides an overview of the available tidal resources as well as the devices being developed to exploit them.
- Chapter 3 describes the analytical model, which has been created to aid in the study of the device.
- Chapter 4 displays the CFD modelling work that has been undertaken to validate the analytical model and to provide incite into the flow pattern around the unit.
- Chapter 5 describes the structural analysis undertaken to provide a sound design capable of withstanding the hash marine environment.
- Chapter 6 is a design process to match a device to an environment.
- Chapter 7 provides the costing of the device and the economic potential of the energy produced compared to other devices.
- Chapter 8 provides the conclusion and recommendations of the study as well as future work.

2 CHAPTER TWO – REVIEW OF PREVIOUS WORK

2.1 Tidal Current Resource (UK and Worldwide)

Over the years several studies have been conducted to access the tidal energy resource in the UK but very few worldwide. Countries with an exceptionally high resource include the UK, Ireland, Italy, Philippines, Japan, China, Canada and parts of the United States. A summary of the results of the most recent study (Black and Veatch 2004) to access the global resource is as shown in Table 2.1.

Location	**Total Resource (TWh/y)**	**Technically Extractable Resource (TWh/y)**
UK	110	22
Non-UK European	85	17
Non-Europe Global	600	120

Table 2.1 Tidal current resource estimates

The study used a Flux method to estimate the total tidal resource as opposed to the Farm method used in previous studies which over estimates the resource. The technically extractable resource is obtained by devising a Significant Impact Factor (SIF) which is the percentage of the total resource that can be extracted without significant economic or environmental effect. It is worth noting that a single SIF value was used for all the sites. A SIF value for each site will give a more accurate estimate. In all 57 sites were considered for the UK. The site selection was limited to sites with depth greater than 20m and having maximum flow speed greater than 2m/s. The data used was based on data from previous studies, mostly ETSU 1993 and JOULE 1996. Table 2.2 and 2.3 gives the total and technically extractable resource distribution for the UK respectively.

Depth Range (m)	Total Mean Annualised Energy Distribution (GWh/y) (% in brackets)					
	Velocity Range (m/s)					Total
	2 - 2.5	2.5 - 3.5	3.5 - 4.5	4.5 - 5.5	>5.5	
20 - 25	139 (0.1)	2806 (2.6)	690 (0.6)	0 (0.0)	0 (0.0)	3635 (3.3)
25 - 30	82 (0.1)	1898 (1.7)	0 (0.0)	0 (0.0)	0 (0.0)	1981 (1.8)
30 - 40	865 (0.8)	6468 (5.9)	10338 (9.5)	0 (0.0)	0 (0.0)	17671 (16.2)
>40	2957 (2.7)	19262 (17.6)	12618 (11.6)	31615 (28.9)	19505 (17.9)	85957 (78.7)
Total	4023 (3.7)	30434 (27.9)	23647 (21.6)	31615 (28.9)	19505 (17.9)	109244 (100.0)

Table 2.2 UK total resource distribution (Source: Black and Veatch 2004)

Depth Range (m)	Technically Extractable Annualised Energy Distribution (GWh/y) (% in brackets)					
	Velocity Range (m/s)					Total
	2 - 2.5	2.5 - 3.5	3.5 - 4.5	4.5 - 5.5	>5.5	
20 - 25	26 (0.1)	559 (2.6)	138 (0.6)	0 (0.0)	0 (0.0)	723 (3.3)
25 - 30	16 (0.1)	380 (1.7)	0 (0.0)	0 (0.0)	0 (0.0)	396 (1.8)
30 - 40	173 (0.8)	1294 (5.9)	2068 (9.5)	0 (0.0)	0 (0.0)	3534 (16.2)
>40	558 (2.7)	3852 (17.6)	2524 (11.6)	6323 (28.9)	3901 (17.9)	17158 (78.7)
Total	774 (3.5)	6084 (27.9)	4729 (21.7)	6323 (29.0)	3901 (17.9)	21812 (100.0)

Table 2.3 UK technically extractable resource distribution (Source: Black and Veatch 2004)

The distribution shows that 80% of the UK resource is located in sites with depth greater than 40m. The best sites (velocity range 2.5 – 4.5, depth 30 – 40m) for economic exploitation of the resource make up only 15% of the total resource. This is only true for horizontal axis machines, which are the basis for these estimates. Vertical axis machines, which sweep out a rectangular section rather that a circular section, may offer a wider economic range for the resource.

Moreover, sites with high tidal velocities but depths less than 20m may offer good economic value for exploitation using vertical axis machines. Invariably vertical axis tidal turbines complement other technologies rather than competing outright with them.

It is worth noting these resource studies have used fairly crude methods to arrive at their results on the basis of sparse and barely adequate velocity data. The harsh marine environment, with strong currents, is partly to blame for this. Newly developed technologies such as the Acoustic Doppler Current Profiler (ADCP) will help providing more accurate data, leading to rapid improvement in the knowledge and understanding of this resource. Computer-based flow modelling also offers a valuable albeit relatively new tool for identifying promising sites and rapidly assessing them.

2.2 Technology Review

Several developers have sprung up to develop technologies to tap this resource. These technologies are at a very early stage of development and no clear front runner has yet emerged. The diversity and number of technical solutions implemented in these devices suggests that there is no single answer to how a tidal energy plant should be designed. The primary contenders are passive pitch vertical and active pitch horizontal axis rotor systems. There are other more exotic concepts that may have potential. Vertical axis tidal rotor systems may be particularly attractive at locations where limited maintenance facilities and resources are available (remote islands, etc) as it is not expected to require any special infrastructure. Since most developers have placed a high commercial value on the devices they are reluctant to divulge any information for the fear of competition. Below are some of them.

2.2.1 Marine Current Turbine Ltd (MCT)

MCT is pioneering the development of the horizontal axis tidal turbine. It has established a Consortium Group of industrial partners who between them have most of the capabilities necessary to complete the first phase of their R&D programme. The first "full-size" marine current turbine known as "Seaflow" was installed during May 2003 on the North Devon Coast, UK using a damp load instead of grid connection due to cost considerations.

Binnie Black and Veatch [4](2001) assessed the commercial prospects of the MCT technology. The baseline design used was rated at 1MW. It consists of two 500kW-power trains mounted at either end of a cross arm that is itself mounted on a mono-pile founded in the seabed. Each power train consists of a generator and gearbox driven by a rotor with a variable pitch mechanism. A lifting system is provided to raise the cross-arm, with the power trains, above sea level for maintenance and inspection. A jack-up barge is to be used to install the whole unit with the mono-pile grouted into a socket bored in the seabed. The barge will also be used to lay the cables connecting the units to the on-shore sub-station.

The unit cost of energy was determined for schemes with 1, 5 and 30 of these units. The resource assumed had a water depth of 30m and a peak current speed of 3m/s. The optimum rotor diameter for these conditions was evaluated as 15.9m (~198m^2 rotor swept area), mono-pile length of 59m and diameter of its lower section 4.22m. The results are as summarised as below:

Scheme size	**Unit costs of energy (p/kWh)**		
Discount rate	**5%**	**10%**	**15%**
1 units (1MW)	7.30	9.69	12.75
5 units (5MW)	4.56	6.12	8.05
30 units (30MW)	3.36	4.56	6.00

Table 2.4 Unit costs of MCT's 1MW baseline design schemes

[4] Binnie, Black & Veatch, "The Commercial Prospects for Tidal Stream Power", ETSU T/06/00209/REP, 2001

Verbal communication with MCT confirms these values are still valid and applicable in present conditions. According to MCT plans are far advanced to install a prototype of this size in the Strangford Narrows, Northern Ireland. This project is expected to cost £8m including grid connection. The DTI have offered a grant of £3.85m to this effect with the rest being financed by the operating partners.

2.2.2 TidEl (SMD Hydrovision)

The TidEl concept under development by *SMD Hydrovision*, a subsea equipment manufacturer, is a pair of contra-rotating turbines, mounted together on a single crossbeam. The complete assembly is buoyant and tethered to the seabed by a series of mooring chains. The mooring system allows the turbines to align themselves in the direction of the tide automatically, i.e. following the tide backwards and forwards as it changes direction. During slack water, the turbines float in a vertical position. This position allows for access for maintenance and repair works. The turbine is variable speed operated with fixed pitch blades.

A commercial size, 1MW (2 x 500kW), prototype with 15m diameter rotor is at the development stage for offshore testing to be carried out in 2006. It is likely to be tested in an offshore environment with peak tidal speed of 2.5 m/s or more and a water depth of 30 m. This will help to prove its viability and numerous perceived advantages. This phase in the technology development plan is being funded partly by a DTI grant of £2.7m. Meanwhile a 1:10 scale system, also partly funded by DTI, has undergone a seven-week trial program at the New and Renewable Energy Centre (NaREC) in Blyth.

In August 2004 SMD Hydrovision was awarded a grant by the DTI to prepare reports on the commercial and technical prospects for a 1MW TidEl system. PB Power was in charge of auditing the commercial prospects for the system while Newcastle Upon-Tyne and Robert Gordon Universities were to audit the technical aspects. The full report was not available to comment on during the preparation of this document.

2.2.3 Lunar System (Lunar Energy Limited)

The system being developed by *Lunar Energy* features a ducted turbine, fixed to the seabed via a gravity foundation. The blades are bi-directional rather than variable pitch, and there is no yaw mechanism as the Venturi effect caused by the ducting is said to help maximise the capture of energy from the water flow, even when the flow is not parallel to the turbine axis. This reduction in complexity should translate into improved reliability. This type of turbine is likely to be deployed in deep waters where it will not impact shipping traffic. A 1/20th model was tested in April 2004 in the Department of Naval Architecture and Marine Engineering test tank, University of Strathclyde.

A 1MW prototype, part funded by a £5.66m DTI grant, is expected soon, with commercial launch in 2006. This prototype will be installed at the European Marine Energy Centre where it will undergo operational and commercial development, testing and evaluation. Southampton University is to carry out experiments to test the duct and support structure.

2.3 Vertical Axis Tidal Turbine

The UK has been left out in the development of vertical axis tidal devices, despite the fact that their unique configuration makes them suitable for a category of sites, namely narrow and shallow channels, for which the UK is well endowed. In contrast to this, several American companies and European groups have invested in Vertical axis tidal turbines. Below is a review of two such devices currently undergoing development.

2.3.1 Kobold Turbine (Ponti Di Archimede SpA)

This straight-blade vertical axis tidal turbine makes use of purposely designed blade sections (HLIFT-18) to give high performance while avoiding cavitation. *Ponte di Archimede SpA* in association with the University of Naples is developing this concept. The whole system is mounted under a floating platform. The platform provides housing for the gearbox and generator units.

A prototype known as Enermar has been installed in the Strait of Messina, Italy where peak current speed is 2m/s and the depth is 20m. It is 3-bladed, has a diameter of 6m and a blade height and chord length of 5m and 0.4m respectively. This turbine has a passive blade pitch control system, which is made up of two balancing masses for each blade. In this way, it is possible to alter the blade's centre of gravity and hence pitch to enhance the rotor performance. The turbine has been reported to have high starting torque (Coiro et al 2005), eliminating the need for starting devices associated with its wind counterpart (e.g. Darrieus type). *Ponte di Archimede SpA* [5] has reported global efficiency of approximately 23% with the turbine generating a power of 20kW in a tidal current speed of 1.8m/s. This level of efficiency is comparable to that of wind turbines which have been under development for more than thirty years.

A pilot Kobold turbine farm is to be set up on the Zhoushan Archipelago in East China's Zhejiang Province later this year. The units of rating 70 - 250kW are expected to cost £325 000 each. Funding is partly by UNIDO and the World Bank.

2.3.2 Davis Hydro Turbine (Blue Energy)

Blue Energy has over two decades developed over two decades a straight-blade vertical axis turbine known as Davis Hydro Turbine. The turbine is mounted in a concrete marine caisson, which is then anchored to the seabed. The caisson also directs the water flow

[5] Ponte di Archimede SpA (www.pontediacrchimede.com.)

through the turbine and supports the coupler, gearbox, and generator above. Prototypes ranging from 4kW to 100kW have been tested mostly in rivers. For large-scale power production, multiple Davis Hydro Turbines are to be linked in series to form a tidal fence across an ocean passage or estuary.

Blue Energy is currently pursuing the development of a 500kW pre-commercial demonstration project off the coast of British Columbia, Canada. The project is comprised of two floating 250kW units. The unit is said to be viable in ocean currents of 1.75 m/s on the average (www.bluenergy.com). The Company has also proposed the development of a four-kilometer long tidal fence between the islands of Samar and Dalupiri in the San Bernardino Strait in the Philippines.

2.4 The Variable Pitch Foil Vertical Axis Turbine

The pitch control mechanisms for the vertical axis turbines under development are all passive. These methods are only effective over a range of flow conditions. A flexible approach will be to use an active pitch control system. That is what this technology seeks to implement.

2.4.1 Concept

The variable pitch foil tidal stream turbine is based on a straight-bladed vertical axis turbine concept. It is a lift device. The design is intended to optimise use of sea space and its rectangular swept area is particularly suited to shallower channels, avoiding both the seabed boundary layer and wave zones. It has rings at both ends of the blade to enhance flow into the turbine and to prevent tip losses. This device incorporates continuous blade pitch control, which brings significant advantages namely:

- The blades can be maintained below stall in all operating conditions
- High capture efficiency can be maintained, superior to a fixed-pitch VAWT

- Operation at relatively low blade speed to avoid cavitation

The variable pitch tidal stream rotor operates below stall at all times, using variable pitch control to limit the angle of attack of the blade. The result is a significant increase in rotor power coefficient, $C_{p,}$ compared to fixed pitch operation, especially at low tip speeds, which would translate directly into increased energy capture.

2.4.2 Turbine Configuration

The turbine comprises a number of evenly spaced blades oriented parallel to the axis of rotation. Each blade, with a constant foil cross-section, is supported and pivoted on a strong central strut that is rigidly mounted at both ends into the upper and lower rings. Tension spokes then connect the rings to the hub. These spokes transmit torque from the rings to the hub and can therefore be thin and highly stressed. A bicycle wheel with a similar structure has an unbeaten strength to weight ratio and the London Eye is an example of how the technology can be scaled up.

The floating structure provides housing for the power take-off mechanism. This allows for easy access and maintenance. The floating structure also provides flexibility over where it could be moored.

2.4.3 Potential Advantages of the Variable Pitch Foil Vertical Axis Technology

- Accessibility for inspection and maintenance

  One significant advantage of the vertical axis turbine configuration is that the power train equipment and control is above the surface of the water, allowing easy access for inspection and routine maintenance.

  A floating turbine has a low freeboard and is moored in a tidal stream. Technicians can visit the device with a small boat, much in the same way as fish farmers regularly inspect fish cages without requiring specialist vessels. The turbine unit will be deployed with a light tug rather than a heavy lift crane barge and can be towed back to port for maintenance.

In contrast, the fully submerged power generation equipment required for a horizontal axis tidal turbine is more complex and requires an expensive heavy lift crane barge for access and maintenance.

- Suitability to sites that are uneconomical for other technologies

  With vertical axis tidal rotors the aspect ratio of the cross-sectional flow area can be chosen to fit the water depth and tidal height variation. In other words, with floating vertical axis tidal turbine technology, it is possible to install large capacities in shallower water sites with large variations in tidal height. This is not possible with, for instance, horizontal axis tidal turbines.

- Flexible configuration

  While the rotor aspect ratio can be optimised for shallow or deep water sites, additionally vertical axis tidal turbines can either be moored or pile-mounted depending on local topography. Moreover it can be integrated into a civil engineering structure (e.g. Churchill Barriers).

- Complements other technologies

  This technology has the potential for increasing the economically exploitable fraction of the total resource. Consequently it complements other technologies rather than competing outright with them.

3 CHAPTER THREE – ANALYTICAL MODEL

3.1 Overview

A fast running but detailed analytical model based on the double disk multiple stream tube approach has been programmed in MatLab. The model makes use of standard airfoil characteristics to generate the load and performance characteristics of different sizes of turbines with respect to stream speed and rotational speed. The choice of this model, theoretical basis, structure, validation and limitations are as discussed as follows.

3.2 Model Requirements

The key requirements of the model for the variable pitch foil vertical axis tidal turbine are:

- Allow calculation of power output in a representative environment
- Calculate forces acting on key system elements, for the operating range of the unit, in support of system design and cost analysis
- Enable a range of suitable environment to be represented
- Allow dynamic behaviour of the generating system to be understood
- Allow different design parameters to be evaluated, specifically:
    - ✓ Changes in number of blades
    - ✓ Changes in turbine diameter
    - ✓ Performance of different blade profiles
    - ✓ Alterations to pitch angles with rotational speed

3.3 Theory

The way vertical axis tidal turbines extracts energy from the water is described in this study by the actuator disk principle. The approach is similar to that used to model horizontal axis turbines, with the difference that, here the machine sweeps out a cylinder, instead of a disk, and so intersects any given stream tube twice as in Figure 1. Effectively there are two actuator disks in tandem, each set at an angle to the flow and each extracting energy from the flow. Various theories have been put forward to deal with the presence of the two disks. The simplest theories lump all the two disks together and assume that all the energy is extracted at the mid-vertical plane of the cylinder. Such an approach can be treated in the single stream tube (Templin, 1970) manner in which the entire rotor is enclosed in one stream tube or the multiple stream tube (Strickland, 1980) manner where the swept volume of the rotor is divided into a series of adjacent stream tubes. While these models predict reasonable turbine global performance, they do not take into account local blade Reynolds's number and dynamic stall effects. These models are therefore inadequate to describe such flow details as those related to stream tube distortions and interactions.

A state of the art multiple stream tube analysis, which takes into account the double intersection and flow expansion through the rotor, as in Freris (1990) has therefore been considered for this study. It makes use of the conservation of momentum principle in a quasi-steady flow and equates forces on the rotor blades to changes in streamwise momentum through the turbine. With reference to Figure 1 and for a free stream velocity of U_∞ the rate of change of momentum, *dFu* and *dFd* for the upstream and downstream disk of an elementary stream tube are given by:

$$dF_u = 2\rho U_\infty^2 a_u (1 - a_u) dA_u \qquad (1)$$

$$dF_d = 2\rho U_a^2 a_d (1 - a_d) dA_d \qquad (2)$$

Ua is the upstream velocity of the downstream disk while a_u and a_d are the upstream and downstream induction factors. Equating these forces to the time-averaged force on the blade as it crosses each stream tube then the problem is reduced to the calculation of the induction factors. However because of the non-linearity of the resulting equation an iterative procedure is employed.

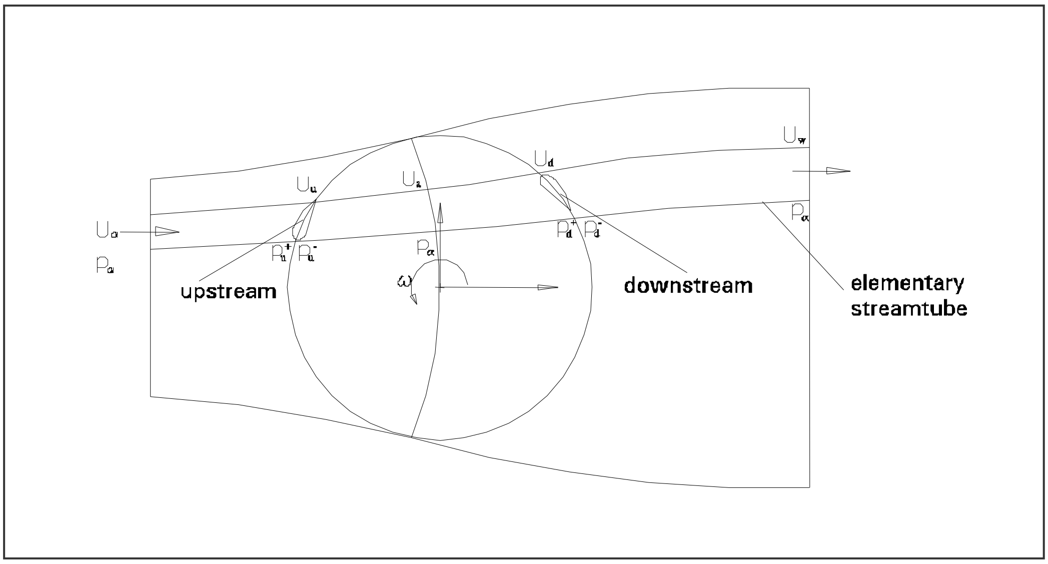

Figure 3.1 Plane view of actuator cylinder

It is worth noting that the upstream disk area is less (and the downstream disk area greater) than half the blade swept area as in Figure 1. Invariably a large proportion of the upstream velocity is lost and the velocity entering the downstream part of the rotor is distorted. This model provides much detail about hydrodynamic response, such as the variation of torque and blade normal loading relevant for performance prediction and design. However it assumes inviscid flow, therefore does not take into account turbulence.

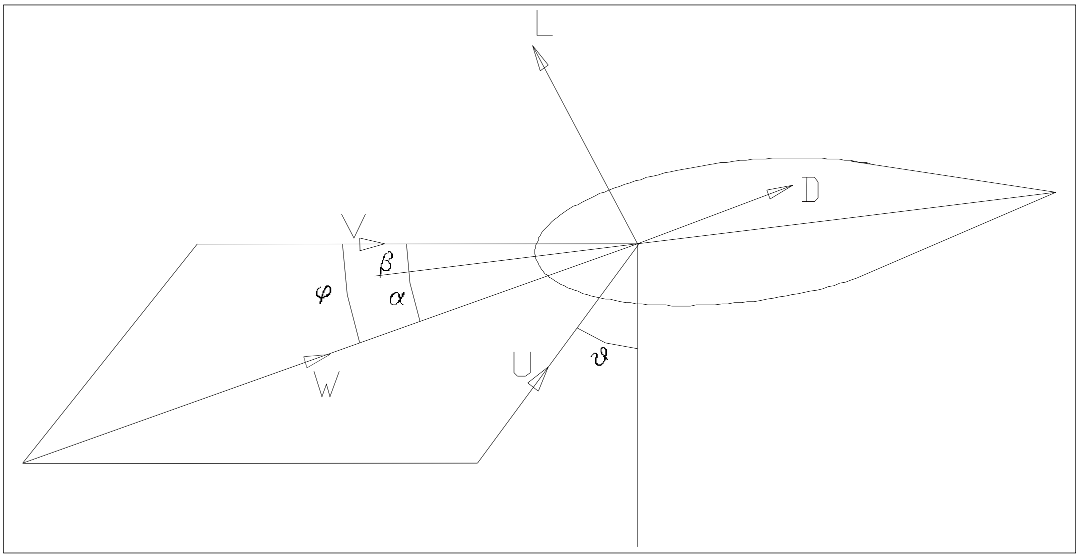

Figure3.2 Velocity vectors and forces on a blade section

3.3.1 Blade Pitch Control

As a blade progresses through a revolution the relative flow angle, φ (attack angle plus pitch angle) will be constantly changing, even under a steady flow speed. If the angle of attack exceeds the stall angle there will be a considerable decrease in lift and an increase in drag, both of which lead to reduced (potentially negative) torque and thus output power. Stall can be avoided by feathering the blade, this means altering the blade pitch angle, β, for a given relative flow angle, φ, to reduce the angle of attack, α (see Figure 3.2). Secondly, at other points during the revolution of a blade pitching the blade towards stall can increase the torque.

Variable-pitch in the context of the vertical-axis turbines implemented in the model refers to cyclic pitch variations on individual blades. Attack angle, which determines the lift/drag coefficients of the foil, depends on other factors (tip speed ratio, solidity) than blade orientation. This makes it difficult to schedule a pitch variation to prevent stall and maximise performance. Therefore the pitch control adapted into the model is to limit the foil attack angle to a set value to prevent stall and avoid cavitation while increasing efficiency. This means the set value known as “limiting attack angle” is input while the pitch angle needed to achieve this becomes an output. Setting this limiting attack angle very high (say 90 degrees) results in fixed pitch operation.

3.3.2 Cavitation

This forms another block of the performance prediction model. The inception and intensity of cavitation is dictated by:

- Local stream velocity
- Rotor rotational speed
- Depth of deployment of rotor
- Pressure distribution across the foil

A cavitation number σ is defined as:

$$\sigma = \frac{P_{atm} + \rho g h - P_v}{0.5 \rho V^2}$$

and the pressure coefficient as:

$$C_p = \frac{P_L - P_0}{0.5 \rho V^2}$$

where P_L is the blade local pressure and V the stream velocity relative to the blade. Cavitation inception can therefore be predicted from the pressure distribution across the blade profile. A good indicator of cavitation, applied in the model, is therefore a ratio of the blade local pressure to the vapour pressure of seawater, P_v. This means that cavitation sets in once this ratio falls below unity. It must be noted that the vapour pressure of water is site specific (for seawater: P_v = 2.4kPa at 20 ^{0}C and 4.3kPa at 30^{0}C).

3.3.3 Pitching Moments

Experiments have shown that the centre of pressure changes with operating conditions. This means for every flow conditions the foil can be pivoted at a point to give a zero pitching moment.

With reference to Figure 3.3 if the foil is pivoted in front of the quarter chord the lift force will generate an anticlockwise turning moment.

Conversely if the foil is pivoted behind the quarter chord the lift force will generate a clockwise turning moment. These moments may be desirable in certain positions in a revolution thereby requiring less effort to change the blade pitch. The quarter chord is therefore the reference pivot point and any pivot point outwit this position generates an extra moment, which is taken into account. This block in the model will aid in investigating the optimum pivot point of a blade considering a site's velocity distribution.

3.4 Model Structure

The model was created in MatLab. It makes use of the characteristics of an airfoil profile (lift, drag, pressure coefficient etc) to calculate torque, power output and main forces on various sizes of vertical axis tidal turbines (fixed and variable pitch) as a function of stream velocity and rotor rotational speed. These airfoil characteristics are evaluated at the local blade Reynolds Number and local attack angle. The model can be divided into two main sections as described as follows:

Velocity Diagram: This section deals with the evaluation of the velocity vectors of a blade element as a function of position and orientation. It also generates the pitch angle schedule for a prescribed attack angle as well as indicates if a blade or part(s) of it is in the cavitation region.

Loads and Performance: Armed with the velocity vectors and machine size, the loads (thrust force, normal force, blade-pitching moment) on the machine are then evaluated. This is followed by evaluation of performance (power output, torque, and rotor power coefficient).

Input: Basic turbine size (rotor diameter, blade number, height and chord, blade pivot point, blade profile characteristics) and operating conditions (stream speed, rotational speed, depth of deployment, attack angle schedule).

Output: velocity vectors, pitch angle, blade local pressure, loads, and performance characteristics

3.5 Numerical Results

The numerical results are grouped into two main areas. The first is the examination of the effect of variable pitch control on performance as opposed to fixed pitch. Different pitch control regimes were applied and their effect on performance investigated. Control options to maximise output while avoiding cavitation and minimising pitching moment were then explored. The implications of these control options on torque variation as well as load on blades makes up the second part. The dimensions of the turbine used for the analysis are as in Table 3.1. It has been estimated that if such a machine is rated at a stream speed of 2.3m/s it will have a rated power of 45kW. These dimensions have been carefully selected to reflect the proposed 45 kW prototype in the technology development plan.

Parameter	**Value**
Diameter (m)	5
Chord (m)	0.375
No. of blades	3
Height of blade (m)	4
Foil profile	NACA0018[6]
Stream speed (m/s)	2.2

Table 3.1 Basic Dimensions of Turbine

3.5.1 Performance Characteristics

Figures 3.3 shows the variation of attack angle for a fixed and variable pitch machine at a TSR (tip speed ratio) of 2.25 over a revolution. The pitch angle variation required to achieve a limiting attack angle of 12 degrees is also shown. A positive pitch angle means foil nose down

[6] Foil Characteristics from: Paraschivoiu, I. (2002), *Wind Turbine Design (With emphasis on Darrieus concept)*

while a negative is nose up as shown for the downstream and upstream sections respectively. Figure 3.4 shows how well the turbine performs with respect to different operating conditions (flow speed and rotational speed) for fixed pitch and variable pitch (different limiting attack angles).

Comments

- A greater degree of control (i.e. larger pitch angles) is required when the blade is in the upstream region as supposed to the downstream.
- A significant increase in performance (approx. 30 %) is achieved for the variable pitch machine over the fixed pitch one and is comparable to state of the art horizontal axis turbines
- For different limiting attack angles an optimum operating point is achieved at different operating conditions. This has two main implications:
    - ✓ For a fixed speed machine, any change in stream speed requires a different pitch control profile to maximise output.
    - ✓ For a variable speed machine, any change in stream speed requires a change in rotor rotational speed, maintaining the same pitch control profile, to maximise output.

The limit of the extent of this control will be dictated by the inception of cavitation, which will be investigated in the next section.

3.5.2 Cavitation

This section looks at the extent to which the machine can be controlled to maximise output while avoiding cavitation. Figure 3.5 and 3.6 show the evolution of the local pressure ratio at the top part of the blade at stream speeds of 2.2 and 3m/s respectively. The evolution of the local pressure ratio for these speeds is evaluated at the optimum condition for each limiting attack angle setting as in Figure 3.4. The machine is operating free from cavitation at the lower stream speed over the range of limiting attack angle while operating partly cavitating for the higher stream speed. This is due to the fact that the blade local pressure is proportional to the square of the local stream speed.

Comments

- With regards to fixed speed operation, cavitation puts a limit on pitch control (limiting attack angle) to maximise output. For variable speed operation cavitation puts a limit on the rotor rotational speed needed to maximise output as the stream speed increases. A combination of pitch and rotational speed control may therefore be necessary to maximise output as well as avoid cavitation.

- It is obvious that increasing the depth of deployment of the rotor will minimise and possibly eliminate cavitation, however the implications for the control system need to be addressed.

- Thicker foils are less likely to cavitate. Very thick foils may not perform better, due to their very large nose leading to high suction coefficients at low attack angles.

- Generally the foils cavitate over a short period and mostly at the top part. It may be necessary to condition this part of the blade to withstand cavitation to some extent. This may need further investigation (materials).

3.5.3 Blade Pitching Moment

In this section the variation of blade pitching moment at different pivot points for different limiting attack angles have been investigated. Figure 3.7 shows the variation of pitching moment for fixed and variable pitch operation pivoted at the quarter chord. In this case the maximum pitching moment for variable pitch operation is 530Nm. Figure 3.8 therefore explores the variation of the pitching moment when the blade is pivoted before and behind the quarter chord. From this figure two pitch control strategies come to mind:

- **Low Limiting Attack Angle Operation**:

  In this case a low limiting attack angle is chosen and a pivot point found to minimise pitching moment. There is a small reduction in efficiency (2% for this case, 6deg) since lift is not maximised. However this system generates very low pitching moments requiring a smaller motor for blade control resulting in a reduction in costs.

- **Range of Limiting Attack Angle Operation:**

  A blade pivot point can be found for a range of limiting attack angle to minimise the pitching moment. Even though this system requires a relatively larger motor it is more flexible. Cavitation can be avoided while maximising output by switching between different limiting attack angles with respect to flow conditions.

3.5.4 Rotor Torque

Torque variation plays an important role in determining the complexity of the drive train. This section looks at the effect of different limiting attack angle on torque variation. The torque variation as a result of increasing the number of blades has also been investigated. In this case the number of blades was increased while decreasing the blade chord to maintain the same solidity and performance curve.

Comments

- Figure 4 showed approximately equal maximum efficiency for a range of limiting attack angles to prevent stall. However a higher limiting attack angle in the range gives in a slightly higher peak torque as a result of maximising lift.

- For the same pitch control regime a higher number of blades give a smoother torque requiring a relatively simpler drive train. In this case the range is 26 – 28kNm (27kNm ± 4%) for six blades and 23 – 33kNm (28kNm ± 18%) for three blades for a limiting attack angle of 12 degrees.

- It may be possible to schedule a pitch variation scheme to give an approximate constant torque output. The resulting simpler drive train translating into lower costs must be justified by the value of the energy lost.

- For smaller machines, the configuration can be 3- or 6-bladed without significant design implications. For larger machines the gearbox design will be simpler and hence cheaper for a 6-bladed version, so it is expected that larger designs use 6 blades.

3.5.5 Blade Normal Loads

The choice of material for a blade will depend on the loads and stress it will be subjected to. This also has implications on cost. For a given blade section the normal force will also put a limit on the height of the blade. This section looks at the typical normal hydrodynamic load variation on the blade for different limiting attack angles.

Comments

- A change in limiting attack angle does not result in any appreciable difference in the normal loads on the blade for the range of attack angles under consideration. This is because the blades remain unstalled at all times. It is more sensitive to the stream speed and the rotor rotational speed.

- The blades experience greater loading during the upstream turn where the rotation of the rotor opposes the stream flow.

3.6 FIGURES (ANALYTICAL MODEL)

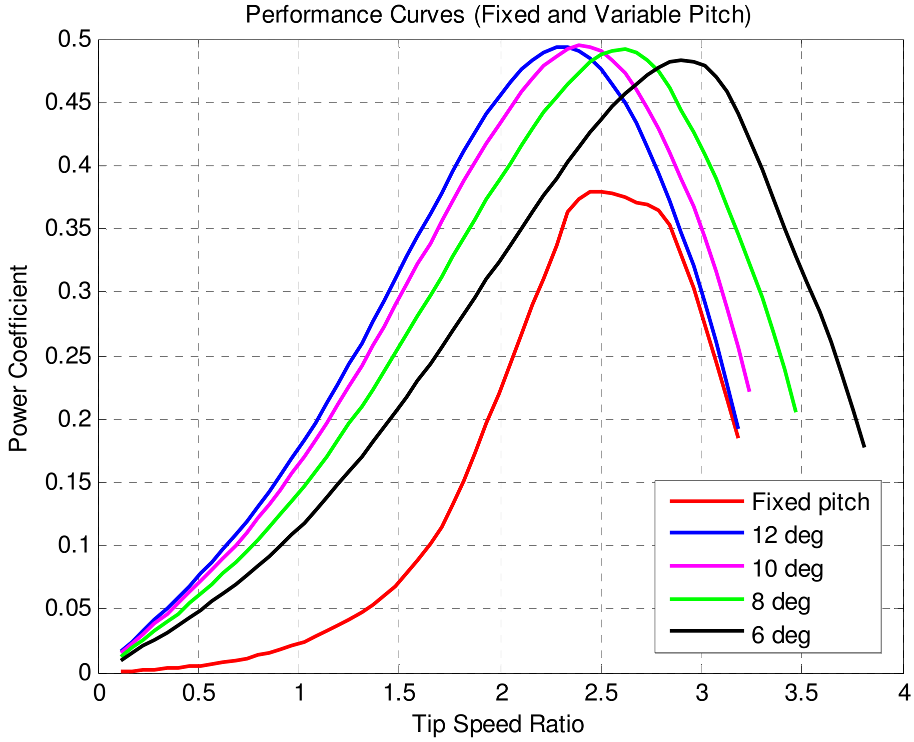

Figure 3.3 Evolution of attack angle for fixed and variable pitch (limiting attack angle is 12 degrees)

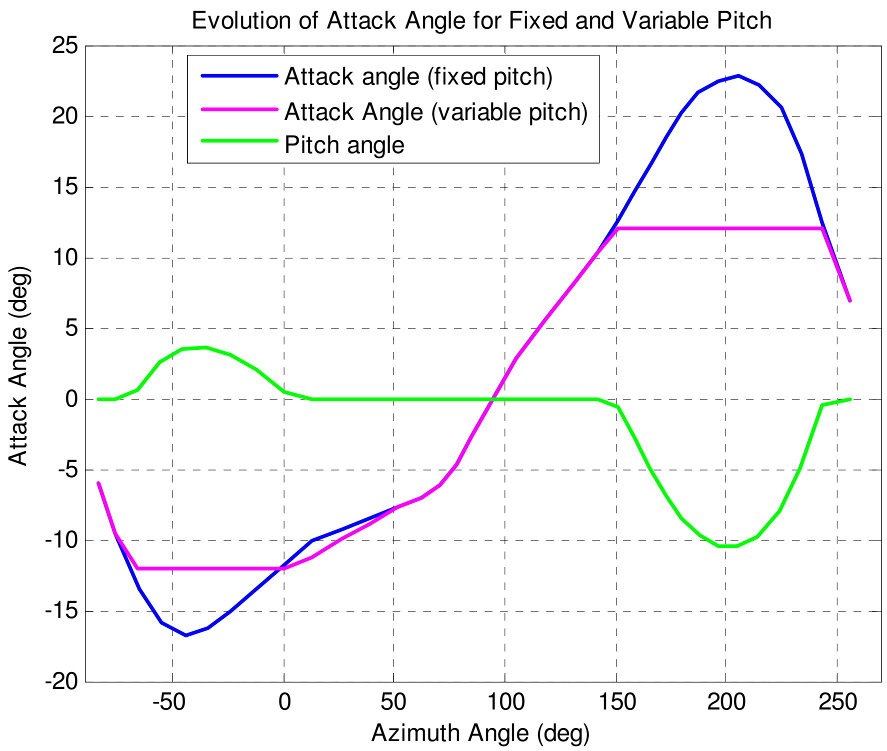

Figure 3.4 Performance curve at fixed pitch and variable pitch (different limiting attack angles)

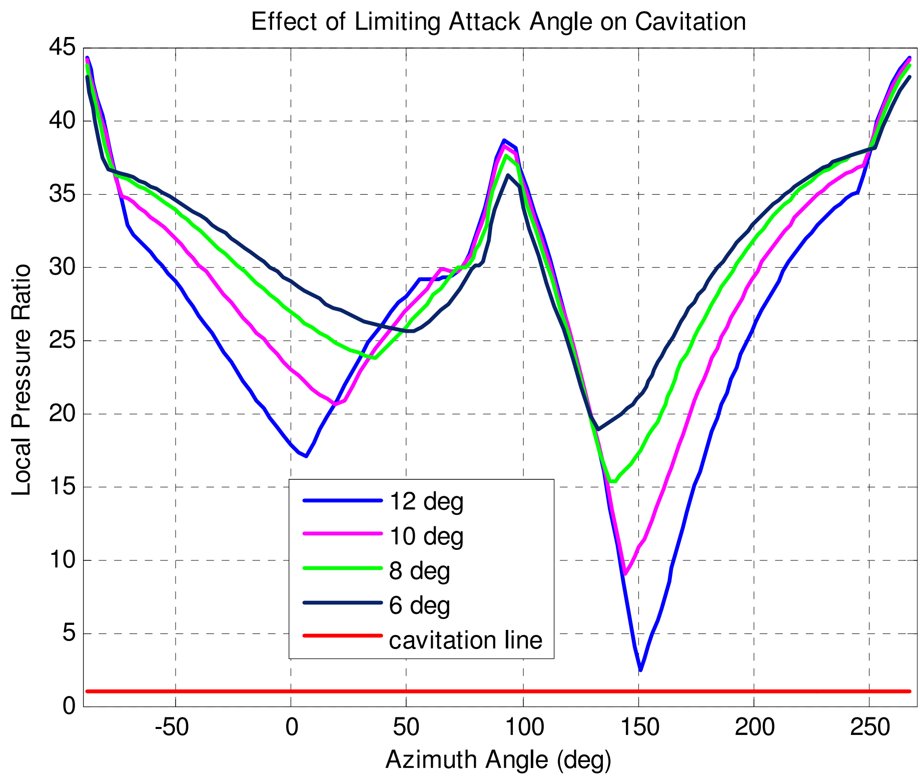

Figure 3.5 Evolution of blade local pressure ratio at stream speed of 2.2m/s

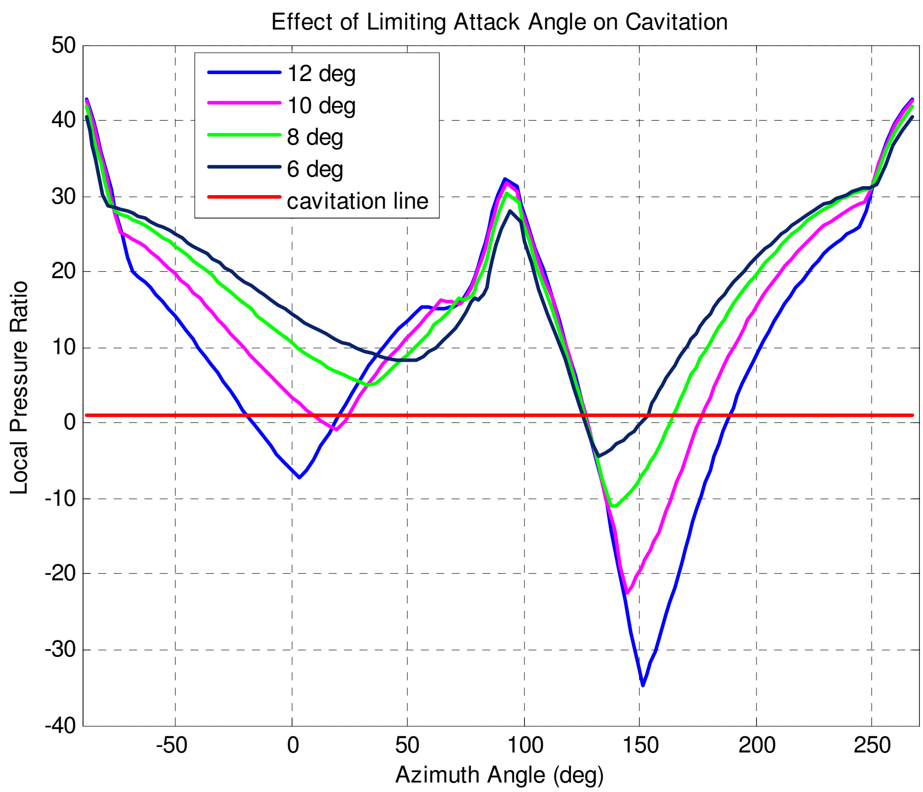

Figure 3.6 Evolution of blade local pressure ratio at stream speed of 3m/s

BLADE PITCHING MOMENT

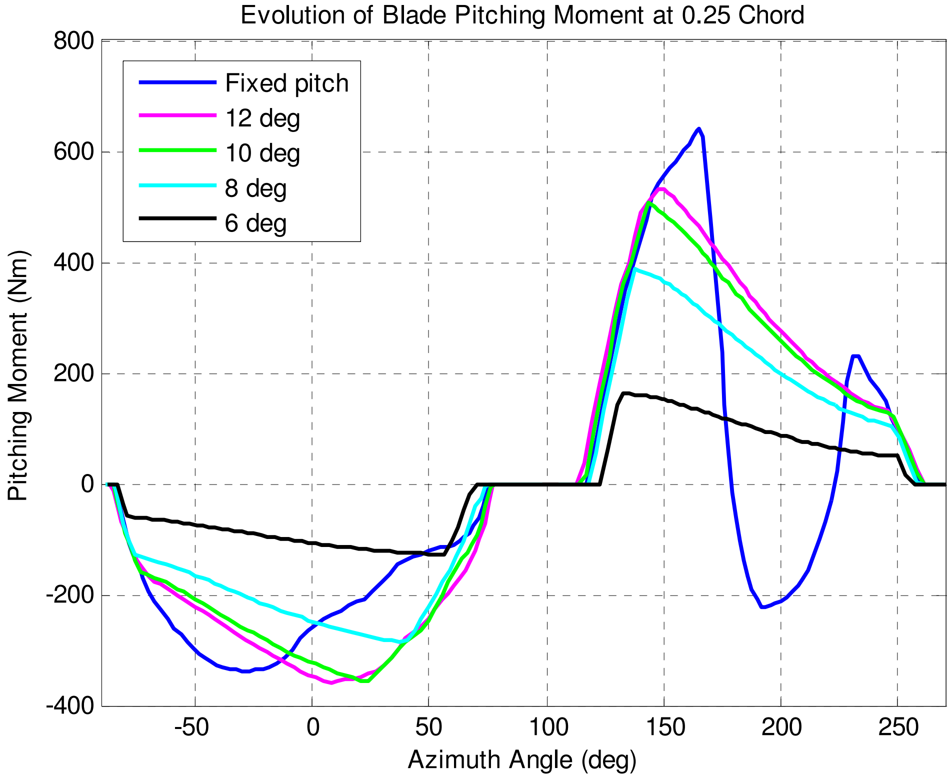

Figure 3.7 Blade pitching moment variation at a pivot point 25 % of chord

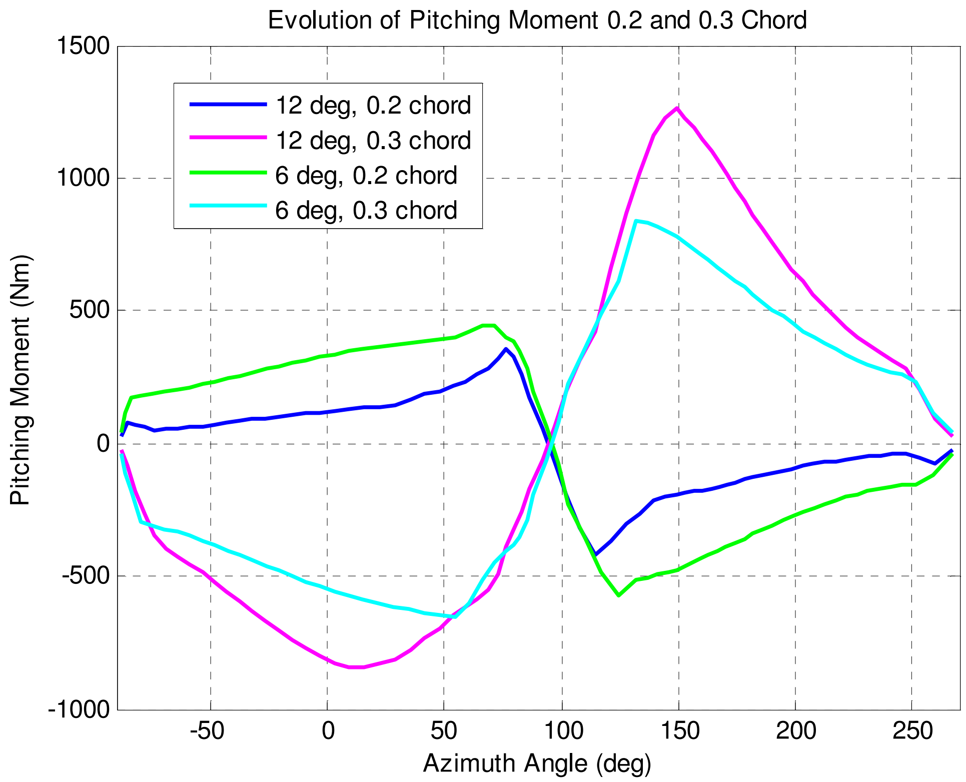

Figure 3.8 Blade pitching moment variation at pivot points 20% and 30% chord

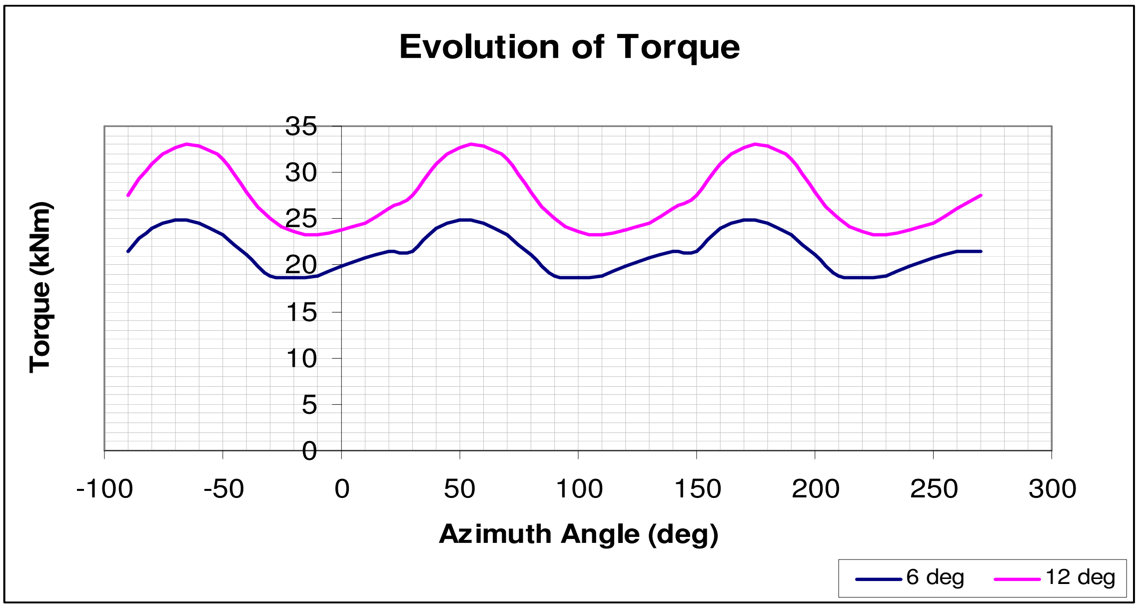

Figure 3.9 Evolution of torque for different limiting attack angles

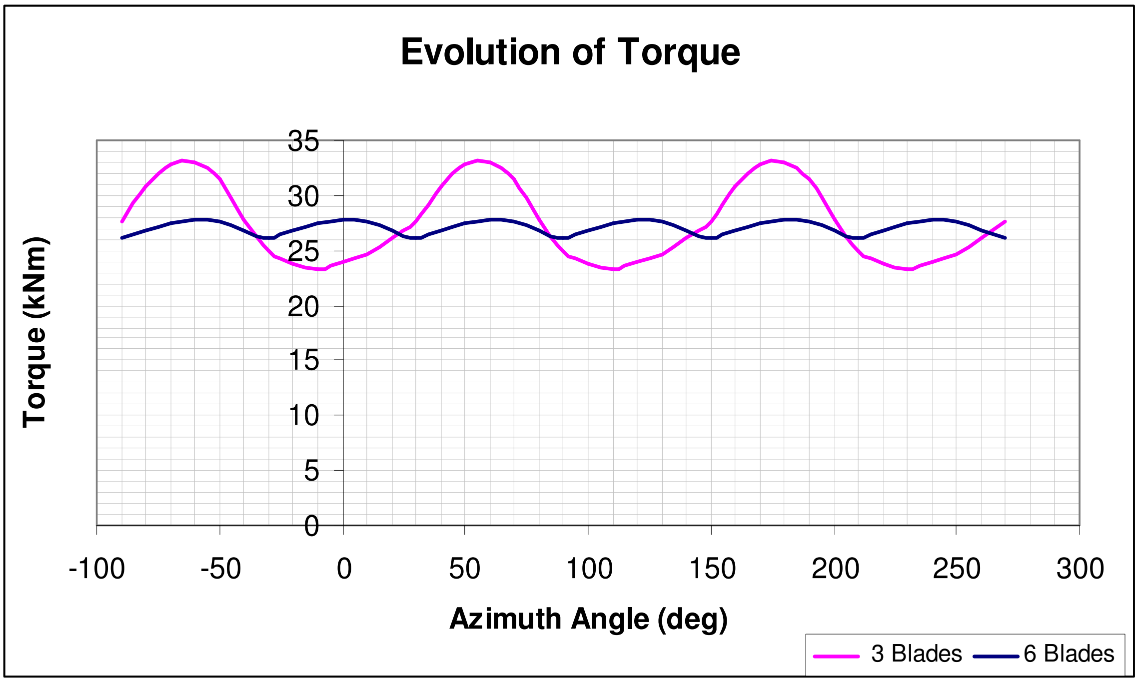

Figure 3.10 Evolution of torque for different number of blades

BLADE NORMAL LOADS

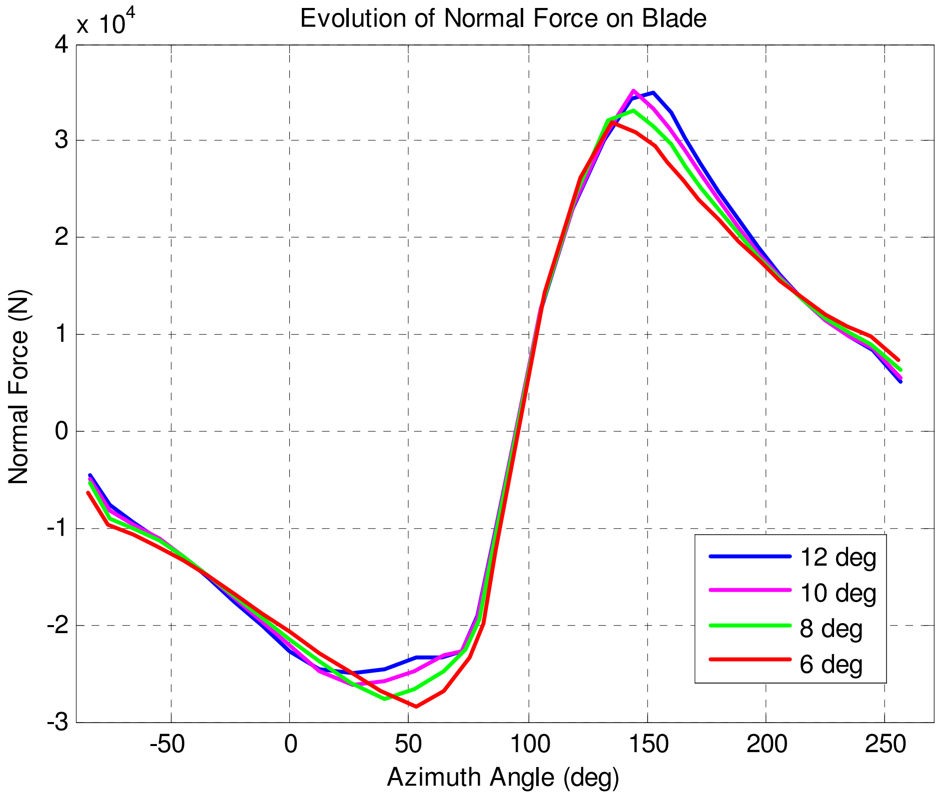

Figure 3.11 Variation of normal force on a blade

4 CHAPTER 4 – CFD ANALYSIS

4.1 Introduction

This section gives preliminary results of the CFD model under development, initially based on a fixed pitch blade configuration. Comparisons are made between results from the CFD model and the analytical model discussed in the previous section. The areas of agreement and discrepancies have also been highlighted and the improvements necessary to make a better comparison.

4.2 Simulation Setup

The turbine configurations and stream velocities used in the simulations are show in Table 4.2.1. The geometric and boundary layer setup is the same for all of the simulations performed and is described in Table 4.2.2.

Table 4.2.1- Simulation configurations

Blades	Chord (m)	Diameter (m)	TSR	Stream Speed (m/s)
3	0.25	5	3	2
3	0.25	5	4	2
4	0.1875	5	3	2

Table 4.2.2 - Boundary setup

Boundary	Geometry (m)	Condition
Inlet	5d	Velocity
Outlet	9d	Constant static pressure
Sides	5d	Free slip walls
Front & Back	N/a	Symmetry

In Table 4.2.2:

- ***Geometry*** refers to the distance from the perimeter of the turbine to the boundary
- ***Condition*** is the imposed boundary condition.

Figure 4.8, Figure 4.9 and Figure 4.10 show some details of the meshing used for the simulations. The overall meshing domain (Figure 4.8) was chosen sufficiently large (15 diameters long, 11 diameters wide) to minimise any flow artefacts that could be introduced by the chosen boundary conditions. The meshing disk that contains the turbine blades has its own rotating frame of reference. As expected, the meshing becomes progressively denser in the neighbourhood of each blade, with the cell thickness of the order of 1mm at the surface of the blade.

Velocity and pressure for inlet and outlet conditions respectively is a common and robust boundary condition. Using average static pressure on the outlet was found to be unstable. The symmetry condition on the front and back faces is determined by the notionally 2-D setup for a 3-D solver. Other conditions would represent possible alternatives for the sides. These include a pressure opening condition and symmetry. CFX technical support has advised against the use of symmetry conditions on the sides due to the possibility of un-physical reflections being created, while the use of a pressure opening on the sides may not constitute a well-posed problem.

The CFX flow solver uses a finite volume approach, combined for this study with a k-Epsilon turbulence model.

4.3 Simulation Results

4.3.1 Qualitative Discussion

Figure 4.1, Figure.4.2 and Figure .4.3 show the torque variation with azimuth angle for individual blades and the sum from all blades. The azimuth angle is that of foil A, which is at $\psi = 0$ at $t = 0$. Each simulation has been run for five complete revolutions, with the initial condition being an impulse start. This is a poor approximation as there is no wake at $t = 0$. The formation of this wake as the simulation proceeds has a clear effect on the torque output, namely that of an exponential decay of the function $Q = f(\psi)$. We can judge whether the simulation represents a fully developed picture of the turbine flow-field by comparing the relative difference of the average torque over successive revolutions:

$$e_{\overline{Q}} = \frac{\overline{Q}_{n-1} - \overline{Q}_n}{\overline{Q}_{n-1}}$$

For the two geometries at a tip speed ratio of 3, the relative difference is less than 2.5%, but for the case of the 3-blade machine at a tip speed ratio of 4, the relative difference is approximately 5%. At a higher tip speed ratio the axial induction factor is likely to be higher, and so we would reasonably expect the wake to take longer to develop.

With respect to the summation of the torque from all foils, we have a near sinusoidal pattern of torque fluctuation. This pattern is determined by the torque patterns from the individual foils (the same, but with a phase shift), and the number of foils. In respect of the former, a good pattern (equal or nearly equal peaks) will result in a good pattern from the summation. In respect of the latter, it is clear that four blades are considerably better than three blades. The CFD simulation therefore confirms, as expected, that a higher number of blades leads to less variation in shaft torque.

4.4 Quantitative Results

Simulations for 3- and 4-bladed configurations were performed each at two tip speed ratios, 3 and 4. Table 4.4.1 shows quantitative data relating to the simulations performed, for a tip speed ratio of 3.

		Foil A			Sum of Torque		
Configuration	**Cp**	**U Max T_z**	**L Max T_z**	**U Max / L Max**	**Max**	**Min**	**Max/ Min**
3 x 0.25, 3	0.536	4997	1301	3.84	6335	2943	2.15
4 x 0.1875, 3	0.533	3544	1076	3.30	4843	4224	1.15

Table 4.4.1 : Quantitative results from CFD simulations

U Max T_z and L Max T_z are the upper and lower maximum torque on foil A in a complete revolution, which occur at an azimuth angle of approximately 180° and 0° respectively. Ideally we would like these to be even, something that could be achieved with variable pitch. In the present fixed pitch configuration, it is visible that operating a three blade machine at a tip speed ratio of 4 instead of 3 affects adversely its torque output. What is happening is that the blades are extracting a considerable amount of the energy in the upstream pass such that there is less available for the downstream pass.

4.5 Comparison of Analytical Model with CFD Model

Table 4.4.1 and Table 4.5.1 list results from the analytical model and the CFD simulation respectively. Figure .4.4 and Figure 4.5 present results for the 3 blade, 0.25m chord configuration at a tip speed ratio of 3 from the analytical model and the numerical simulation. Figure 4.6 and Figure 4.7 present results for the 4 blade, 0.1875m chord configuration at a tip speed ratio of 3.

		Blade A			Sum of Torque		
Configuration	**Cp**	**U Max T_z**	**L Max T_z**	**U Max / L Max**	**Max**	**Min**	**Max/Min**
3 x 0.25, 3	0.452	3597	1886	1.91	5522	2423	2.28
4 x 0.1875, 3	0.434	2610	1470	1.78	4655	2471	1.88

Table 4.5.1 : Results from Analytical Model

4.6 Observations

Some clear observations can be made: the numerical simulation predicts higher C_P for both cases where comparisons can be made. In each case this appears to be due to the numerical simulation not predicting stall. For the 3 blade case, the largest discrepancy between analytical and numerical solutions occurs for azimuth angles between ~ 160 and 225°. Over this range the angle of attack is greater than 14°, for which the lift and drag coefficient data used as input to the analytical model predicts stall.

It is interesting to note that the difference between the two models in the 260 – 360 Degree azimuth range is consistent with the numerical model predicting more energy capture in the upstream pass, and therefore extracting less than the analytical model on the downstream pass.

This discrepancy is more pronounced for the 4 blade case. The pattern is the same – the largest discrepancy again occurs for azimuth angles between ~ 160 and 225°, for which range the angle of attack is greater than 14° – and the greater discrepancy may be accounted for by the fact that the chord Reynolds number is lower (the chord is shorter), and so stall will occur at a lower angle of attack.

Qualitatively, the wake structure, as shown in Figure 4.11 and Figure 4.12, shows features that are consistent with power extraction at the rotor.

4.7 Conclusions

Overall, there is promising evidence of agreement between the two models, both in terms of magnitude of the predicted torque and its azimuthal distribution. The numerical model is not yet capable of predicting stall, but nevertheless the results of this simulation already give a degree of confidence in the predictions of the analytical model.

There is a clear need for greater understanding of how CFX (and CFD in general) predicts the lift and drag coefficients from lifting bodies. This is being pursued in earnest to give a better comparison between the two models.

The effect of side baffles on the turbine flow has not been investigated, but the numerical simulation may prove a useful tool for this. There are some claims that ducting the flow may improve the capture efficiency of the turbine; this, if proved correct, could contribute positively to the economics of the device.

4.8 Figures (CFD analysis)

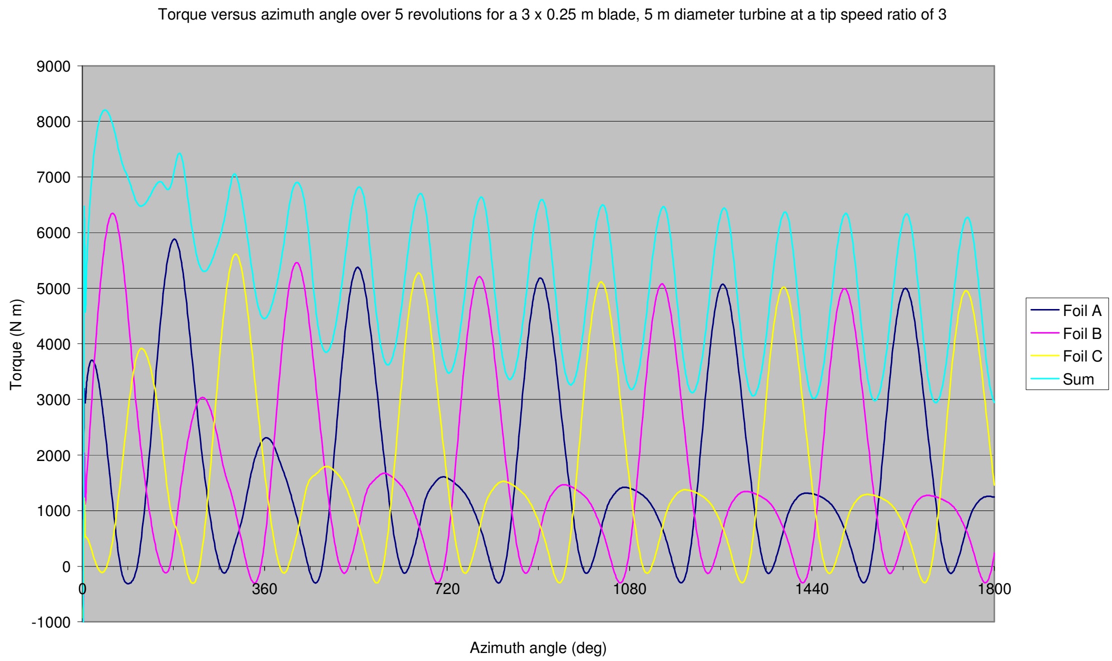

Figure 4.1 : Torque variation for 3 x 0.25m blade turbine at a tip speed ratio of 3

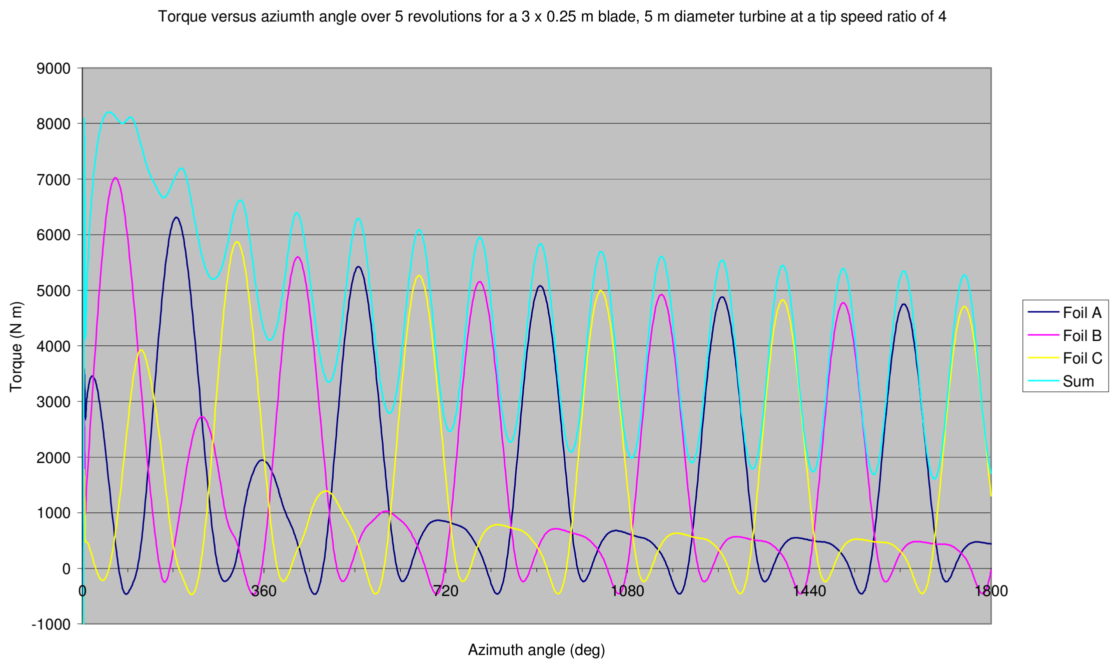

Figure.4.2 Torque variation of 3 x 0.25m blade turbine at a tip speed ratio of 4

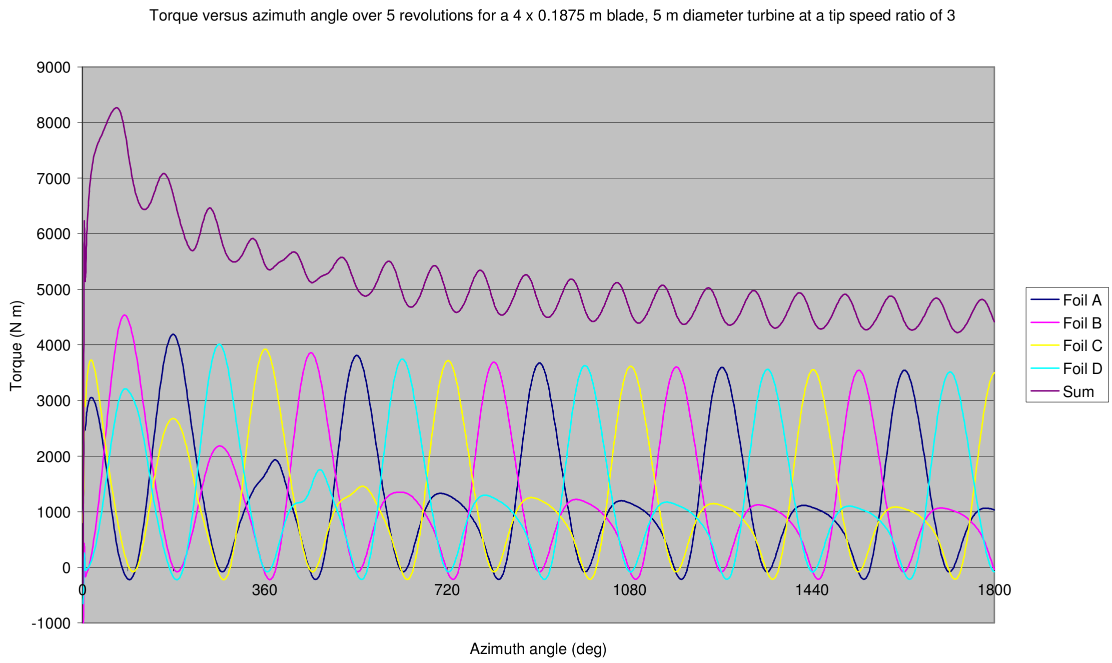

Figure .4.3 Torque variation of 4 x 0.1875 m blade turbine at a tip speed ratio of 3

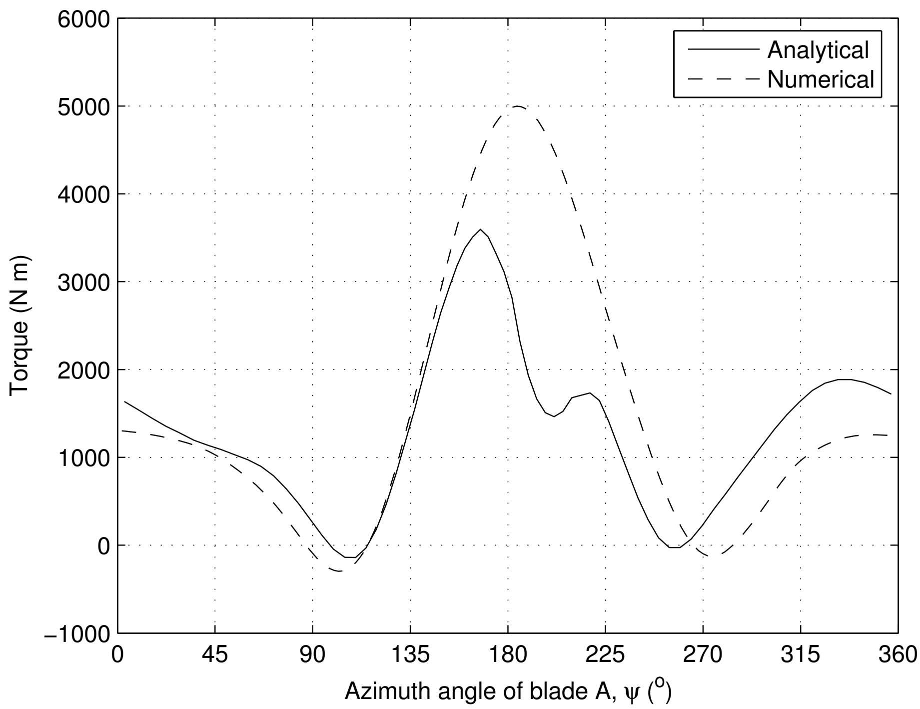

Figure .4.4 – 3 blade, 0.25 m chord, 5 m diameter turbine at a tip speed ratio of 3: a comparison of torque for 1 blade (versus azimuth angle) between the analytical model and the CFD simulation.

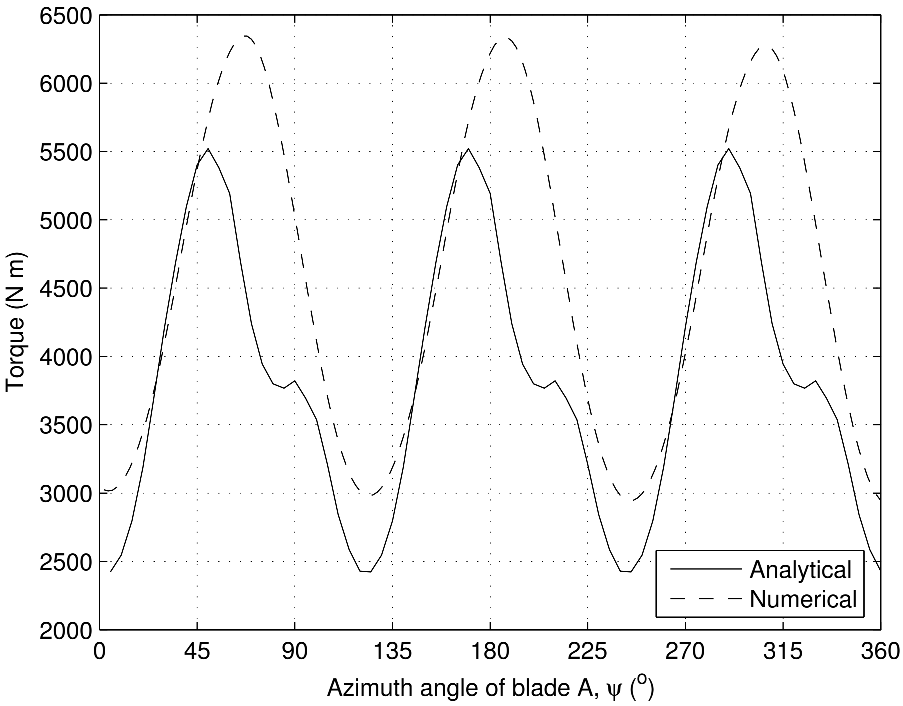

Figure 4.5 – 3 blade, 0.25 m chord, 5 m diameter turbine at a tip speed ratio of 3: a comparison of the sum of torque from all blades (versus azimuth angle) between the analytical model and the CFD simulation.

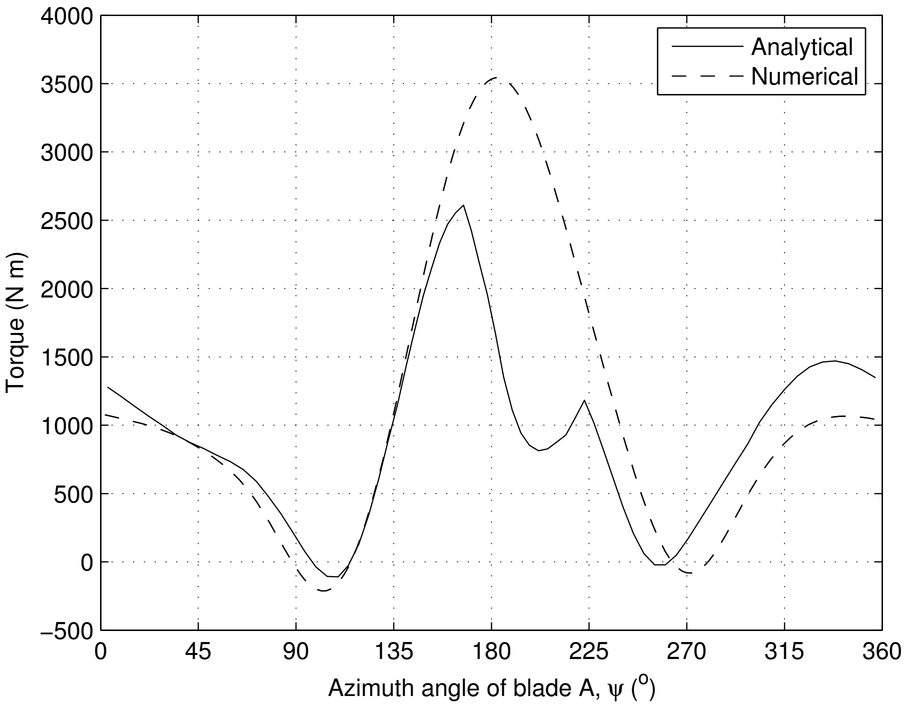

Figure 4.6 – 4 blade, 0.1875 m chord, 5 m diameter turbine at a tip speed ratio of 3: a comparison of torque from 1 blade (versus azimuth angle) between the analytical model and the CFD simulation.

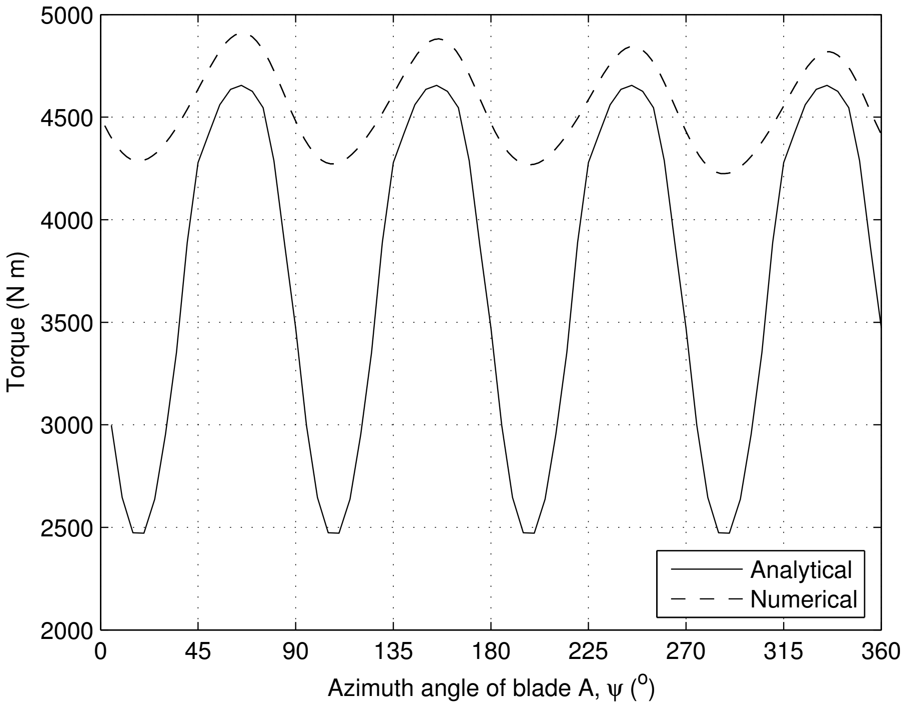

Figure 4.7 – 4 blade, 0.1875m chord, 5m diameter turbine at a tip speed ratio of 3: a comparison of the sum of torque from all blades (versus azimuth angle) between the analytical model and the CFD simulation.

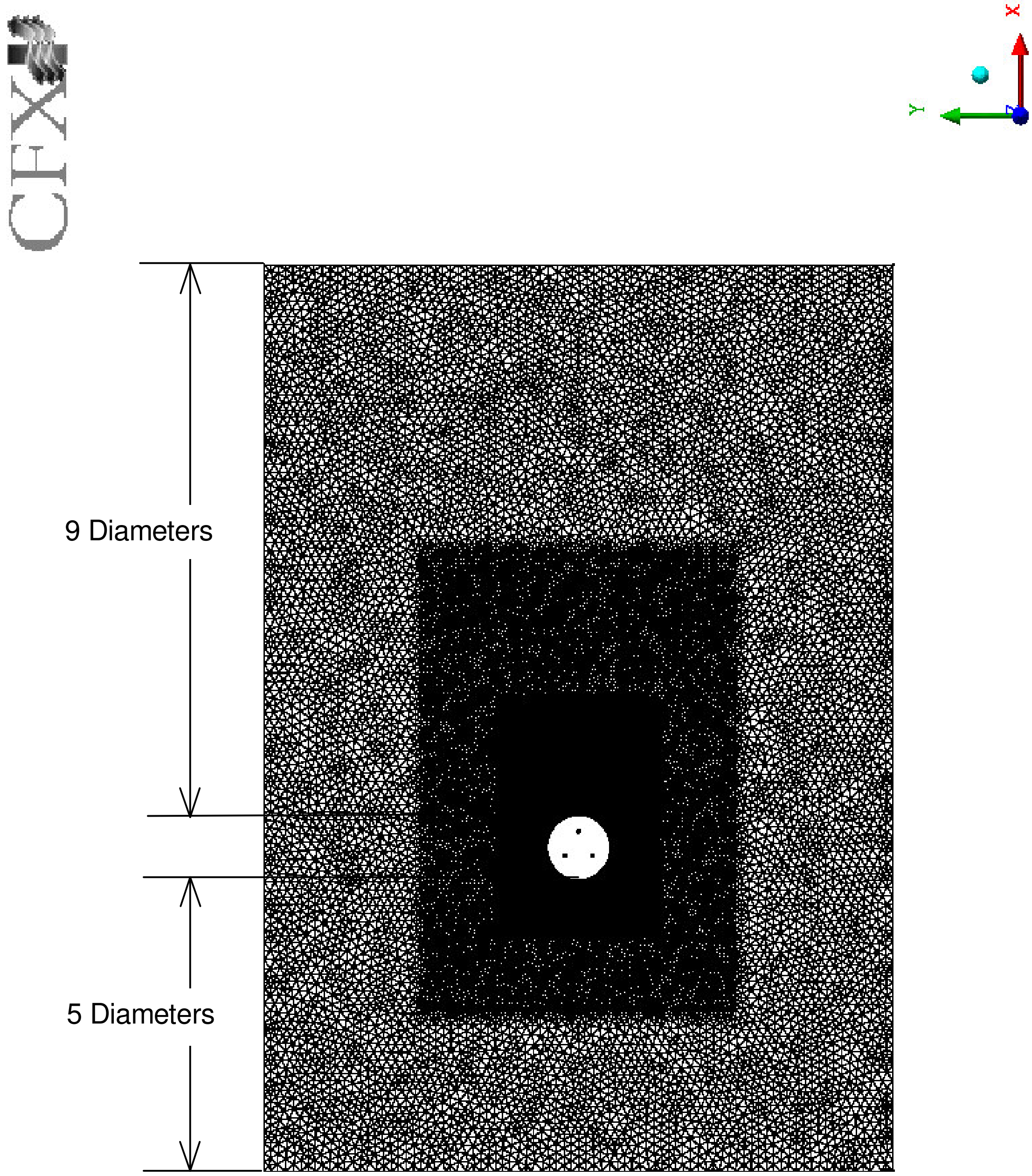

Figure 4.8: View of Complete Simulation Mesh

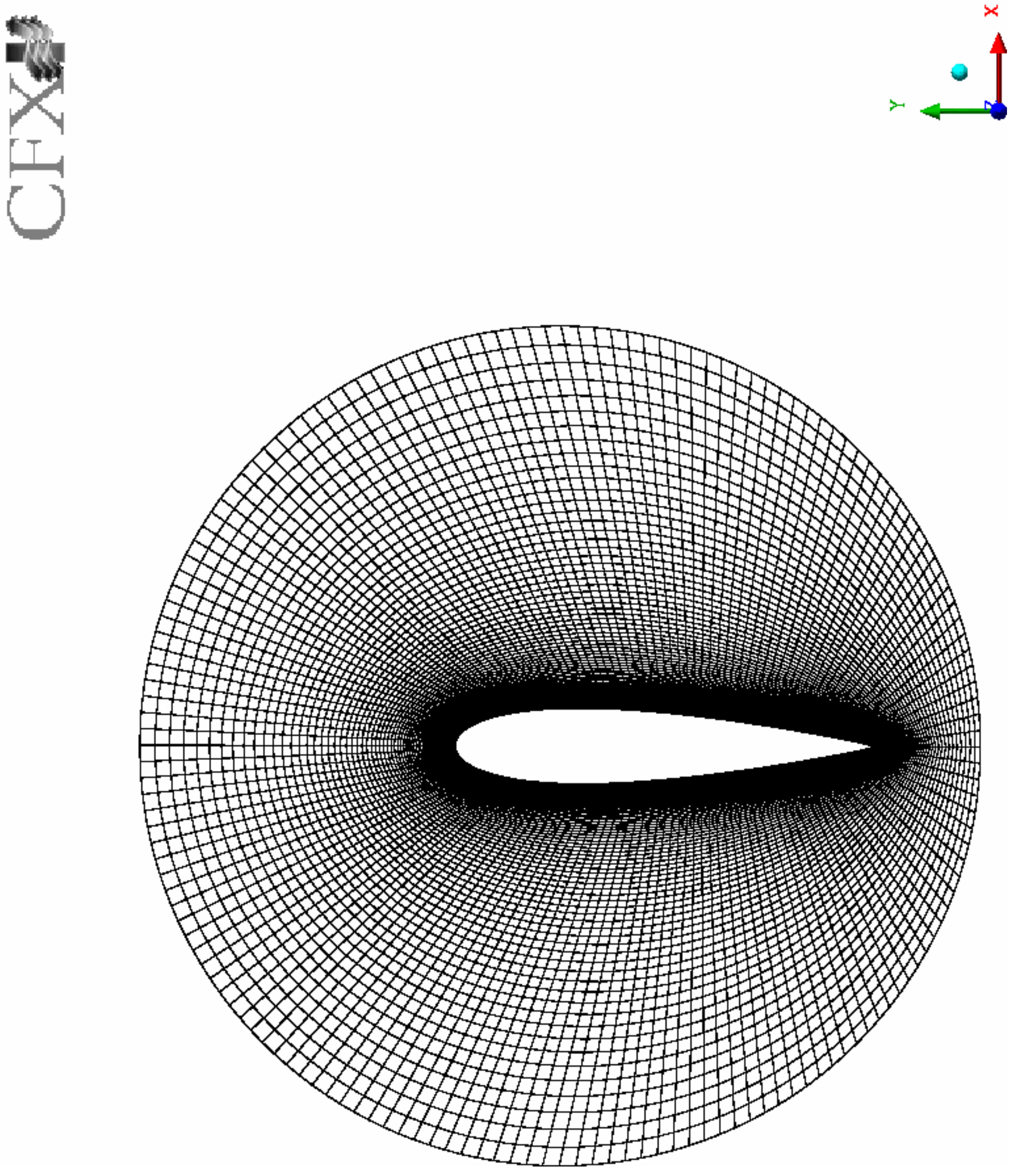

Figure 4.9 : View of Mesh around Turbine Blade

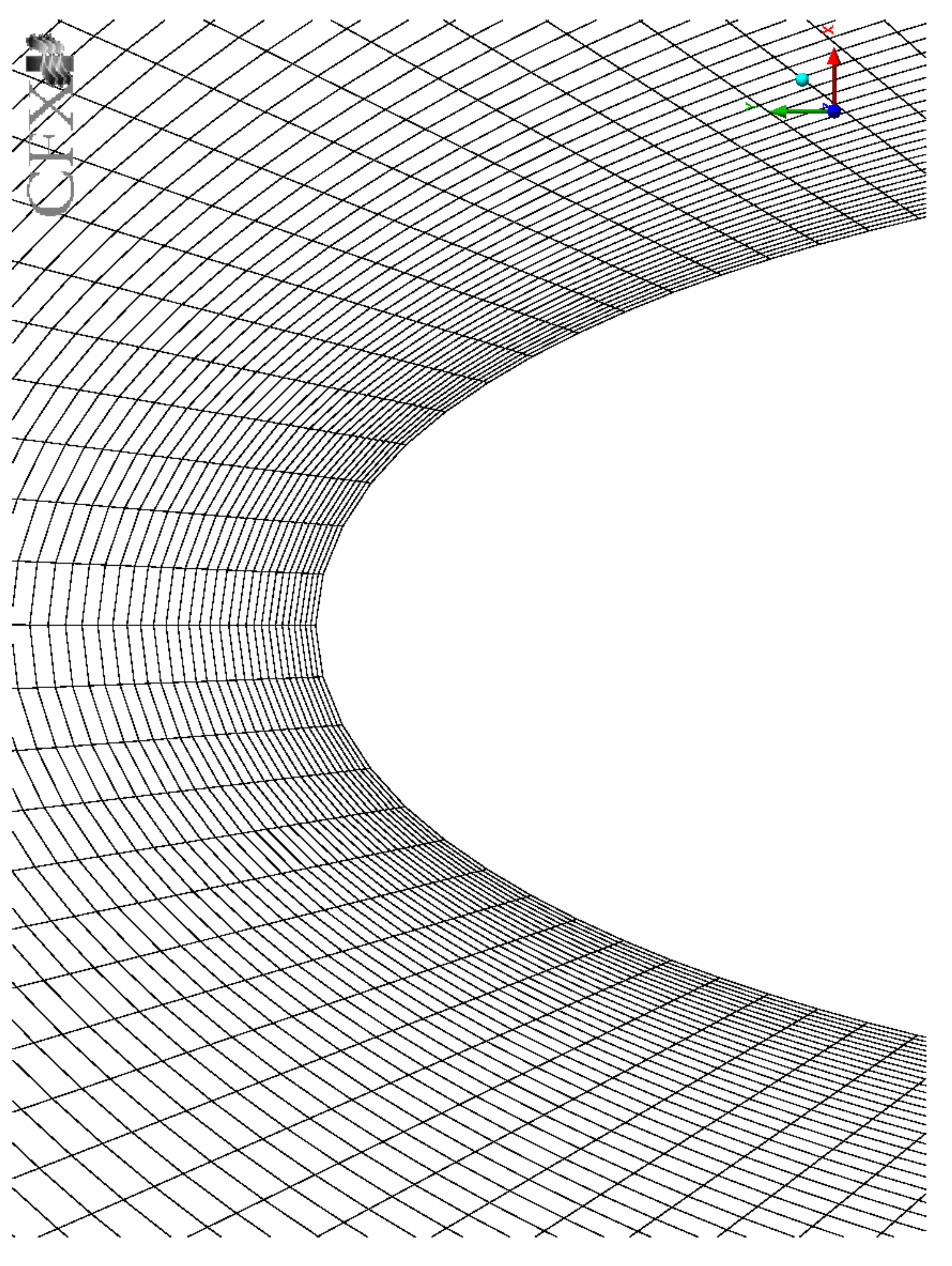

Figure 4.10 : Detail of simulation mesh around a blade leading edge

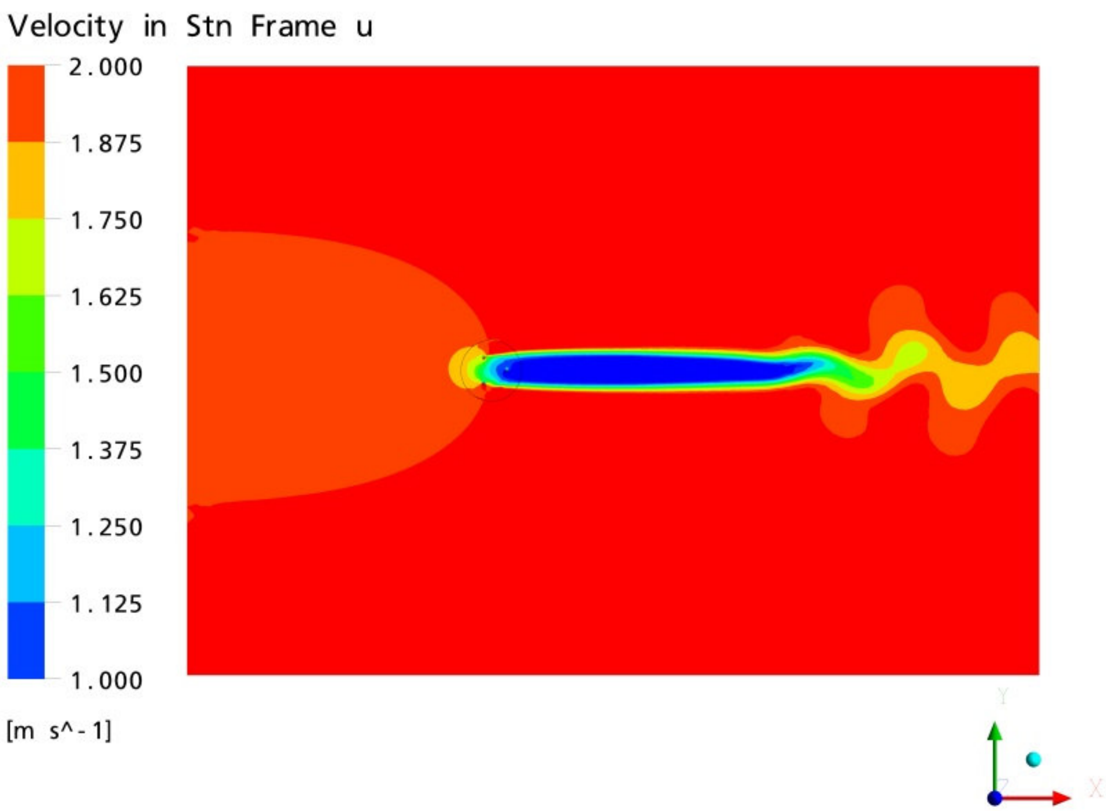

Figure 4.11 : Axial (U) velocity component in fixed reference frame.

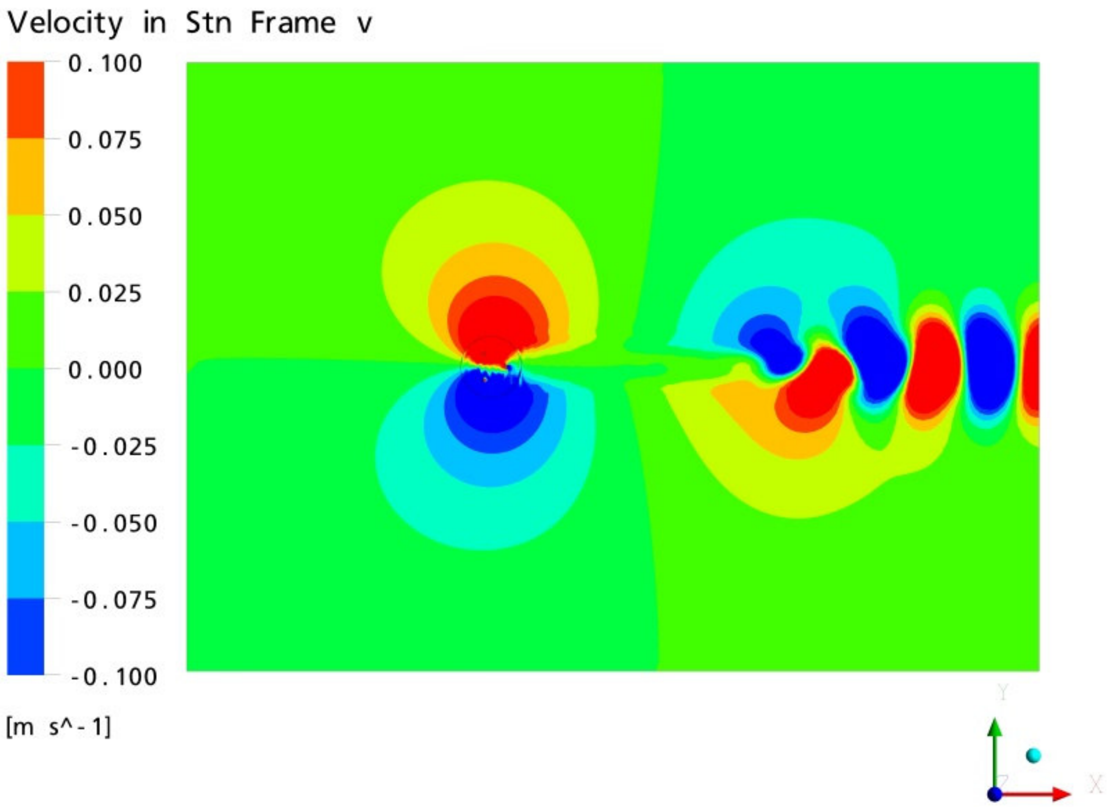

Figure 4.12 : Transverse (V) velocity component in fixed frame of reference

5 CHAPTER FIVE - STRUCTURAL ANALYSIS

The marine environment is a hostile one. The rigidity and stability of the whole machine can not be overemphasised. This chapter looks at the structural requirements of the floating vessel as well as the mooring system required to keep the system stable. The natural dynamic characteristics of two designs have been simulated under the same load conditions.

5.1 Objectives

- Investigate the natural dynamic characteristics of the structure
- Design a simple but effective mooring system
- Determine optimum point of application of mooring force
- Determine optimum inclination of mooring force

5.2 Turbine Description

The turbine system can be divided into three principal components: the rotor, power train and support structure. The rotor consists of evenly spaced vertical blades of symmetric foil section held between two rings by a central strut. The top ring houses the pitch control mechanism for each blade. Thin spokes in tension then connect the blade to the hub for the purpose of transferring torque. A shaft then connects the hub to the gearbox coupled to the generator forming the power train. This power train is supported and housed on a platform above the water surface. The platform is kept floating by a pair of hulls much like the catamaran. Invariably the rotor hangs from the bottom of the platform while the gearbox/generator units as well as its attended electrical equipment sit on the platform.

5.3 Approximation in Prokon

Three models were created in PROKON[7] for analysis to aid in the design. The models are:

- Rotor frame
- Support structure
- Turbine unit (rotor, power train and support structure)
- Below is a brief description of the models.

5.3.1 Rotor Frame

The rotor frame is as detailed in Figure 5.1. The rings were approximated to straight beams connecting each blade. The blades were represented by rectangular hollow tubes. The pitch control mechanism at the top of each blade was represented by point load whiles the drag and lift forces on the blades were represented by beam loads. The whole structure was supported at the end of the shaft.

5.3.2 Support Structure

The shape of the support structure, as in Figure 5.2, was influenced by that of the catamaran. This boat when designed well slides through water effortlessly while maintaining its stability even under turbulent conditions. The hulls are made of a steel frame covered by steel plates. Hollow square tubes form the platform with the necessary crossbeams for rigidity.

The last model is a combination of the support structure and the rotor hanging beneath it and the gearbox/generator unit housed on the platform as in Figure 5.3.

5.3.3 Loads on Structure

The loads on the structure operating in a stream speed of 3m/s are as shown in Table 5.1. The unit is intended to be floating and therefore the structure should be such that there is enough buoyancy to do that. Hollow beam sections were chosen so as to provide some buoyancy.

Since there is more than enough buoyancy to keep the unit floating with all its ancillary equipment it may be necessary to fill up the hulls with water to keep the deck just touching the water surface.

The drag forces acting on the system include the drag on the structure acting on the shaft and hulls as well as the thrust force on the rotor. The horizontal mooring force is therefore equal and opposite to the total drag on the structure. It is worth noting that the drag forces on the structure have been overestimated. This is because a hydrodynamically designed hull will result in lower drag coefficients and hence lower drag forces.

Table 5.1 Loads on structure

Parameter	Value
Weight (kN)	71
Buoyancy (kN)	137
Drag (kN)	18.5
Horizontal Mooring Force (kN)	56
Rotor Thrust Force (kN)	37.4

5.3.4 Mooring Design

The main requirement of the mooring design is to resist forces in the opposite directions of the stream speed as well as some variability. Figure 5.4 shows the mooring system under consideration for a water depth of 30m. It is made up of a pair of light chains connected to the sides of the turbine structure. A heavy chain then links the light one to the anchor or pile via a clump weight in both directions of flow. This design was developed in consultation with Balmoral Marine Ltd, an authority in the field of offshore mooring design and supply.

The use of an anchor or pile will depend on the seabed geology. Well-scrubbed rocky bottoms such as that found in the Pentland Firth will require drilled piles while hard clay to soft soil seabed can be handled by existing anchors used in the oil industry. The *Mk* range of anchors provides a viable solution in this regard. The estimated lengths of chain

[7] PROKON – Structural modelling and analysis software

required are displayed in Table 5.2 while Table 5.3 contains the fittings needed for the mooring system.

Table 5.2 Chain length

Chain	Length (m)
Light chain/Synthetic rope	4 x 42
Heavy chain	4 x 3
Distance between anchor points	78
Water depth	30

Table 5.3 Fittings

Fitting	Number
'D' Anchor Shackles	16
Long 'D' Hanging Shackles	2
2 Tonne Stevin MK3 Anchors	2

5.4 Simulation Analysis

Each model described above was subject to two main analyses as described below.

5.4.1 Linear Analysis

This static analysis was first performed to verify the basic integrity of structure. It involves subjecting the structure to the necessary external forces as well as its own weight and adjusting the members (position and size) such that their deflection is within 1.5% of their length. For instance the weight of the pitch control mechanism is considered an external load for the rotor model while the weight of the rotor is considered as an external load when analysing the support structure.

5.4.2 Dynamic Analysis

The dynamic analysis simulates the frame's natural modes of vibration. The main objective here is to minimise or possibly eliminate the pitching, rolling and yawing characteristics of the structure. This was performed on the turbine structure only, which consists of the rotor, power train and support structure. The buoyancy and mooring force were specified as spring forces. The following simulations were carried out:

- *Variation of point of application of mooring force*: This simulation involves changing the point of application of mooring force along the vertical of the hulls taking the bottom of the deck of the structure as the datum. The inclination of the mooring force was assumed to be at 45^0 to the horizontal.
- *Variation of the inclination of the mooring force:* In this case the effect of angle of inclination of mooring force on stability was simulated. The angle of inclination was varied from 20 to 60 degrees to the horizontal. It is worth noting that the structure has enough buoyancy to prevent it from sinking under the influence of the vertical component of the mooring force.
- *Variation of angle of heel*: Assuming a wave of length 8m and amplitude of 2m then a maximum angle of heel of 14^0 is expected considering the geometry of the structure. The stability of the structure to rolling and pitching up to this heel angle is traced. Assuming 45^0 inclination of mooring forces.

5.5 Results - Dynamic Modes

Locating the point of application of mooring force 0.5m below the deck gave the best stability against pitching. This is as a result of its short and fat hulls. The only visible motion in this case is surfing in the stream direction. Locating the mooring point outwit this point induces a moment in the x-direction leading to pitching.

The floating turbine structure is relatively stable under a wide range of angle of inclination of the mooring force. This is as a result of the fatter hulls providing enough damping, as the buoyancy is concentrated there. A choice of 45 degrees angle of inclination of mooring force was made due to cost considerations. That inclination uses less mooring rope compared to a 30 degrees inclination, which also gave reasonable stability.

The last investigation involves subjecting the structure to a wave of length 8m and amplitude of 2m. An angle of heel of 14^0 is possible with respect to the length of the support structure. The structure was found to be stable even when the angle of heel was 14^0. The reason is that the moment arm of the resultant vertical force is too short to induce significant pitching and rolling motion.

The structure was found to be immune to yawing motion. This is because the simulation was carried out for steady conditions and the mooring forces were equally distributed on either side of the hulls and the torque was constant. A transient definition of the problem is likely to induce an unequal distribution of the mooring force. As the torque transmitted by the rotor changes with stream speed the structure is likely to yaw. This effect will increase with decreasing moment of inertia of the rotor and shaft.

One solution to this problem is to have two contra-rotating rotors on the same support structure to cancel out the torques. This translates into cost savings with regard to the structure as well as mooring requirements. The drawback may be that in the case of a fault in one rotor the other may have to be shut down so the fault can be rectified. During maintenance also, both rotors may have to be shut down. This increases the machine's idle time and the costs involved must be accounted for. The effects of having these units close together on the flow needs to be evaluated as well. This needs further investigation.

Another solution is to have a restrictive mooring system. In this case the mooring forces are applied at four different points on the support structure. The mooring system is such that the mooring chains or ropes are always in tension eliminating the six degrees of freedom (translation and rotation in the x, y and z-direction) on the structure. It is obvious that the mooring chains will undergo cyclic loading, being heavily stressed during high current speeds. They will have to be checked regularly and replaced if necessary. This brings into play added cost which must be factored into the maintenance cost.

5.6 Figures

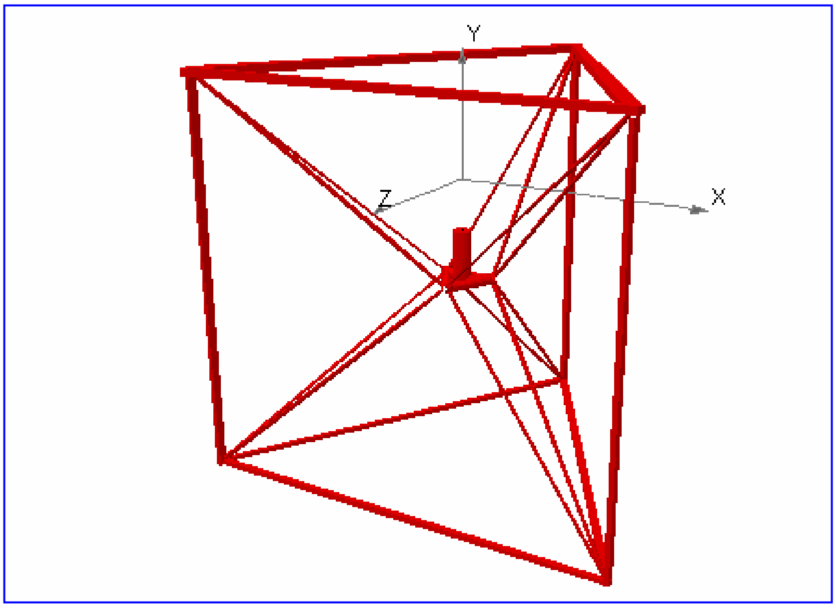

Figure 5.1 Rotor frame

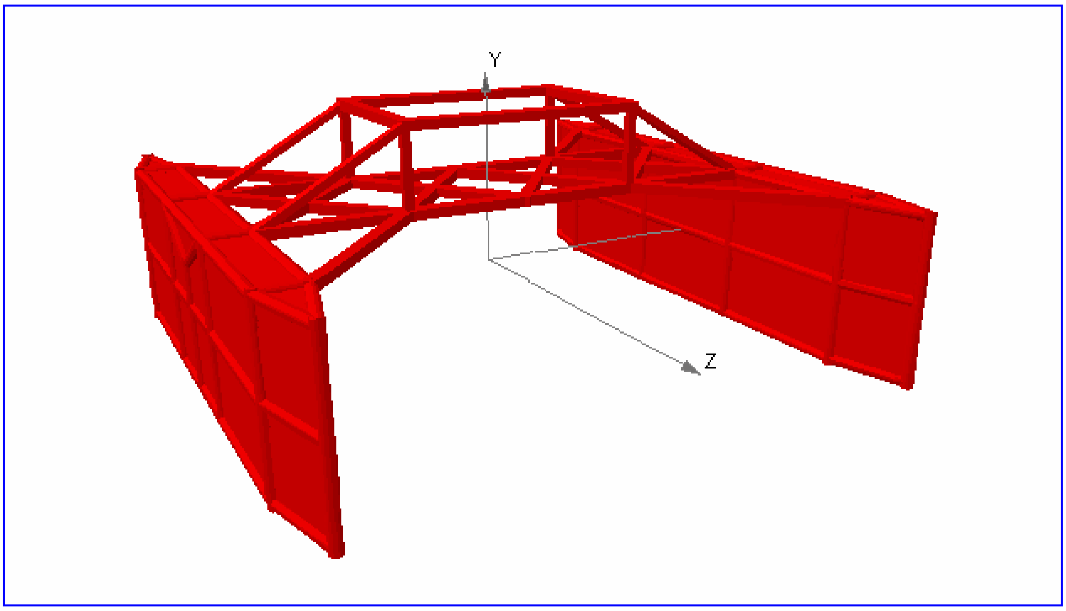

Figure 5.2 Turbine support structure

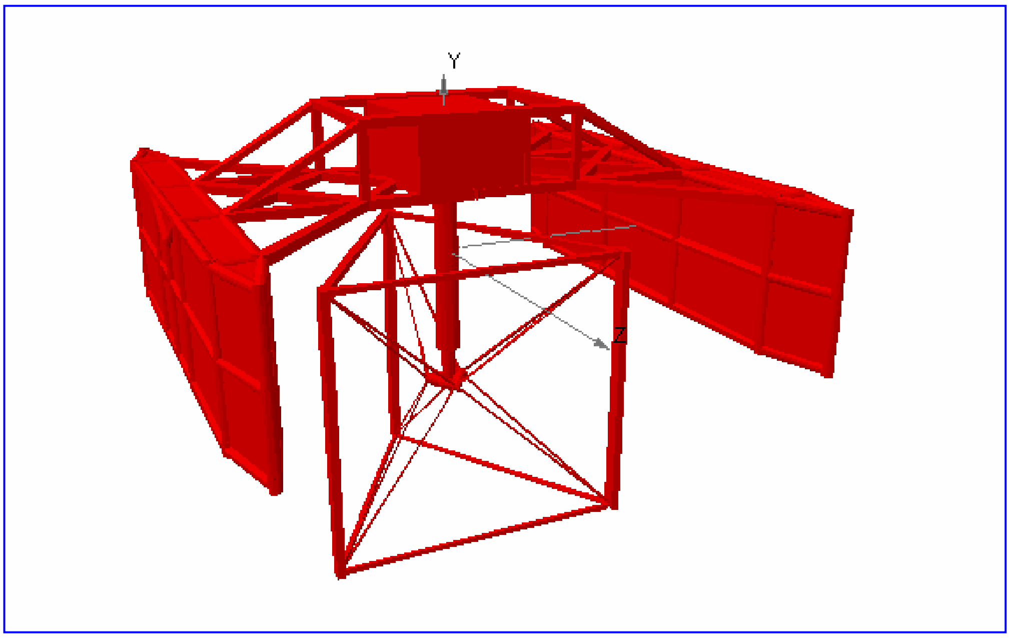

Figure 5.3 Turbine structure

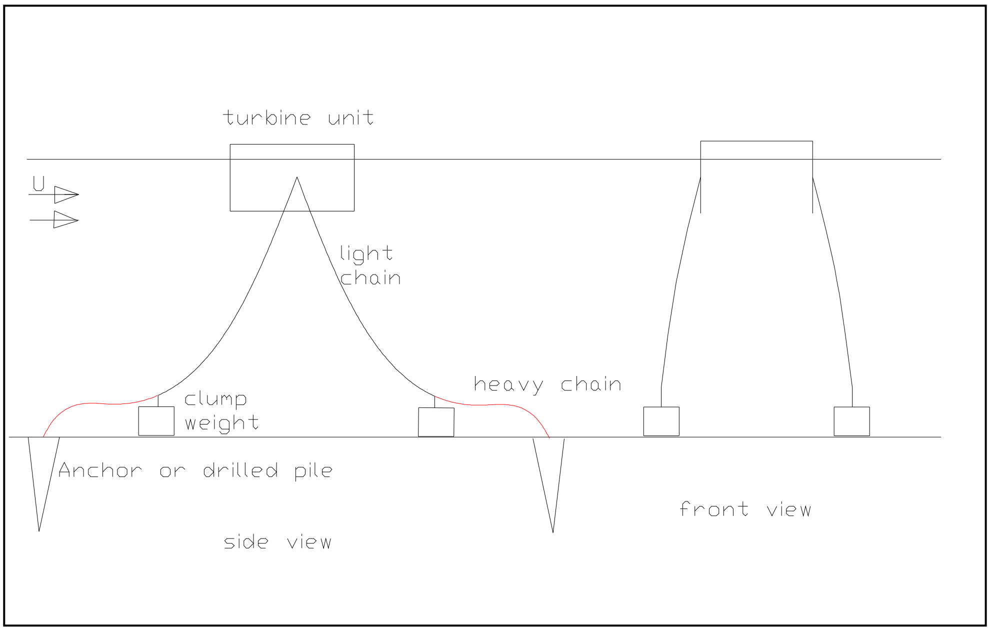

Figure 5.4 Schematic diagram of mooring design

6 CHAPTER SIX – OPTIMISATION AND POWER PREDICTION

6.1 Introduction

The typical operating characteristics of the turbine have been outlined and investigated in section 3 while the proposed structure was subjected to vigorous tests in section 4. The benefits of active variable pitch control were clear while the stability of the structure was satisfactory. This section goes through a design process of matching the technology to a specific site. The first part looks at the resource characteristics followed by the design of a device to exploit the resource. The expected average power and energy generated per annum are also calculated.

Tidal currents are known to vary periodically with time. Moreover it is constrained between the seabed and the free surface. Knowledge of the typical daily variation and vertical velocity profile will play a major role in designing a machine to suit those conditions. The energy model set up in MatLab also gives statistical summary of site specific tidal data (time and height dependent stream speed and direction). These include the annual velocity distribution of the site and a tidal rose. The annual velocity distribution gives the percentage of time a particular velocity occurs in a year while the tidal rose gives the relative frequency of the tidal stream direction.

6.2 Site Data Analysis

Tidal data (stream speed and direction) have been acquired for Cullivoe Ness over a period of 15 days from 23/03/2004 to 06/04/2004. This therefore covers both spring and neap tides. The data is taken at 10-minute interval and at a vertical height variation of 1m. The average depth of the site is 28m and the typical tidal range is 0.8m.

Figure 6.1 shows a typical daily variation of the stream speed for spring and neap tides. On the average the maximum stream speed during spring tides is about 2.4m/s whiles that for neap tides is 1.4m/s. This means this a fairly low tidal current site. It is also worth noting that the variation with time is different from the assumed sinusoidal variation. The period for a cycle is however close to the mostly assumed value of 12.4hours (744minutes) even though this is not constant, 740minutes on the average.

Figure 6.2 also shows the typical vertical variation for springs as well as neaps. The stream speeds seem fairly constant over the upper 40% of the vertical especially at low speeds. This is due to the fact that the seabed friction increases with the square of the stream speed. The boundary layer effect becomes more paramount towards the seabed resulting in a considerable drop in the speed 5m towards the seabed. It is advisable to avoid the first 10m from the seabed due to diminishing stream velocity. The sudden drop in speed towards the free surface is likely due to the action of waves on the instrument. This zone must also be avoided. In this case 2-3m below the surface.

The directions of flood and ebb flow are not always directly opposite (Figure 6.3), an offset of 15 degrees average in this case. This poses no problem for the vertical axis configuration as it can take incoming flow from any direction.

Figures 6.3 and 6.4 show a statistical summary of tidal stream data for Cullivoe Ness. Since the 15 days data covered both spring and neap tides it was scaled up to one year assuming that period was typical of the site.

6.3 Matching Design to a Specific Site

The main constraint in the matching a device to a specific site is the height of the blades. It should not be too long to operate in the decreasing part of the velocity profile. Also its length should be such that the loading on it leads to an acceptable deflection.

6.4 Design Procedure

This procedure incorporates a front and backward technique. First a reasonable rated speed is chosen knowing the velocity distribution of the site. A limit is also set on the blade height depending on the vertical velocity profile. It must be such that it avoids both the wave zone and seabed boundary layer. The rotational speed corresponding to the rated speed is found from the performance characteristics once solidity is chosen. The corresponding load on a unit length of blade is found. This paves the way to calculate a blade height necessary to keep the deflection under 1.5% of the length. The procedure can be repeated starting from the top or in the middle, i.e. choosing a new rated speed or varying the solidity until a favourable aspect ratio is found.

6.5 Performance Predictions

6.5.1 Number of Blades

Two systems were investigated. The dimensions were chosen such that they had the same solidity and hence same performance curves. The blades are made of GRP with a central steel strut.

Table 6.1 Design parameters for 3 and 6-bladed system

Rotor	A	B
No of blades	3	6
Chord length (m)	1.2	0.6
Normal Force/unit length (KN/m)	33	16.5
Optimised blade height (m)	12	7

For the same height of blade the 3-bladed system was found to be more rigid structurally owing to the larger chord and hence higher moment of area. The 6-bladed rotor blades can be reinforced with steel lining thereby driving up the optimised height.

6.5.2 Speed Control

Figure 6.5 shows shaft power as a function of rotational speed at different stream speeds for the 3-bladed variable pitch machine. The shaft power is seen to increase to a maximum for each stream speed for a particular value of rotational speed. Higher stream speeds have more power in the currents, and the change in tip speed ratio with increasing stream speed causes the maximum to shift to a higher rotational speed. Maximum power is reached at 4.3rpm in a stream speed of 1.6m/s and 8.6rpm in a stream speed of 3.2m/s.

Turbine can be operated at its maximum coefficient of performance over the entire current velocity range between cut-in and rated speed.

6.5.3 Pitch Control

Table 6.2 show that variable pitch operation results in a 20% increase in rotor efficiency. This accounts for the extra 250kWh of energy generated per annum for the variable pitch machine.

Table 6.2 Performance predictions of fixed and variable pitch machine

	Fixed Pitch	Variable Pitch
Average Cp	0.40	0.48
Rated Speed (m/s)	2.4	2.4
Rated rotational speed(rpm)	7.2	6.2
Average power (kW)	114	139.5
Annual Energy (kWh)	873	1122

6.6 Conclusion

The site data analyses for Cullivoe Ness have provided a better understanding of the nature of the tidal stream resource. Notably the effect of the seabed boundary layer on the vertical velocity profile and the offset between the flood and ebb flow. This has helped to design a suitable device for the site most especially the limitation placed on the blade height.

The 3-bladed turbine was found to be more rigid then the 6-bladed one for the same height and solidity. Noting that the 6-bladed turbine has a better torque fluctuation characteristics the blade can be reinforced with a steel lining to make it rigid thereby driving up the optimum height.

Introducing pitch variation has demonstrated its benefits, which accounted for 20% increase in efficiency.

6.7 FIGURES (DESIGN OPTIMISATION)

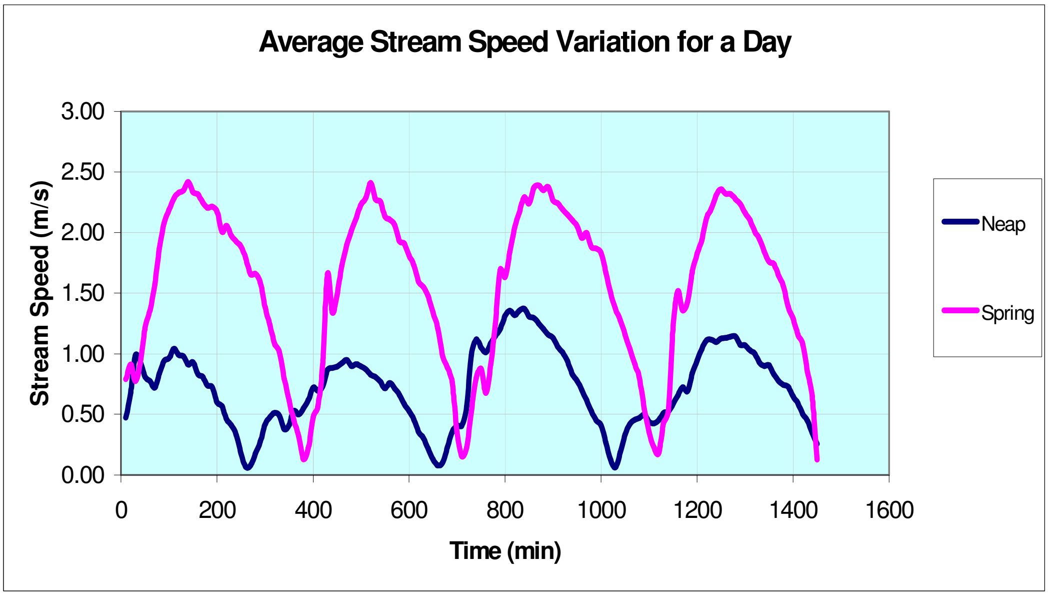

Figure 6.1 Variation of stream speed with time

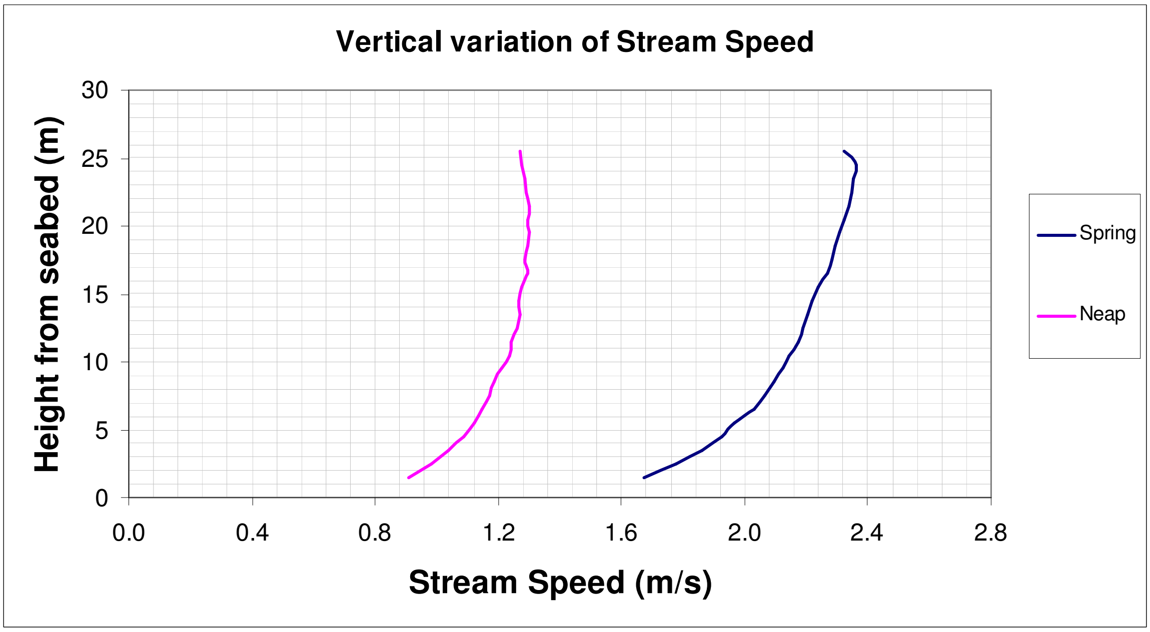

Figure 6.2 Variation of stream speed with height for Cullivoe Ness

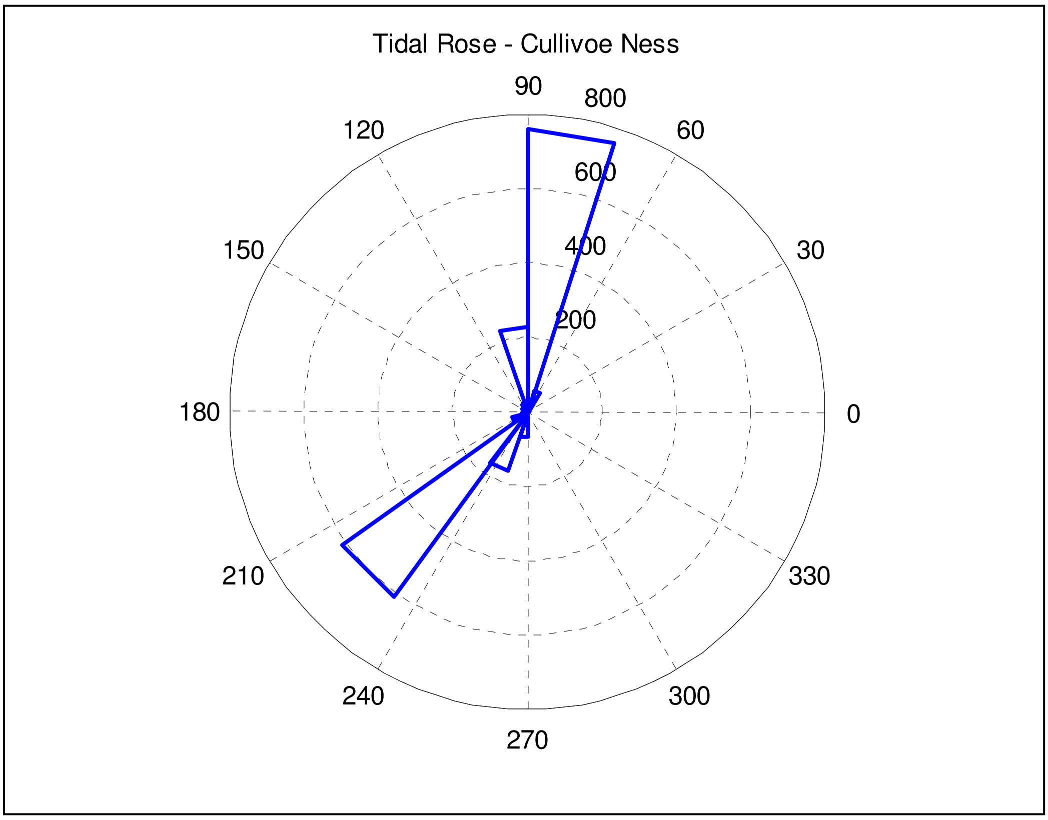

Figure 6.3 Tidal rose indicating directions of stream speed for Cullivoe Ness

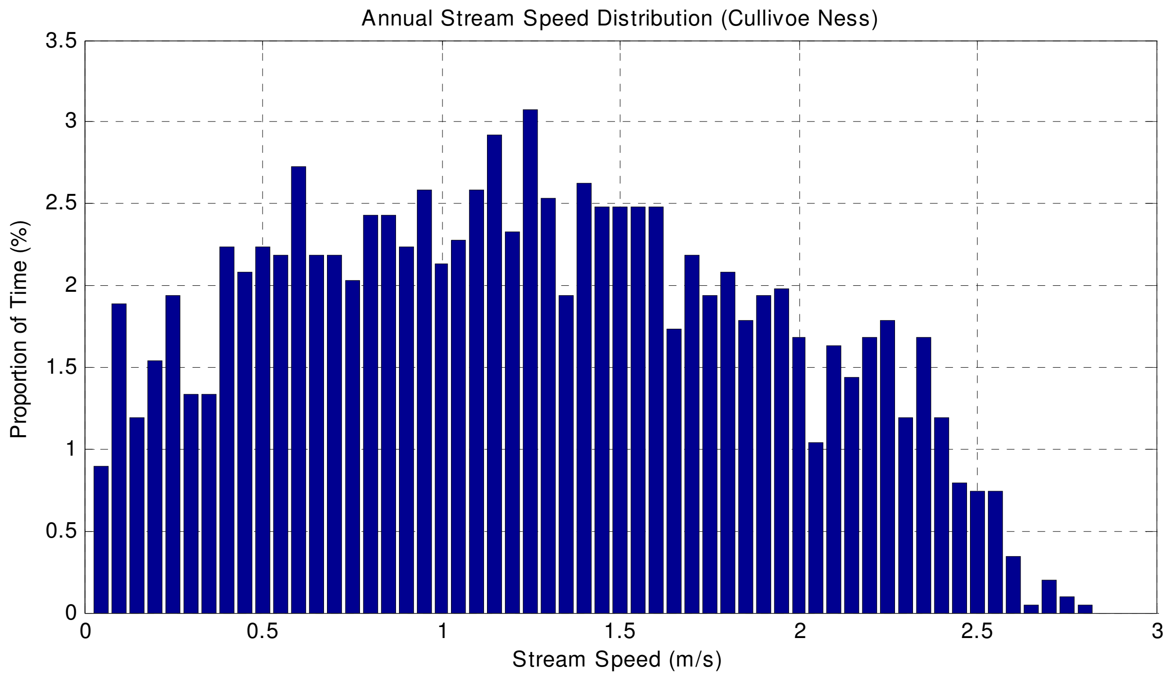

Figure 6.4 Annual stream speed distribution for Cullivoe Ness

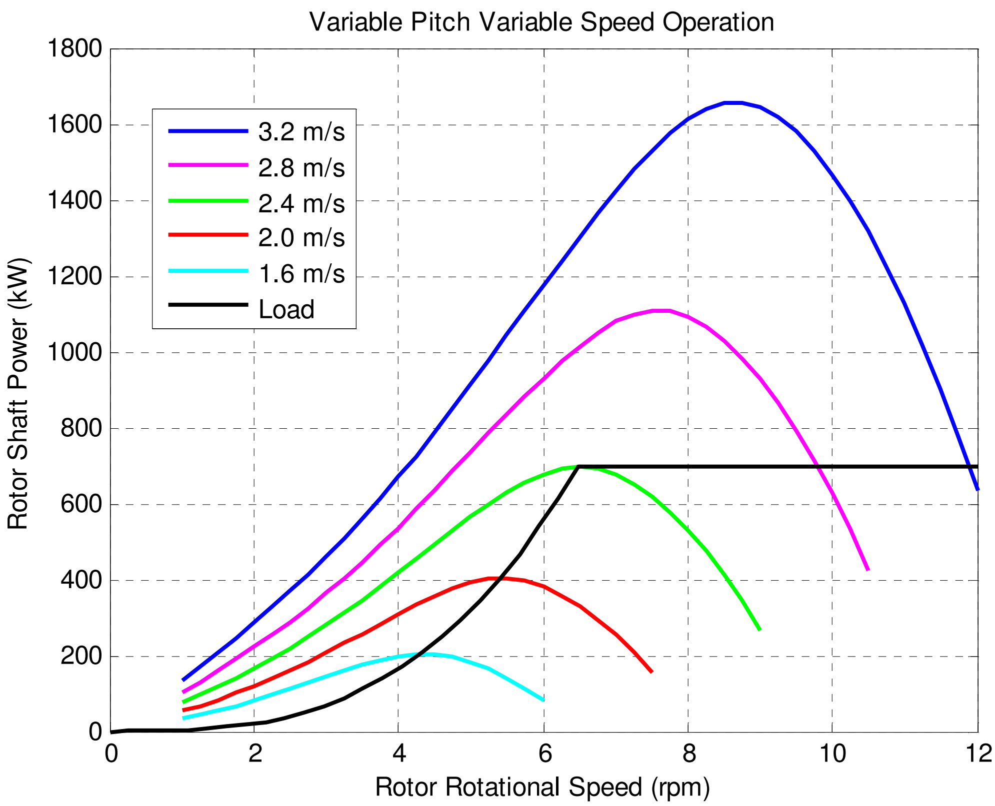

Figure 6.5 Shaft power in variable speed operation

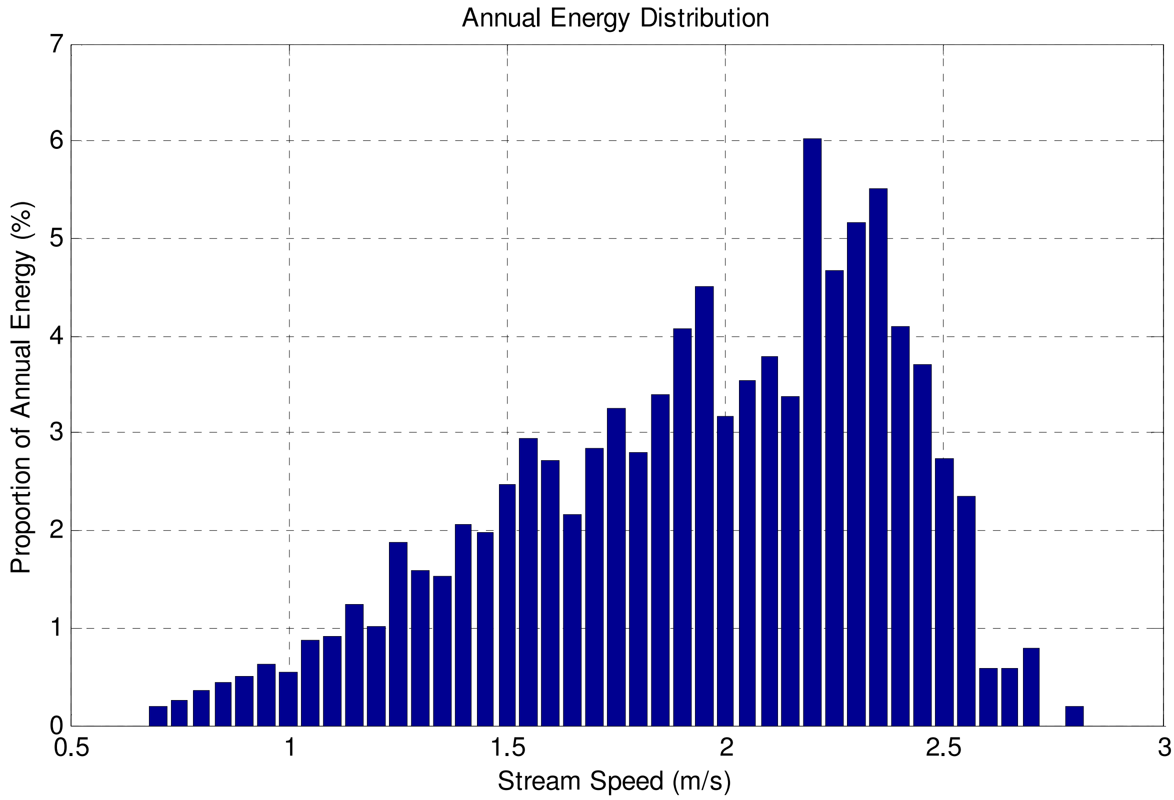

Figure 6.6 Annual energy distribution

7 CHAPTER SEVEN – UNIT COST ANALYSIS

7.1 Introduction

The objectives of this cost analysis are as follows:

- Estimation of site specific annual energy production of machine(s)
- Costing of components of a baseline design
- Calculation of the unit cost of electricity
- Comparison of the unit cost of electricity of this machine to that for a horizontal axis turbine design for the same baseline site
- Determination of the effect of certain factors on the unit costs of electricity

The calculations were made on the basis of small-scale manufacture of single units, at two power ratings:

- 45kW, which corresponds to the size of a small-scale prototype unit.
- 500kW, corresponding to the size of a commercial production unit.

7.2 Energy Output Estimates

The energy output from a unit depends principally on the annual stream speed distribution of the site, the rotor swept area, the rotor efficiency and rated velocity of the machine. Three typical sites representing low, medium and high current sites were selected for the energy estimate. Knowing the depth of the site under consideration the height of the rotor blade is selected. The rotor diameter as well as the rated speed is then altered until the required rated power is obtained taking account of the typical average rotor efficiency. At this stage attention is also paid to the changes in the requirement of the drive train.

The selection of the rated speed of the unit is critical to the economic performance of the machine. This is because if it is too high the added cost of the drive train may not be justified by the energy gained and if it's too low the cost of the energy lost may be higher than the savings made for a simpler drive train.

An energy model set up in Matlab was created to aid the analysis, as shown in Chapter 6. The model outputs a statistical summary of tidal data (time and height dependent stream speed and direction) from a site. These include the annual velocity distribution of the site and a tidal rose. Using as an input the turbine specification (global efficiency, rotor swept area, rated speed) and depth of deployment, the annual energy and distribution is computed. The captured annual energy figure serves as input for the discount cash flow (dcf) model.

7.3 Cost of Real Machine

Using the baseline design outlined in Table 7.1 and Table 7.2 as a reference, quotes were obtained for major components including the pitch motor and gearbox and generator. For the parts to be fabricated the weight, material type and size were used to estimate their costs and are shown in

Table 7.3. The cost of the mooring system was provided by Balmoral Marine. Estimates for other costs including survey of sites and permission, insurance and seabed rent were taken from Binnie Black and Veatch (2001). The operation and maintenance cost was set at three percent of the capital cost. These were all set up in an excel spreadsheet linked to the dcf to be described in the next section.

7.4 Unit Cost of Electricity

The unit cost of electricity is computed by a discounted cash flow method. This approach makes allowance for the distribution of expenditure during the construction and decommissioning phase of the project. Energy production starts after installation (after year 1) and ends just before decommissioning. The life of the unit is pegged at 25 years. Decommissioning cost is taken as five percent of the capital cost. Knowing the capital and annual cost as well as the annual energy production of the machine the unit cost of electricity is evaluated at a known discount rate. A range of costs is presented for discount rates between 5 and 15%, with a reference discount rate of 8%.

The cost analysis was performed for both the baseline and Cullivoe Ness sites, and leads to the following observations:

- The introduction of variable pitch has a significant impact on the unit cost for M.S peak stream velocities considered (2.3 to 3m/s). This is shown in Table 7.5 and
- Table 7.6.
- The value of the performance coefficient Cp has also a strong effect on the unit cost, as shown in Table 7.7
- The unit costs calculated for the 45kW and 500kW machines are in line with that predicted for variable-pitch, optimised horizontal axis tidal turbines.

7.5 Comparison with the MCT Horizontal Axis Design

The variable pitch vertical axis device was compared with the MCT horizontal axis tidal turbine in the case of the baseline tidal site, for which sufficient data were available for both machines. The rated power was 500kW for the Edinburgh Designs machine, and 1MW (2 x 500kW) for the MCT device.

Table 7.4 shows the detail of capital and operational costs for both machines, while Table 7.8 shows the unit cost comparison at different discount rates.

The unit costs are very similar for the two machines, but it is interesting to note that the vertical axis turbine achieves its unit cost figure at half the rated power of the MCT turbine. Additionally, at depths less than 25m, the unit costs calculated for the horizontal axis design increase significantly due to lower rated power imposed by the smaller rotors. In contrast, the vertical axis configuration can be operated at depths significantly below 25m without having to scale down its energy capture cross-section and with it its rated power. At shallow sites, this should result in an economic advantage.

More generally, the results obtained here indicate that the minimum rated power for economical operation is lower for the variable pitch VATT than it is for the MCT device. This suggests the VATT, while lending itself well to large tidal farm operation, could be an attractive alternative for remote sites with smaller grid capacity and limited maintenance resources.

7.6 Tables

Table 7.1

Baseline Design Assumptions			
Variable	**Unit**	**45 kW**	**500 kW**
Peak stream speed	m/s	3	3
Depth at low tide	m	30	30
Tidal range	m	5	5
Distance to shore station	m	3000	3000
Rotor tip speed	m/s	5.11	5.19
Average power coefficient (rotor eff)		0.45	0.45
Cut-in stream speed	m/s	0.7	0.7
Annual operation and maintenance costs	% of capital cost	3	3
Gearbox efficiency	%	0.9	0.9
Generator efficiency	%	0.94	0.94
Reliability	%	0.95	0.95

Table 7.2

Key Technical Characteristics of Baseline Designs			
Characteristic	**Unit**	**45kW**	**500 kW**
Rotor Diameter	m	5	16
Blade height	m	4	12
Rotor Swept Area	m^2	20	192
Blade number		3	3
Blade chord	m	0.375	1.2
Rated rotor rotational speed	rpm	19.5	6.2
Rated mechanical power output	kW	53	591
Rated electrical power output	kW	45	500
Rated stream speed	m/s	2.3	2.4
Annual energy Output (variable pitch)	MWhr	117.47	1122.4
Capacity factor (variable pitch)		0.31	0.25
Annual energy Output (fixed pitch)	MWhr	91.6	873

Table 7.3

Unit Costs for Steel Fabrications	
Type of fabrication	**Unit cost (£/kg)**
Pipes, tubes	1.2
Light fabrications, walkways	1.4
Basic machined items, flanged pipes	2
Complex machined items	2.75

Table 7.4: Cost Estimates for 45 and 500kW VATT and 1MW MCT Scheme

COST ESTIMATE				
		ED VATT		**MCT HATT**
		45 kW	**500 kW**	**2 x 500 kW**
CAPITAL COSTS				
Power Train				
Rotor blades	£'000	6.0	18	
Rotor ring	£'000	2.0	14	
Shaft and bearing	£'000	2.0	10	
Hub and Spokes	£'000	4.0	12	
Gearbox	£'000	2.0	20	
Bearing	£'000	2.0	6	
Slip rings	£'000	0.5	1	
Generator	£'000	2.0	10	
Controller (variable speed drive, carbinet)	£'000	8.0	18	
Coupling	£'000	1.5	3	
Sub-total	**£'000**	**30.0**	**112.0**	**288.1**
Support Structure				
Floating Vessel	£'000	2.9	30.1	0
Moorings/Mono-pile	£'000	32.0	71.1	261.2
Sub-total	**£'000**	**34.9**	**101.2**	**261.2**
Pitch Control Mechanism				
Pitch Motor	£'000	2.4	9	0
Amplifier and control cards	£'000	3.0	6	0
Bearing	£'000	1.2	2.4	0
Sub-total	**£'000**	**6.6**	**17.4**	**0**
Additional Offshore Items				

Offshore electrical equipment	£'000	0.0	20	138
5MW substations	£'000	0.0	0	0
30MW substations	£'000	0.0	0	0
Support piles	£'000	0.0	0	0
Omitted items	£'000	0.0	9.2	13.8
Sub-total	**£'000**	**0**	**29.2**	**151.8**
Installation				
Assembly and transport	£'000	2.0	17	26.2
Mooring/Mono-pile	£'000	0.0	100	688.8
Submarine Cabling	£'000		326.11	489.2
Sub-total	**£'000**	**2.0**	**443.1**	**1204.2**
Onshore Items				
RPA Substation	£'000	0.0	25	25
Spares	£'000	5.0	15	21.3
Sub-total	**£'000**	**5.0**	**40.0**	**46.3**
Overhead Items				
Surveys and permission	£'000	9.0	92.1	92.1
Design (power, electrical plant)	£'000	8.0	10	21.3
Management and profit	£'000	10.0	60	97.6
Sub-total	**£'000**	**41.0**	**162.1**	**211.0**
TOTAL CAPITAL COST	**£'000**	**106.3**	**905.01**	**2162.7**
ANNUAL COST				
Insurance	£'000	2.0	15.5	15.5
Seabed rent	£'000	1.0	2	2
Operation and maintenance	£'000	3.2	27.15	43.0
TOTAL ANNUAL COST	**£'000**	**6.2**	**44.7**	**60.5**

Table 7.5: Comparison of Unit Costs between Fixed and Variable Pitch Designs for a 500kW Design

ED (500kW)	Unit costs of energy (p/kWh)			
Discount rate	5%	10%	15%	8%
Fixed Pitch, baseline site	9.1	11.9	15.2	10.7
Variable Pitch, baseline site	7.2	9.4	12.1	8.5
Variable Pitch, Cullivoe Ness site	10.3	13.6	17.3	12.2

Table 7.6: Comparison of Unit Costs between Fixed and Variable Pitch Designs for a 45kW Design

ED (45kW)	Unit costs of energy (p/kWh)			
Discount rate	5%	10%	15%	8%
Fixed Pitch, Cullivoe Ness Site	16.7	21.7	27.4	19.6
Variable Pitch, Cullivoe Ness Site	14.3	18.7	23.7	16.8

Table 7.7: Variation of Unit Cost against Cp for Variable Pitch Design

ED (500kW)	Unit costs of energy (p/kWh) at 8% discount rate			
Av rotor Cp	0.35	0.4	0.45	0.5
Variable Pitch, Baseline Site	10.9	9.5	8.5	7.6

Table 7.8: Comparison of Unit costs between Vertical Axis (ED) and Horizontal Axis (MCT) Designs

	Unit costs of energy (p/kWh)			
Discount rate	5%	10%	15%	8%
MCT (1MW), Baseline Site	7.3	9.69	12.75	8.63
ED (500kW), Baseline Site	7.2	9.4	12.1	8.5