POTENTIAL APPLICATIONS FOR FLETTNER ROTORS AND TURBOSAILS IN TIDAL STREAM TURBINES

CONTRACT NUMBER: T-06-00217/00/00

URN NUMBER: 05/636

dti

The DTI drives our ambition of ‘prosperity for all’ by working to create the best environment for business success in the UK. We help people and companies become more productive by promoting enterprise, innovation and creativity.

We champion UK business at home and abroad. We invest heavily in world-class science and technology. We protect the rights of working people and consumers. And we stand up for fair and open markets in the UK, Europe and the world.

POTENTIAL APPLICATIONS FOR FLETTNER ROTORS AND TURBOSAILS IN TIDAL STREAM TURBINES

REPORT NUMBER
T-06-00217-00-REP

Contractor

Oreada

First published 2005

EXECUTIVE SUMMARY

Introduction

Tidal stream energy is now recognised as a potentially extremely valuable source of renewable energy. However, a great deal of technology development is required before commercial exploitation of this resource becomes a reality. In particular, some of the existing prototype systems are based on technology derived from the wind energy industry, and it is by no means clear that this will provide the optimum configuration for the exploitation of tidal currents.

With this in mind, Oreada have been investigating two novel lifting devices that can be used to power tidal stream generators, they are Flettner Rotors and Turbosails™ (a trademark of the Cousteau Society). This report presents the details of our work to date with regard to the potential application of these devices.

We would like to thank the DTI (New and Renewable Program) for funding 75% of the costs of this work. Our DTI project reference is T/06/00217/00/00.

Project Aims and Objectives

The fundamental aim of this project is to determine if either the Flettner rotor or the Turbosail™ are suitable lifting devices for use on a vertical axis tidal stream turbine. To accomplish this a mathematical model has been developed to simulate a vertical axis turbine. This model has been used to calculate and compare the power generated by the different lift devices, and a conventional hydrofoil.

The overall project is divided into two stages. The first stage is to develop the numerical model and, if appropriate, to progress with phase 2 that involves scale model tests in a laboratory. This report presents the results of Phase 1.

Summary of Methodology Adopted

The mathematical model has been developed using Matlab and is based on theory and experiments from the study of vertical axis wind turbines. The mathematical model assumes that the base case configuration for the turbine will be four parallel lift devices located at a constant radius from the centre of the turbine. Adjacent lift devices will therefore subtend an angle of 90° at the centre of the turbine.

The model has been used to compare Flettner rotors and Turbosails™ with conventional hydrofoil sections to see if there is a likely net benefit from using such devices.

Conclusions and Recommendations

Vertical and horizontal axis machines have been compared, and vertical axis machines clearly provide increased flexibility when the flow is constrained top and bottom (i.e. shallow seas) and they give added control of torque and angular frequency compared with horizontal axis machines.

Machines powered by Flettner rotors or Turbosails™ can, for certain configurations, produce larger driving forces than conventional hydrofoils. The peak value, with respect to solidity, of the mean power output corresponds to a higher optimum solidity for the hydrofoil powered machine compared to the Flettner rotor and Turbosail™ machines. The peak in the mean power is also dependent on the tip speed ratio and occurs at slightly higher values of tip speed ratio for the hydrofoil powered machine, but remains below the cavitation limit. The result of these characteristics is approximately equal peak values of mean power output but with a smaller corresponding torque for the hydrofoil case.

The Flettner rotor or Turbosail™ powered machine is more sensitive to drag because of the lower lift to drag ratio when compared to the hydrofoil powered machine. The model currently underestimates drag because no allowance is made for the supporting structure. There is also uncertainty regarding the behaviour of the lift coefficient for Flettner rotors operating in an unsteady manner. It is almost certain that these two points will combine to significantly lower the performance of the Flettner rotor powered machine, and may even prevent the machine from operating.

After considering these results we believe that there is not a clear case in support of either Flettner rotors or Turbosails™ in application as vertical axis tidal stream turbines, and as such we do not propose to continue to phase 2 of this project.

CONTENTS

LIST OF FIGURES

LIST OF TABLES

1 INTRODUCTION

1.1 Background

Tidal stream energy is now recognised as a potentially extremely valuable source of renewable energy. However, a great deal of technology development is required before commercial exploitation of this resource becomes a reality. In particular, some of the existing prototype systems are based on technology derived from the wind energy industry, and it is by no means clear that this will provide the optimum configuration for the exploitation of tidal currents.

With this in mind, Oreada have been investigating two novel lifting devices that can be used to power tidal stream generators, they are Flettner Rotors and Turbosails™ (a trademark of the Cousteau Society). This report presents the details of our work to date with regard to the potential application of these devices.

We would like to thank the DTI (New and Renewable Program) for funding 75% of the costs of this work. Our DTI project reference is T/06/00217/00/00.

1.2 Aim and Objectives of the Project

The fundamental aim of this project is to determine if either the Flettner rotor or the Turbosail™ are suitable lifting devices for use on a vertical axis tidal stream turbine. To accomplish this a mathematical model has been developed to simulate a vertical axis turbine. This model has been used to calculate and compare the power generated by the different lift devices, and a conventional hydrofoil.

The overall project is divided into two stages. The first stage is to develop the numerical model and, if appropriate, to progress with phase 2 that involves scale model tests in a laboratory. This report presents the results of Phase 1.

2 TECHNICAL BACKGROUND TO THE PROJECT

2.1 Operating Principles of the Flettner Rotor and Turbosail™

The Flettner rotor is shown below in operation as ship "sails".

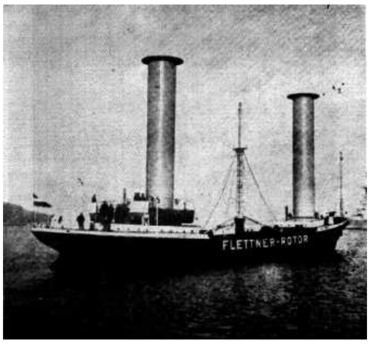


Figure 2-1 Flettner's full-size Rotorship "Bruckau"

These devices generate lift by creating circulation in the adjacent fluid boundary layer. A conventional aerofoil creates vortices and circulation by having an asymmetric shape and by careful orientation to the flow ("angle of attack"). However, the Flettner rotor and Turbosail™ create circulation by performing mechanical work and thereby putting energy into the fluid. The aim is to put a small amount of energy into the fluid in order to extract a large amount of energy from the flow stream. The Flettner rotor is a rotating cylinder and the Turbosail™ (a similar shaped structure) sucks the driving fluid through holes in one side to produce lift. The following "end on" schematics show how the wake behind the cylinders are deflected, as a result of the work done, by altering the point of normal flow separation which introduces a lift component of force.

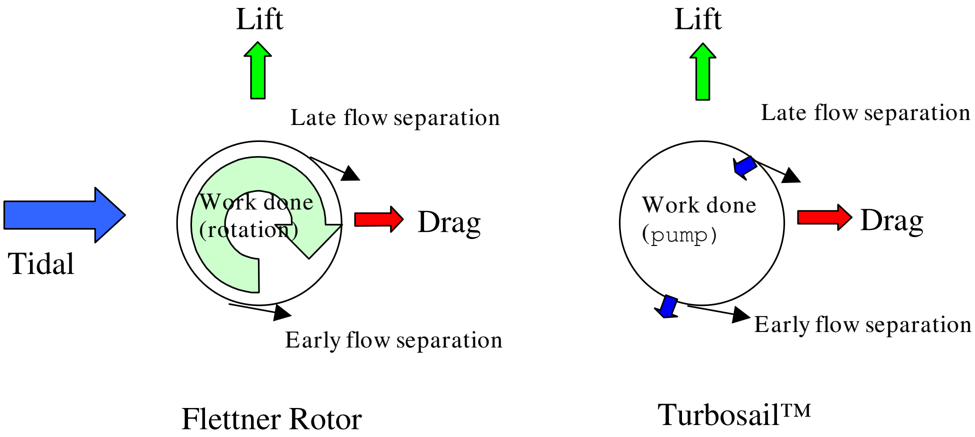


Figure 2-2 Lift creation using Flettner Rotor or Turbosail™

Generating lift in this way produces very high lift coefficients of 8 or more, but the drag is also high (C_D of the order 1).

2.2 Vertical versus Horizontal Axis Machines

The designation of a machine as either a horizontal or vertical axis machine refers to the orientation of the axis of rotation. A horizontal axis machine is designed to work when its axis of rotation is parallel to the tidal stream. Conversely, a vertical axis machine is designed to work when its axis of rotation is perpendicular to the tidal stream (and vertical as the name implies).

Vertical axis turbines have essentially been rejected by the wind industry for several valid reasons. However, tidal streams are very different to wind and the same logic does not apply in all cases.

Why are conventional wind turbines horizontal axis?

(i) The aerofoils have a very efficient lift/drag ratio – more power per square metre.
(ii) They are relatively easily mounted high off the ground where the wind is stronger.
(iii) They don't suffer from high fatigue loads caused by uneven torque associated with vertical axis machines.

The main differences subsea are the two boundaries, the sea surface and the seabed, and the effects of buoyancy. The distance between the two boundaries (water depth) is a major design concern while buoyancy is a potential benefit.

If tidal stream turbines are to succeed on an industrial scale then we must consider the lessons being learnt in the offshore wind industry – availability is a key factor. High availability (circa. 95%) can only be achieved by using very reliable equipment with built in redundancy and an active maintenance and repair programme. Offshore operations are costly especially subsea, and therefore the machines will need to be very powerful to recover the cost. Because of this 5MW offshore wind turbines are planned and it is envisaged that tidal stream turbines will need to go the same way – or even bigger. A 5MW horizontal axis tidal stream turbine will need a diameter of about 40m (assuming 2m/s peak current and 50% overall efficiency). Wave-induced currents of the order 2m/s often reach 40m below the sea surface. Therefore, industrial scale horizontal axis machines may be limited to water depths of 85m or more to allow seabed clearance.

One advantage of vertical axis turbines over horizontal axis machines is that they can cater for the fact that the current can vary both in strength and direction with time and depth. The direction variations are automatically catered for by a vertical axis machine (using hydrofoils), whereas on a horizontal axis machine it is necessarily the case that when flow direction varies with depth the flow over part of the turbine will be non-axial. The variation in speed can be catered for by making the blades (or rotors) of the turbine inclined to the vertical and/or of varying cross-section.

It is possible that the Turbosail™ can be designed to have a higher efficiency in a wave environment. For example the top of the Turbsail could have water being sucked in through holes on the left side while the bottom could be sucking in through holes on the right side (and varied between). This would ensure that the lift generated by the wave at the top of the sail (which may reverse the flow past the cylinder) would work with the lift generated by the tide at the bottom of the cylinder rather than opposing. At the very least the holes in the top part of the cylinder could be shut off when the wave-induced velocity is in the wrong direction thereby eliminating an opposing lift force. This cannot be done with a horizontal axis machine.

Although the flow orientation may be constant (with reversal of direction) in confined waters, in the open ocean the flow orientation may rotate throughout the cycle. Therefore a fixed-direction horizontal axis machine would only be of use in confined waters, whereas a vertical axis machine could operate in any location including deep water.

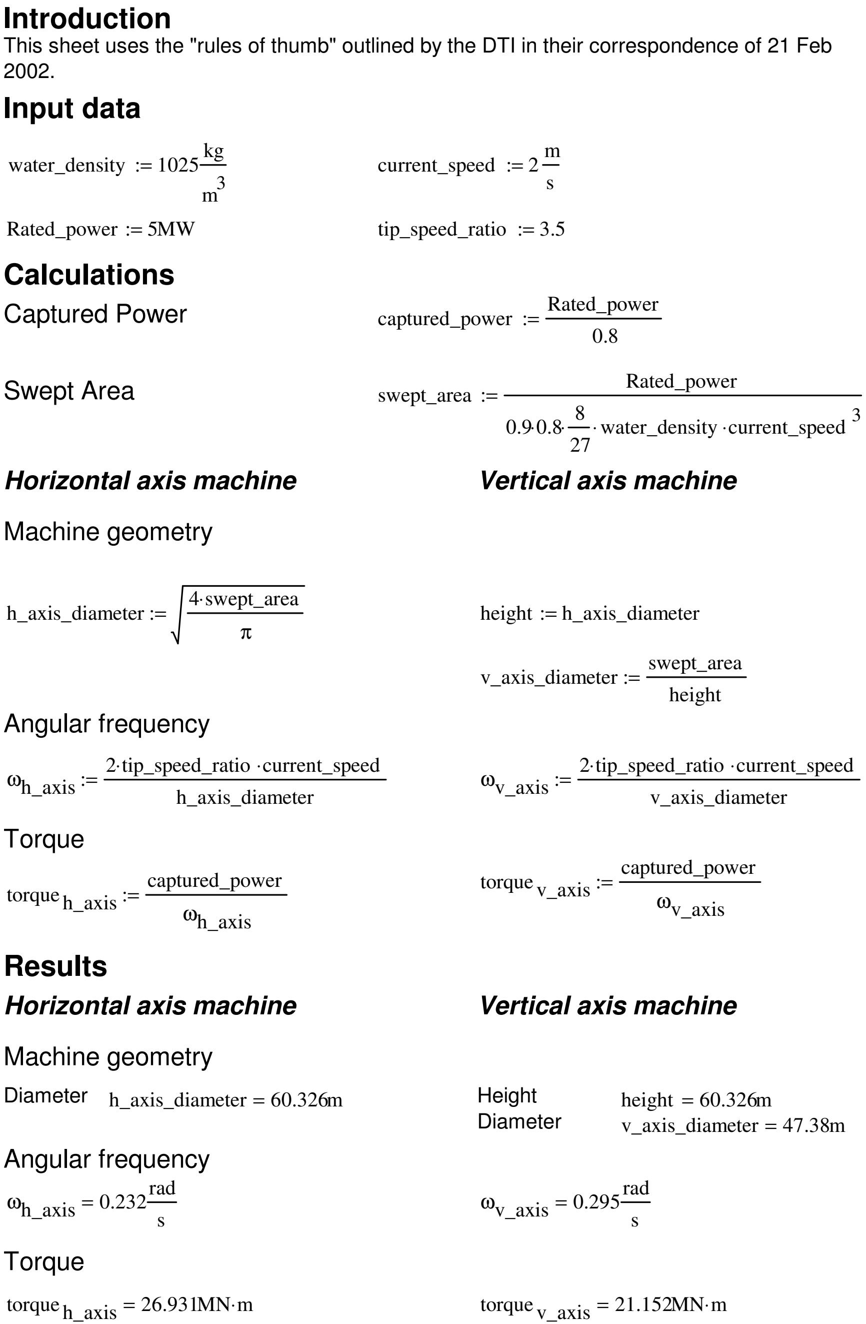
The maximum amount of power that a machine can exploit when operating at its maximum efficiency is controlled by the swept area of the blades. Increasing the swept area increases the total energy flux through the machine. With a horizontal axis machine the diameter is the only dimension that can be increased to enlarge the swept area. This means that any increase in the swept area must be coupled with increases in shaft torque and tip speed ratio for a constant angular speed. Both of these consequences have negative implications for the operation of the horizontal axis machine. High torque machines have poor electrical power generation efficiencies and high blade tip velocities can induce cavitation, leading to damage of the blades.

The swept area of a vertical axis machine can be increased without altering the diameter because the axis of rotation is perpendicular to the flow. Unlike a horizontal axis machine, where the swept area is explicitly tied to a unique diameter, a vertical axis machine can be configured to have a large swept area with a relatively small diameter by making it tall. Increasing the height of a vertical axis machine will still produce a higher torque for a constant angular frequency but the tip speed ratio will not increase. If the diameter is decreased and the height is increased to maintain a constant swept area, the torque can be reduced and the angular frequency increased for a constant power output. This allows the vertical axis machine to be optimised to suit the power generation efficiency curve. The swept area for a given diameter is however limited by structural constraints on the slenderness of the machine.

Vertical axis machines also have the advantage that the swept area can be increased when the fluid flow is confined above and below the machine by the water surface and sea bed as it is in relatively shallow water. A horizontal axis machine will have a limiting diameter in shallow water to avoid breaching the surface or impacting on the sea bed.

These points are illustrated in Figure 2-3. Using “rules of thumb” the diameter required to produce 5MW using a horizontal axis machine is 60m. If a vertical axis machine is to be used in the same depth of flow it would have a height of 60m resulting in a diameter of only 47m for the same power output (neglecting any extra losses due to the supporting arms in the vertical axis case). This gives the vertical axis case a higher angular frequency and lower torque, which may improve electrical generation efficiency. Alternatively, a vertical axis machine with the same diameter and therefore the same torque and angular frequency could operate in 13m less of water depth compared to the horizontal axis case. This flexibility could be beneficial if it allowed a device to be positioned in a location with high tidal velocities.

One final advantage of the proposed cylindrical devices is that they can utilise controlled buoyancy to reduce installation and maintenance costs.



Introduction

This sheet uses the "rules of thumb" outlined by the DTI in their correspondence of 21 Feb 2002.

Input data

$\text{water\_density} := 1025 \frac{\text{kg}}{\text{m}^3}$

$\text{current\_speed} := 2 \frac{\text{m}}{\text{s}}$

$\text{Rated\_power} := 5\text{MW}$

$\text{tip\_speed\_ratio} := 3.5$

Calculations

Captured Power

$$\text{captured\_power} := \frac{\text{Rated\_power}}{0.8}$$

Swept Area

$$\text{swept\_area} := \frac{\text{Rated\_power}}{0.9 \cdot 0.8 \cdot \frac{8}{27} \cdot \text{water\_density} \cdot \text{current\_speed}^3}$$

Horizontal axis machine

Machine geometry

$$\text{h\_axis\_diameter} := \sqrt{\frac{4 \cdot \text{swept\_area}}{\pi}}$$

Angular frequency

$$\omega_{\text{h\_axis}} := \frac{2 \cdot \text{tip\_speed\_ratio} \cdot \text{current\_speed}}{\text{h\_axis\_diameter}}$$

Torque

$$\text{torque}_{\text{h\_axis}} := \frac{\text{captured\_power}}{\omega_{\text{h\_axis}}}$$

Vertical axis machine

$$\text{height} := \text{h\_axis\_diameter}$$

$$\text{v\_axis\_diameter} := \frac{\text{swept\_area}}{\text{height}}$$

$$\omega_{\text{v\_axis}} := \frac{2 \cdot \text{tip\_speed\_ratio} \cdot \text{current\_speed}}{\text{v\_axis\_diameter}}$$

$$\text{torque}_{\text{v\_axis}} := \frac{\text{captured\_power}}{\omega_{\text{v\_axis}}}$$

Results

Horizontal axis machine

Machine geometry

Diameter $\text{h\_axis\_diameter} = 60.326\text{m}$

Angular frequency

$\omega_{\text{h\_axis}} = 0.232 \frac{\text{rad}}{\text{s}}$

Torque

$\text{torque}_{\text{h\_axis}} = 26.931\text{MN} \cdot \text{m}$

Vertical axis machine

Height $\text{height} = 60.326\text{m}$

Diameter $\text{v\_axis\_diameter} = 47.38\text{m}$

$\omega_{\text{v\_axis}} = 0.295 \frac{\text{rad}}{\text{s}}$

$\text{torque}_{\text{v\_axis}} = 21.152\text{MN} \cdot \text{m}$

Figure 2-3 Numerical comparison of vertical and horizontal axis machines using "rules of thumb"

2.3 Maximum Power versus Overall Efficiency

The overall efficiency of the device depends on how much of the energy passing through the turbine can be captured. Ideally this should be measured over the whole tidal cycle, not just at the point of maximum flow. A device with a high power coefficient at maximum flow but a high cut-in speed may not necessarily produce more power than a device with a lower maximum power coefficient but also a lower cut-in speed. The best device would have a broad curve of power versus flow speed, not a sharply peaked curve.

Rated output is the power provided by the turbine at the rated speed. It is not the average power, and there is no simple formula to calculate the cumulative power in kWh from the rated power in kW. The rated power is never an entirely satisfactory parameter to use for comparison of different turbines, which ought to be compared according to the cumulative energy produced over a long time period. For a tidal machine, the output over a complete tidal cycle would be a better measure of the machine's efficiency than its performance at the point of maximum flow. The performance at low flow speeds can be more important than at the maximum flow speed, since the maximum speed only occurs for a small proportion of the cycle.

However, this study will use rated power to compare the different turbines as comparing the output over a complete tidal cycle would require modelling of the transient behaviour of the machine. The model in this study cannot model transient behaviour as the calculated power output presently depends upon a specified constant angular frequency for the machine.

3 CONVENTIONAL HYDROFOIL VERSUS PROPOSED DEVICES

The previous section identified several advantages of vertical axis turbines for marine applications. This section considers the performance of a vertical axis machine when driven by the proposed devices and compares them with a conventional hydrofoil. The mathematical model developed for this project includes three different types of lift device and these are listed below along with the key variables:

Hydrofoil – shape, chord length, angle of attack,
Flettner rotor – rotor diameter, rotor angular frequency,
Turbosail™ – shape, chord length, angle of attack, suction velocity.

Of these key variables, the angle of attack, rotor angular frequency and suction velocity can all be varied during a rotation to maximise the favourable tangential force. The behaviour of the Flettner rotor and Turbosail™ are similar in the respect that both can generate large lift forces with the drawback in both devices being the correspondingly large drag force.

The turbine geometry for each lift device is calculated on the basis of equal turbine solidity. From the study of horizontal axis wind turbines solidity is defined as

$$s = \frac{nL}{2\pi r}$$

where *s* is solidity, *n* is the number of lift devices, *r* is the turbine radius and *L* is a characteristic length for the lift device. For the Flettner rotor the characteristic length *L* is the diameter, but for the hydrofoil and the Turbosail™ *L* is the chord length (Since the Turbosail™ is not completely cylindrical).

3.1 Lift and Drag Coefficients

The lift and drag coefficients for each type of lift device is presented in the following figures.

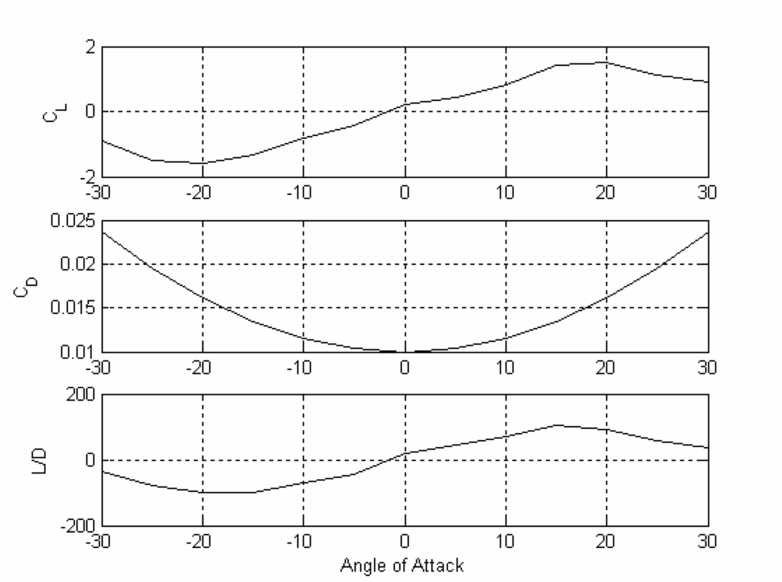


Figure 3-1 Drag and lift coefficients for symmetrical hydrofoil [3]

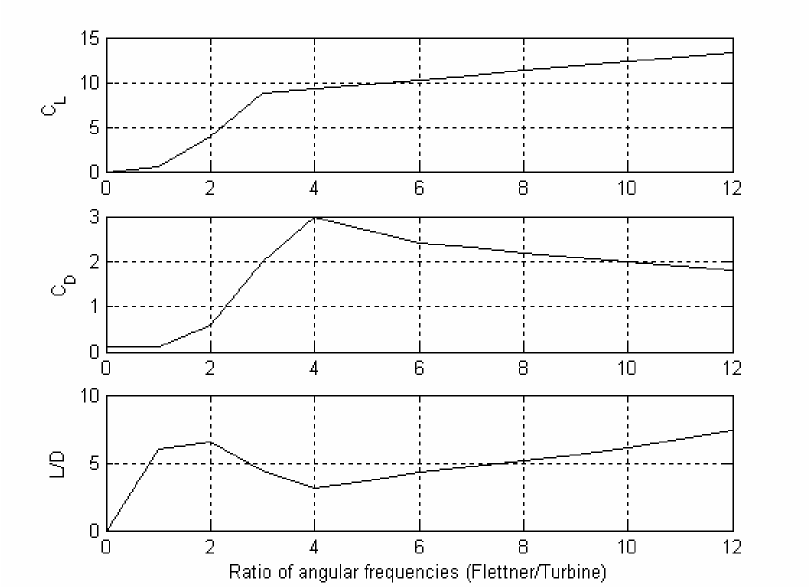


Figure 3-2 Drag and lift coefficients for Flettner rotor [2]

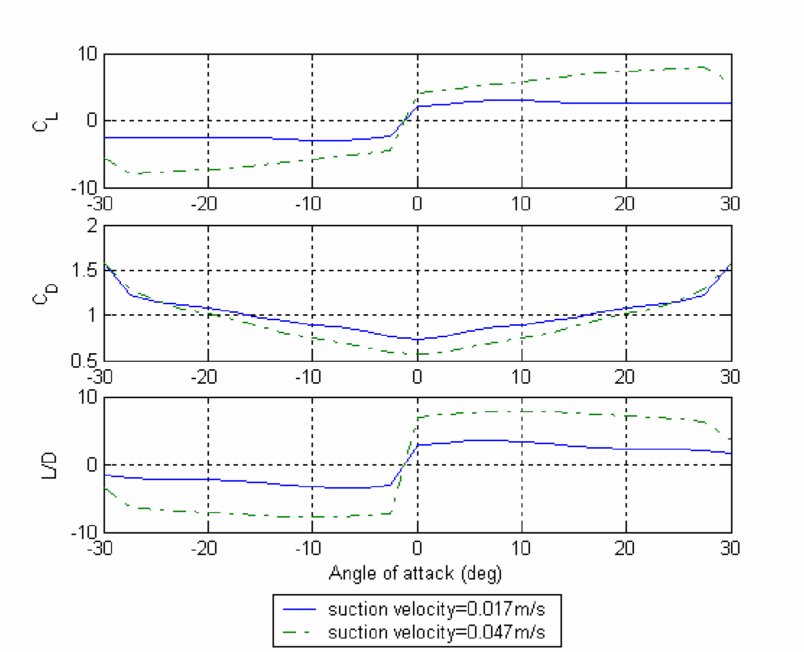


Figure 3-3 Drag and lift coefficients for Turbosail™ [1]

The hydrofoil produces the highest ratios of lift over drag coefficients by an order of magnitude.

3.2 Power and Torque

The higher lift over drag ratio of the hydrofoil does not necessarily mean it will produce more torque than either the Flettner rotor or Turbosail™. This is because both the Flettner rotor and Turbosail™ are capable of producing larger lift coefficients than the hydrofoil and therefore have the potential to create a larger tangential force. The Flettner rotor is the device that will be compared to the hydrofoil from this point onwards (because the Turbosail™ has similar lift and drag characteristics). The following example demonstrates that the Flettner rotor machine can produce a larger power output than a hydrofoil powered machine of equal solidity (the same is true for the Turbosail™). This is because the driving tangential force is larger for the Flettner rotor machine due to its large lift coefficient.

This is demonstrated in the following example.

Input data	
Current velocity	2 ms^{-1}
Turbine radius	25 m
Turbine height	50 m
Tip speed ratio	3
Solidity	0.1
Hydrofoil data	
Chord length	3.93 m
Pitch angle (relative to tangent to hydrofoil path)	5° with sinusoidal variation
Flettner rotor data	

Rotor diameter	3.93 m
Maximum rotor spin to tip speed ratio	2 with sinusoidal variation

Table 1 Example calculation data

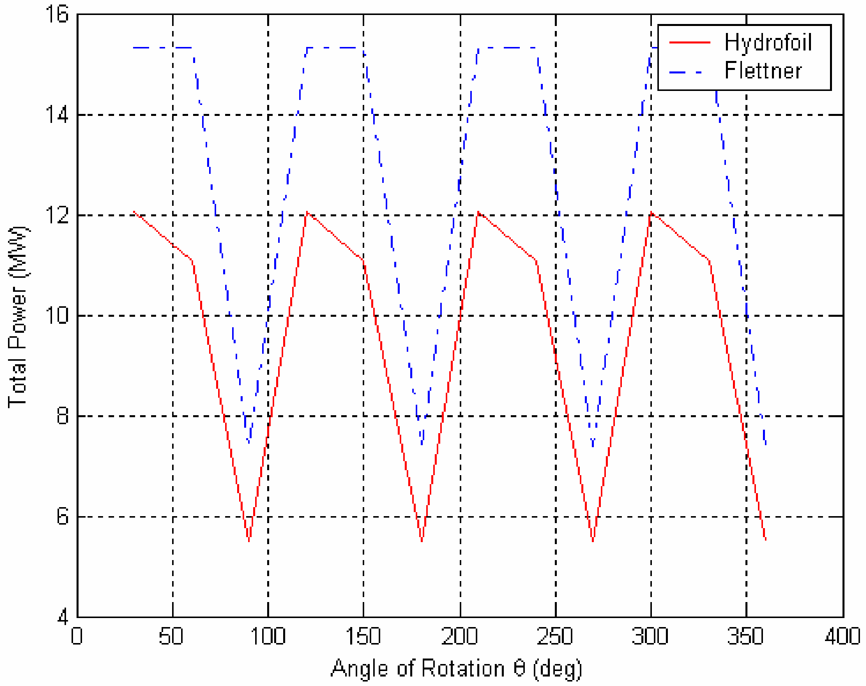


Figure 3-4 Power output for hydrofoil and Flettner rotor data in Table 1

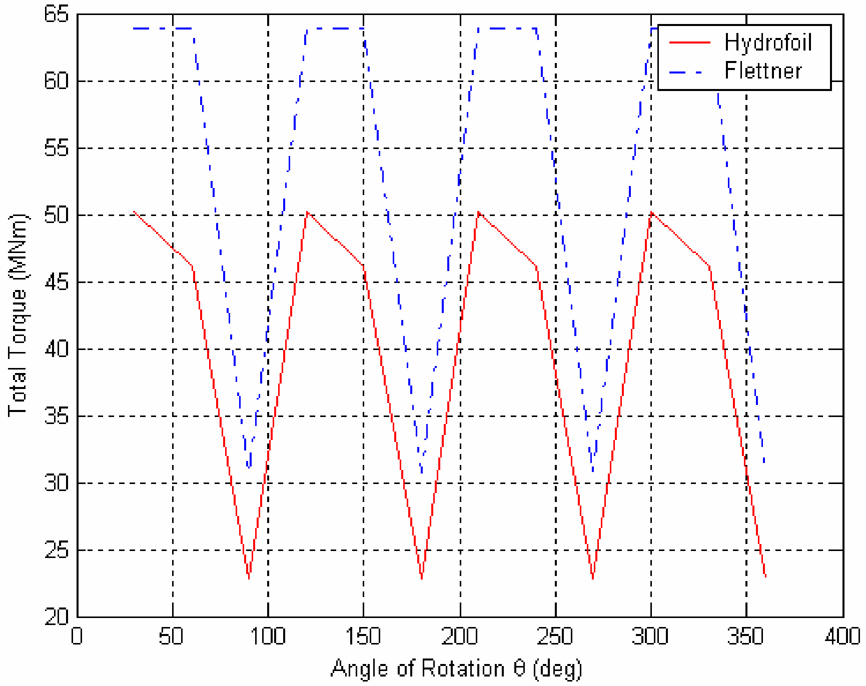


Figure 3-5 Torque for hydrofoil and Flettner rotor data in Table 1

For the example calculation in Table 1 the Flettner rotor powered machine produces a mean power output over one turbine rotation of 12.68MW whilst the hydrofoil powered machine only generates a mean power of 9.5MW. The larger power output at a solidity of 0.1 does not necessarily mean that the Flettner rotor is more suitable than a hydrofoil for powering the machine. The lower drag of the hydrofoil powered machine means that the optimum power output occurs at a larger solidity than the Flettner rotor powered machine. If the solidity of the Flettner rotor machine is increased above 0.1 the power output of the machine drops. However, the mean power output (over one turbine rotation) of the hydrofoil powered machine increases to a peak value (with respect to solidity) of 12.3 MW at a solidity of 0.15. This can be seen from Table 2.

Solidity	Tip speed ratio	Hydrofoil	Flettner rotor
0.1	3	9.5 MW	12.68 MW
0.15	3	12.3 MW	9.2 MW
0.06	3.5	7.6 MW	11.2 MW
0.1	3.5	12.38 MW	(No useful output)
0.25	1.55	5.44 MW	12.76 MW
0.27	1.55	7.26 MW	12.69 MW

Table 2 Mean power output over one turbine rotation

It is advisable to maintain a tip speed ratio of no more than 3.5 to avoid cavitation. This limit gives the maximum angular frequency for a given diameter of machine and therefore will provide the lowest torque solution for a desired power output. With a tip speed ratio of 3.5 and a solidity of 0.1 it is possible to produce a mean power output of 12.38 MW using hydrofoils, but with Flettner rotors and an optimum solidity of 0.06 the mean power output is only 11.12 MW. However, the Flettner rotors outperform the hydrofoils at lower tip speed ratios. For example, a tip speed ratio of 1.55 and solidity of 0.25 can produce 12.76 MW of power using 9.8 m diameter Flettner rotors, whereas the mean power output using hydrofoils peaks at 7.26 MW with a solidity of 0.27.

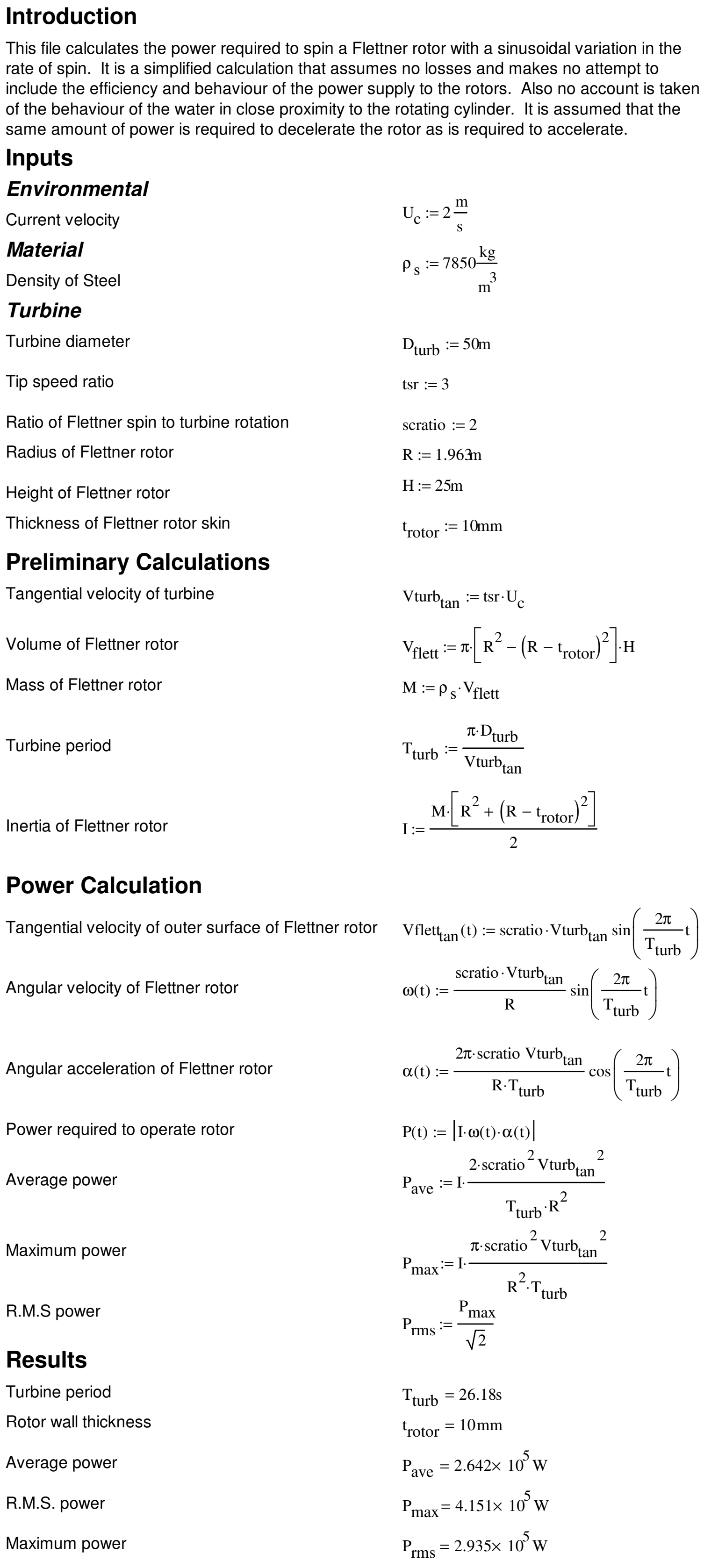
The superior performance of the hydrofoil machine at higher tip speed ratios can be explained by considering the component of drag in the tangential direction. When the machine is operating with a high angular frequency the rotation induced component of effective velocity will start to dominate the current resulting in a small angle between the tangential direction and the effective velocity. The small angle means that the drag is predominantly in the tangential direction and this causes a reduction in performance for a high drag device such as a Flettner rotor but is not so significant for a low drag hydrofoil, which may yield improved performance because of a more constant angle of attack.

The importance of the drag on the Flettner rotor powered machine can be demonstrated by artificially increasing the drag coefficient. For a tip speed ratio of 3 and solidity of 0.1, doubling the drag coefficient prevents the machine producing power, although the machine can still function if the tip speed ratio is lowered to 1.55 and the solidity increased to 0.21. In contrast, the hydrofoil powered machine can still produce useful power with an 8 fold increase in the drag. This is highly significant as the model does not yet account for the drag of the supporting structure or frictional losses.

3.3 Power Required to Generate Lift

Using Flettner rotors to provide the tangential force required to rotate a vertical axis machine will require a significant power input. A simplified calculation is presented in **Figure 3-6**. This calculation looks at the power required to spin a single Flettner rotor assuming no losses and ignoring the behaviour of the water in close proximity to the rotating cylinder. The calculation also assumes that the same amount of power is required to decelerate the spinning cylinder as is required to accelerate it. It is possible that the machine could be designed to use power generated by decelerating a cylinder to assist in accelerating a cylinder 90° out of phase.

Figure 3-6 shows that it takes 0.26 MW to power one Flettner rotor for the example in Table 1, so a total input of approximately 1 MW for 4 rotors could produce an additional 2 MW of net power compared with a hydrofoil of the same solidity (see Table 2). However if the hydrofoil has a larger solidity, 0.15 compared with 0.1 for the Flettner rotor, Table 2 shows that both machines have similar power outputs and this results in approximately 1 MW more net power for the hydrofoil case. It is worth noting that increasing the diameter of the machine for the same swept area results in an increase in the net power because of a drop in the power required to spin the Flettner rotors. This is predominantly due to the increase in period of the machine, in order to maintain a constant tip speed ratio, resulting in lower accelerations of the rotors.



Introduction

This file calculates the power required to spin a Flettner rotor with a sinusoidal variation in the rate of spin. It is a simplified calculation that assumes no losses and makes no attempt to include the efficiency and behaviour of the power supply to the rotors. Also no account is taken of the behaviour of the water in close proximity to the rotating cylinder. It is assumed that the same amount of power is required to decelerate the rotor as is required to accelerate.

Inputs

Environmental

Current velocity $U_c := 2\frac{m}{s}$

Material

Density of Steel $\rho_s := 7850\frac{kg}{m^3}$

Turbine

Turbine diameter $D_{turb} := 50m$

Tip speed ratio $tsr := 3$

Ratio of Flettner spin to turbine rotation $scratio := 2$

Radius of Flettner rotor $R := 1.963m$

Height of Flettner rotor $H := 25m$

Thickness of Flettner rotor skin $t_{rotor} := 10mm$

Preliminary Calculations

Tangential velocity of turbine $Vturb_{tan} := tsr \cdot U_c$

Volume of Flettner rotor $V_{flett} := \pi \cdot \left[R^2 - (R - t_{rotor})^2\right] \cdot H$

Mass of Flettner rotor $M := \rho_s \cdot V_{flett}$

Turbine period $T_{turb} := \frac{\pi \cdot D_{turb}}{Vturb_{tan}}$

Inertia of Flettner rotor $I := \frac{M \cdot \left[R^2 + (R - t_{rotor})^2\right]}{2}$

Power Calculation

Tangential velocity of outer surface of Flettner rotor $Vflett_{tan}(t) := scratio \cdot Vturb_{tan} \sin\left(\frac{2\pi}{T_{turb}}t\right)$

Angular velocity of Flettner rotor $\omega(t) := \frac{scratio \cdot Vturb_{tan}}{R} \sin\left(\frac{2\pi}{T_{turb}}t\right)$

Angular acceleration of Flettner rotor $\alpha(t) := \frac{2\pi \cdot scratio\ Vturb_{tan}}{R \cdot T_{turb}} \cos\left(\frac{2\pi}{T_{turb}}t\right)$

Power required to operate rotor $P(t) := \left|I \cdot \omega(t) \cdot \alpha(t)\right|$

Average power $P_{ave} := I \cdot \frac{2 \cdot scratio^2\ Vturb_{tan}^{\ 2}}{T_{turb} \cdot R^2}$

Maximum power $P_{max} := I \cdot \frac{\pi \cdot scratio^2\ Vturb_{tan}^{\ 2}}{R^2 \cdot T_{turb}}$

R.M.S power $P_{rms} := \frac{P_{max}}{\sqrt{2}}$

Results

Turbine period $T_{turb} = 26.18s$

Rotor wall thickness $t_{rotor} = 10mm$

Average power $P_{ave} = 2.642 \times 10^5 W$

R.M.S. power $P_{max} = 4.151 \times 10^5 W$

Maximum power $P_{rms} = 2.935 \times 10^5 W$

Figure 3-6 Power required to spin Flettner rotor

4 CONCLUSIONS AND RECOMMENDATIONS

Vertical and horizontal axis machines have been compared, and vertical axis machines clearly provide increased flexibility when the flow is constrained top and bottom (i.e. shallow seas) and they give added control of torque and angular frequency compared with horizontal axis machines.

Machines powered by Flettner rotors or Turbosails™ can, for certain configurations, produce larger driving forces than conventional hydrofoils. The peak value, with respect to solidity, of the mean power output corresponds to a higher optimum solidity for the hydrofoil powered machine compared to the Flettner rotor and Turbosail™ machines. The peak in the mean power is also dependent on the tip speed ratio and occurs at slightly higher values of tip speed ratio for the hydrofoil powered machine, but remains below the cavitation limit. The result of these characteristics is approximately equal peak values of mean power output but with a smaller corresponding torque for the hydrofoil case.

The Flettner rotor or Turbosail™ powered machine is more sensitive to drag because of the lower lift to drag ratio when compared to the hydrofoil powered machine. The model currently underestimates drag because no allowance is made for the supporting structure. There is also uncertainty regarding the behaviour of the lift coefficient for Flettner rotors operating in an unsteady manner. It is almost certain that these two points will combine to significantly lower the performance of the Flettner rotor powered machine, and may even prevent the machine from operating.

After considering these results we believe that there is not a clear case in support of either Flettner rotors or Turbosails™ in application as vertical axis tidal stream turbines, and as such we do not propose to continue to phase 2 of this project.

5 REFERENCES

1. Malavard, L., “Un nouveau propulseur eolien de navire”, Comptes Rendus de l'Academie des Sciences, (1984), 57-72.
2. Swanson, W.M., The Magnus effect - a summary of investigations to date, J. Basic Eng. (T ASME), 83, (1961), 461-470.
3. Templin, R.J. & Rangi, R.S., “Vertical axis wind turbine development in Canada”, Proc. Inst. EE (A) 130(1983), p. 555-561.

APPENDIX A

1 INTRODUCTION

This report contains a description of the mathematical model developed for the following study:
Project Ref. T/06/00217/00/00

“Potential Applications for Flettner Rotors and Turbosails™ in Tidal Stream Turbines”

The description presented is the MathCAD sheet used to check and validate the mathematical model. Example calculations are also included to demonstrate that the results accord with common sense.

2 THE MATHEMATICAL MODEL

The mathematical model has been developed using Matlab and is based on theory and experiments from the study of vertical axis wind turbines (VAWT). The mathematical model assumes that the base case configuration, Figure 2-1, for the turbine will be four parallel lift devices located at a constant radius from the centre of the turbine. Adjacent lift devices will therefore subtend an angle of 90° at the centre of the turbine. The sign conventions used in the model are shown in Figure 2-2 along with the sign conventions used in the MathCAD validation file.

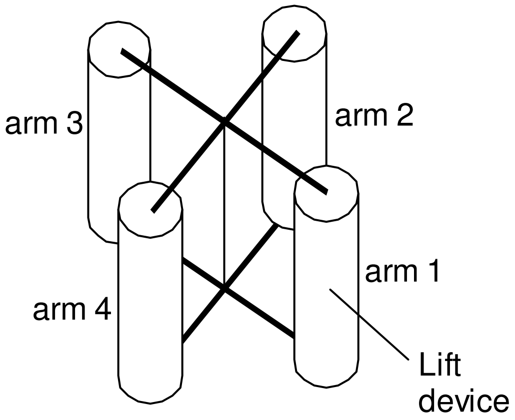
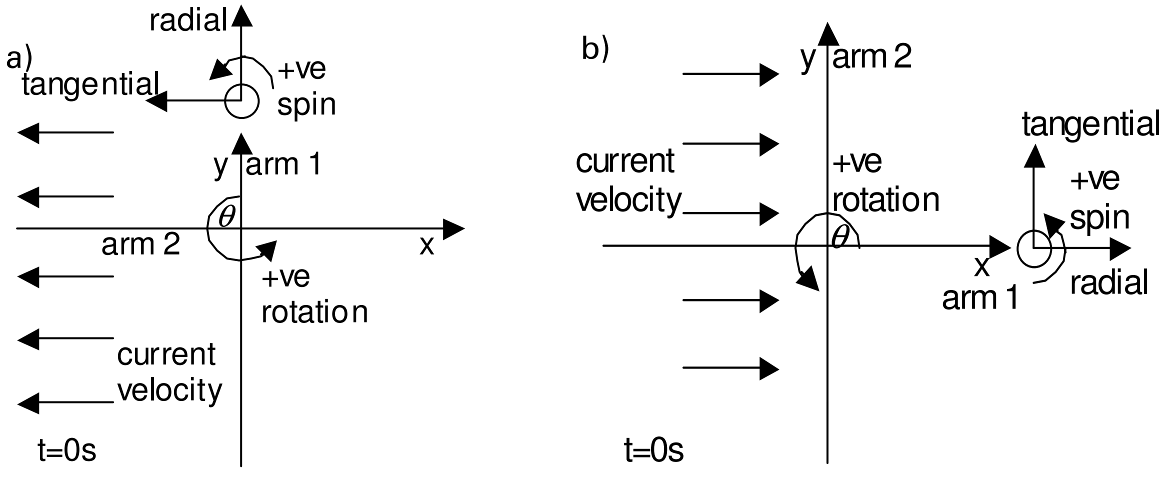


Figure 2-1 Turbine configuration

The lift device shown in **Figure 2-2** is arm 1, with the remaining arms numbered in an anti-clockwise direction, i.e. at t=0s a turbine with 4 lift devices has arm 1 at =0° and arm 2 at =90°, and so on.



Lift +ve when it lags drag by $\pi/2$

Lift +ve when it leads drag by $\pi/2$

Figure 2-2 Mathematical model sign conventions a)Matlab model b)MathCAD validation

The model requires environmental data and turbine geometry as inputs to the calculation. The user has the option of selecting three different types of lift device; hydrofoil, Flettner rotor or Turbosail™. The size of each lift device is calculated by the model from the user defined solidity of the turbine. This allows comparison of different lift devices using constant solidity as a definition of equivalence. The ratio of rotation speed to current speed (TSR – tip speed ratio) at the surface is also entered as a user defined range. The behaviour of the lift devices, pitch angle, spin speed and direction and suction velocity can be specified by the user. The model takes these inputs, and assuming the commonly used $^1/_7$ power law for the current distribution, calculates the effective velocity of flow past the individual lift devices. Blade element theory[4] is used to split the lift devices into independent sections allowing use of the local effective velocity along the length of the lift device.

Knowledge of the effective velocity then allows calculation of the lift and drag forces using the appropriate coefficients. For a hydrofoil the lift and drag coefficients are based on the work of Templin & Rangi [3] in unsteady flow. Swanson [2] provides lift and drag coefficients for a Flettner rotor and Turbosail™ characteristics can be found in Malavard [1].

The radial and tangential forces on the turbine can be calculated from the lift and drag forces on the individual lift devices using basic geometry. The tangential force can be used to calculate the torque and this in turn provides the power output of the turbine. Basic geometry can also provide the forces on the turbine in the direction of, and perpendicular to, the tidal current.

The model uses the single streamtube method[7] to account for the retardation of the current velocity by the turbine and assumes that the current velocity at the rotor is given by

$$V = V_0(1-a) \qquad 2.1.$$

where V is the streamwise current velocity at the turbine, V_0 is the free stream velocity and a is the interference factor. The operation of the turbine will induce rotational and cross-flow elements in the flow but these are currently ignored in the model. The resulting power coefficient is

$$c_P = 4a(1-a)^2 \qquad 2.2.$$

where c_P is the power coefficient, and has a maximum value of 0.592 (in accordance with Betz's Law) when a=0.3. The MathCAD validation file assumes that the interference factor is always optimum with a value of 0.3 but the Matlab code uses an iterative procedure to calculate the actual value. This procedure uses the lift and drag coefficients for the turbine blades to calculate the thrust and torque for the turbine. The thrust can also be calculated from the change in momentum flow rate and the torque from the change in angular momentum flow rate multiplied by the radius. This gives a set of equations that can be solved iteratively for the interference factors.

The model is accessed through a GUI (Graphical User Interface) that allows the user to select various plotting options to display the behaviour of the key turbine variables.

3 DETAILED DESCRIPTION OF MODEL (MATHCAD VALIDATION FILE)

Introduction

This Mathcad sheet performs a basic analysis of a vertical axis tidal current turbine. The lift devices are specified by the user and can be either hydrofoils, Flettner rotors or Turbosails. For a user defined turbine geometry the effective velocity is calculated. This is used to calculate the lift and drag forces which are then resolved into the radial and tangential directions allowing calculation of the power derived from the turbine rotation.

Limitations

This Mathcad sheet calculates the forces and power output in the absence of any wake effects. It is not known precisely how Flettner rotors behave in wakes. Also, all these quantities will vary with height (depth), this analysis assumes a current profile described by the 1/7 power law.

Data

Environmental Data

Surface current velocity	$\mathrm{VcSurf} := 2\frac{\mathrm{m}}{\mathrm{s}}$
Sea water density	$\mathrm{density} := 1025\frac{\mathrm{kg}}{\mathrm{m}^3}$
Sea depth	$\mathrm{wDepth} := 50\mathrm{m}$
Number of depth increments	$\mathrm{nzh} := 5$

Turbine Data

Number of arms (blades, rotors or whatever)	$\mathrm{narm} := 4$	
Overall radius	$\mathrm{radius} := 10\mathrm{m}$	
Overall height	$\mathrm{height} := 20\mathrm{m}$	
Rotation speed / current speed	$\mathrm{rcratio} := 3$	
Turbine type	$\mathrm{itype} := 1$	itype=1 for hydrofoil itype=2 for Flettner rotor
Phase angle increment	$\mathrm{ntheta} := 12$	itype=3 for Turbosail™
Solidity	$\mathrm{sol} := 0.2$	
Interference factor	$a := 0.3$	

hydrofoil characteristics

Lift coefficient constant	$\mathrm{cl0} := 4.297$
Drag coefficient constants	$\mathrm{cd0} := 0.01$
	$\mathrm{cd1} := 0.05$
Pitch Angle	$\mathrm{pitchAng} := 0.0$

Flettner rotor characteristics

Spin speed / rotation speed	$\mathrm{scratio} := 2$

Turbosail characteristics

Suction velocity	$\mathrm{csuck} := 0.04$
Stall Angle	$\mathrm{stallAng} := 406.8\,\mathrm{csuck}\cdot(1 + 14.87\cdot\mathrm{csuck})$

Environmental Calculations

Set up range variable

$$\mathrm{iz} := 0..\ \mathrm{nzh}$$

Locate base of lift device

$$\mathrm{bDepth} := 0.5\cdot(\mathrm{wDepth} - \mathrm{height})$$

Locate top of lift device

$$\mathrm{tDepth} := \mathrm{bDepth} + \mathrm{height}$$

Create Z-position matrix (0 at seabed)

$$\mathrm{zPos}_{\mathrm{iz}} := \mathrm{bDepth} + \frac{\mathrm{height}\cdot\mathrm{iz}}{\mathrm{nzh}}$$

Current profile

$$\mathrm{Vcurr}_{\mathrm{iz}} := \frac{\mathrm{VcSurf}}{\mathrm{wDepth}^{\frac{1}{7}}}\cdot\left(\mathrm{zPos}_{\mathrm{iz}}\right)^{\frac{1}{7}}$$

Turbine Calculations

Basic Calculations

Angular spacing between arms

$$\mathrm{tAngle} := \frac{2\cdot\pi}{\mathrm{narm}}$$

Create phase angle matrix for all arms throughout the rotation

Increment phase angle

$$\mathrm{th0} := \frac{2\cdot\pi}{\mathrm{ntheta}}$$

Set up range variables

$$\mathrm{it} := 0..\ \mathrm{ntheta}$$
$$\mathrm{ib} := 0..\ \mathrm{narm} - 1$$

Azimuthal angle for all arms at 12 phases in cycle

$$\mathrm{theta}_{\mathrm{ib},\mathrm{it}} := \mathrm{it}\cdot\mathrm{th0} + \mathrm{ib}\cdot\mathrm{tAngle}$$

Pitch angle (hydrofoil only)

$$\mathrm{pitchAng}_{\mathrm{ib},\mathrm{it}} := \mathrm{if}\left(\mathrm{itype} = 1, -0.1745\cos\left(\mathrm{theta}_{\mathrm{ib},\mathrm{it}}\right), 0\right)$$

Create user function to give correct sign

$$\mathrm{sign2}(x) := \mathrm{if}\left(x = 0, -1, \frac{x}{|x|}\right)$$

Spin direction (Flettner rotor only)

$$\mathrm{spinDir}_{\mathrm{ib},\mathrm{it}} := \mathrm{if}\left(\mathrm{itype} = 2, \mathrm{sign2}\left(\cos\left(\mathrm{theta}_{\mathrm{ib},\mathrm{it}}\right)\right), 1\right)$$

Device width

$$\mathrm{devWidth} := \frac{\mathrm{sol}\cdot 2\cdot\pi\cdot\mathrm{radius}}{\mathrm{narm}}$$

Effective Velocity

Effective velocity due to rotation and current (fluid velocity relative to lift device, lift device rotating anti-clockwise with anti-clockwise rotation positive measured from +ve x-axis, current in the +ve x-direction)

$$\mathrm{factorveffx}_{\mathrm{ib},\mathrm{it}} := \left[\mathrm{rcratio}\cdot\sin\left(\mathrm{theta}_{\mathrm{ib},\mathrm{it}}\right) + (1 - a)\right]$$

Effective velocity in x-direction

$$\mathrm{veffx}_{\mathrm{iz}} := \mathrm{Vcurr}_{\mathrm{iz}}\cdot\mathrm{factorveffx}$$

$$\mathrm{factorveffy}_{\mathrm{ib},\mathrm{it}} := -\mathrm{rcratio}\cdot\cos\left(\mathrm{theta}_{\mathrm{ib},\mathrm{it}}\right)$$

Effective velocity in y-direction

$$\mathrm{veffy}_{\mathrm{iz}} := \mathrm{Vcurr}_{\mathrm{iz}}\cdot\mathrm{factorveffy}$$

Effective velocity

$$\mathrm{veff}_{\mathrm{iz}} := \overrightarrow{\sqrt{\left(\mathrm{veffx}_{\mathrm{iz}}\right)^2 + \left(\mathrm{veffy}_{\mathrm{iz}}\right)^2}}$$

Angle between -veff and tangential velocity of cylinder

$$\mathrm{rAngle}_{\mathrm{ib},\mathrm{it}} := \mathrm{atan2}\left[\left(\frac{\mathrm{rcratio}}{1 - a} + \sin\left(\mathrm{theta}_{\mathrm{ib},\mathrm{it}}\right)\right), \cos\left(\mathrm{theta}_{\mathrm{ib},\mathrm{it}}\right)\right]$$

effective blade incidence (hydrofoil only)

$$\mathrm{effAlpha}_{\mathrm{ib},\mathrm{it}} := \mathrm{rAngle}_{\mathrm{ib},\mathrm{it}} + \mathrm{pitchAng}_{\mathrm{ib},\mathrm{it}}$$

Lift and Drag Force

Create matrix of spin/rotation (Flettner) $sc_{ib,it} := \text{if}(\text{itype} = 2, \text{scratio}, 0)$

Lift coefficient

$$CL_{ib,it} := \left| \begin{array}{l} \text{cl0}\cdot \text{effAlpha}_{ib,it} \ \text{ if } \ \text{itype} = 1 \\ \text{if } \ \text{itype} = 2 \\ \quad \left| \begin{array}{l} (\text{spinDir})_{ib,it}\cdot\left(0.5\cdot sc_{ib,it} + 6.4\right) \ \text{ if } \ sc_{ib,it} > 3 \\ (\text{spinDir})_{ib,it}\cdot\left(3.5\cdot sc_{ib,it} - 2.7\right) \ \text{ if } \ 0 < sc_{ib,it} \le 3 \end{array} \right. \\ \text{if } \ \text{itype} = 3 \\ \quad \left| \begin{array}{l} \text{sign}\left(\text{effAlpha}_{ib,it}\right)\cdot\left(1.6 + 50\cdot\text{csuck} + 9.2\cdot\left|\text{effAlpha}_{ib,it}\right|\right) \ \text{ if } \ \left|\text{effAlpha}_{ib,it}\right| < \text{stallAng} \\ \text{sign}\left(\text{effAlpha}_{ib,it}\right)\cdot 2.5 \ \text{ otherwise} \end{array} \right. \end{array} \right.$$

Drag Coefficient

$$CD_{ib,it} := \left| \begin{array}{l} \left[\text{cd0} + \text{cd1}\cdot\left(\text{effAlpha}_{ib,it}\right)^2\right] \ \text{ if } \ \text{itype} = 1 \\ \text{if } \ \text{itype} = 2 \\ \quad \left| \begin{array}{l} \left(\left(1.7\cdot sc_{ib,it} - 3.1\right)\right) \ \text{ if } \ \left(sc_{ib,it} < 3.8\right) \\ \left(\left(-0.3\cdot sc_{ib,it} + 4.2\right)\right) \ \text{ if } \ sc_{ib,it} \ge 3.8 \end{array} \right. \\ 0.6 + 3\cdot\left(\text{effAlpha}_{ib,it}\right)^2 \ \text{ if } \ \text{itype} = 3 \end{array} \right.$$

Lift and drag factor $\alpha_1 := 0.5\cdot\text{density}\cdot\text{devWidth}\cdot\dfrac{\text{height}}{\text{nzh}}$

Lift Force $\text{lift}_{iz} := \alpha_1\cdot\overrightarrow{\left[\left(\text{veff}_{iz}\right)^2\cdot CL\right]}$

Drag Force $\text{drag}_{iz} := \alpha_1\cdot\overrightarrow{\left[\left(\text{veff}_{iz}\right)^2\cdot CD\right]}$

Resolve in tangential and radial directions

Radial Component of Resultant Force $\text{Rrad}_{iz} := \overrightarrow{\left[\text{lift}_{iz}\cdot(\cos(\text{rAngle}))\right]} + \overrightarrow{\left(\text{drag}_{iz}\cdot\sin(\text{rAngle})\right)}$

Tangential Component of Resultant Force $\text{Rtang}_{iz} := \overrightarrow{\left(\text{lift}_{iz}\cdot\sin(\text{rAngle})\right)} - \overrightarrow{\left(\text{drag}_{iz}\cdot\cos(\text{rAngle})\right)}$

Power

Torque $\text{Torque}_{iz} := \text{radius}\cdot\text{Rtang}_{iz}$

Power (Torque x angular velocity) $\text{Power}_{iz} := \overrightarrow{\left(\text{rcratio}\cdot\text{Vcurr}_{iz}\cdot\text{Rtang}_{iz}\right)}$

Total power over depth $\text{power} := \sum_{iz} \text{Power}_{iz}$

Total Power summed over all lift devices $\text{totPower}_{0,it} := \sum_{ib} \text{power}_{ib,it}$

Force

Force in x-direction

$$Rx_{iz} := \overrightarrow{\left(Rrad_{iz} \cdot \cos(theta)\right)} - \overrightarrow{\left(Rtang_{iz} \cdot \sin(theta)\right)}$$

Force in y-direction

$$Ry_{iz} := \overrightarrow{\left(Rrad_{iz} \cdot \sin(theta)\right)} + \overrightarrow{\left(Rtang_{iz} \cdot \cos(theta)\right)}$$

Total force in x-direction over depth for individual devices

$$totx := \sum_{iz} Rx_{iz}$$

Total force in x-direction summed over all lift devices

$$totX_{0,it} := \sum_{ib} totx_{ib,it}$$

Total force in y-direction over depth for individual devices

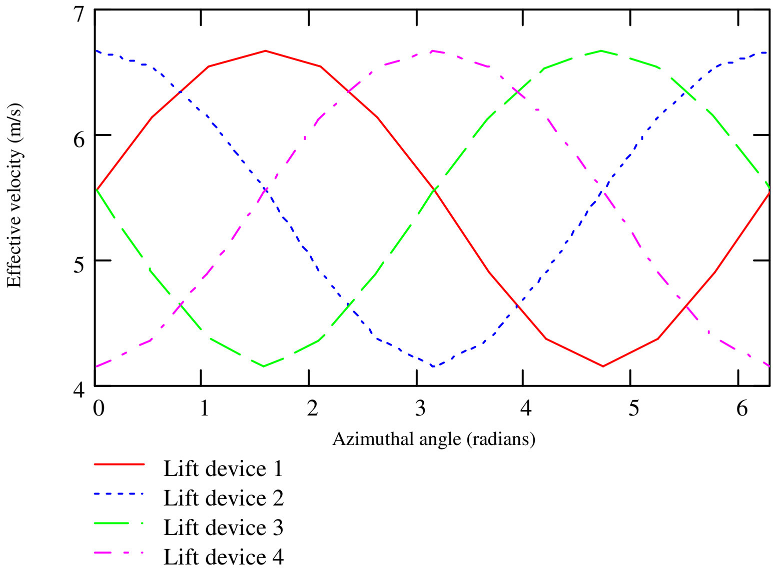
$$toty := \sum_{iz} Ry_{iz}$$

Total force in y-direction summed over all lift devices

$$totY_{0,it} := \sum_{ib} toty_{ib,it}$$

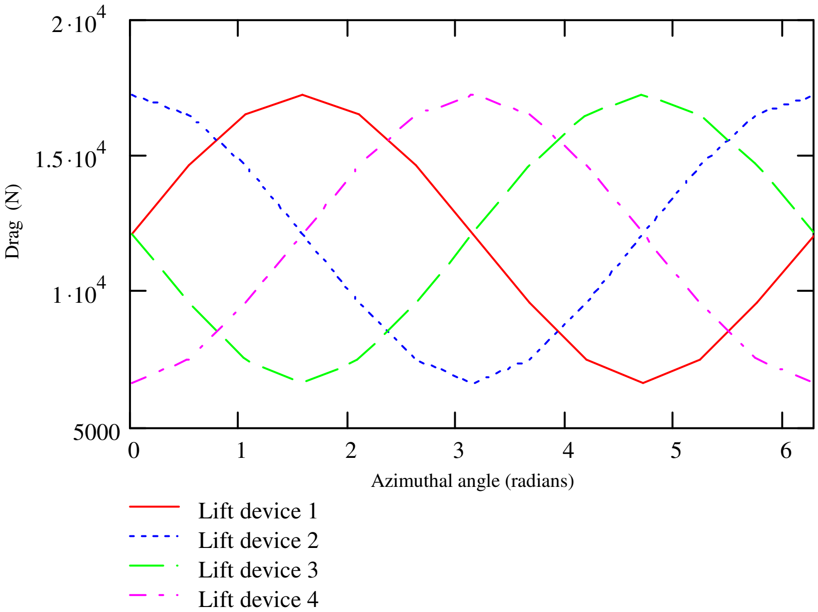
Average effective velocity over depth for individual devices

$$veffa := \frac{\sum_{iz} veff_{iz}}{nzh + 1}$$



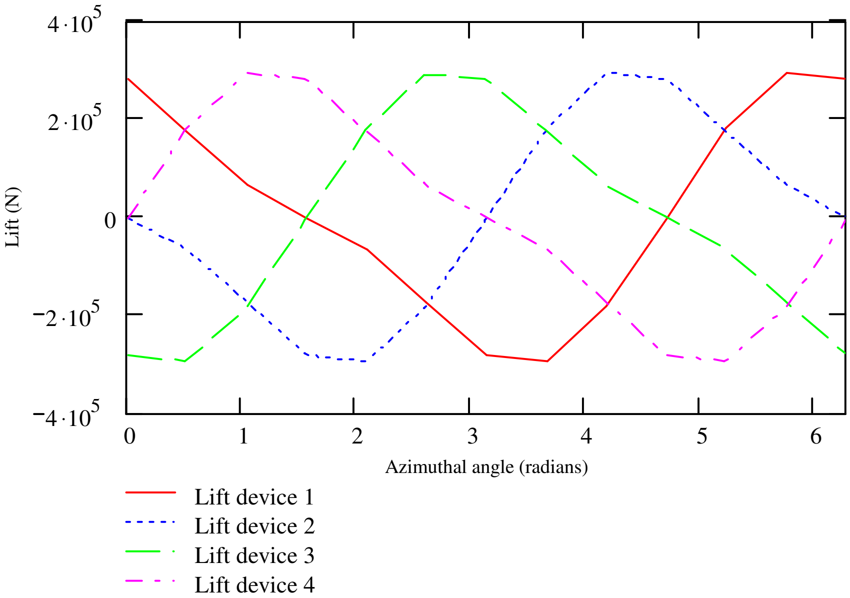
Total drag over depth

$$\text{dragt} := \sum_{iz} \text{drag}_{iz}$$



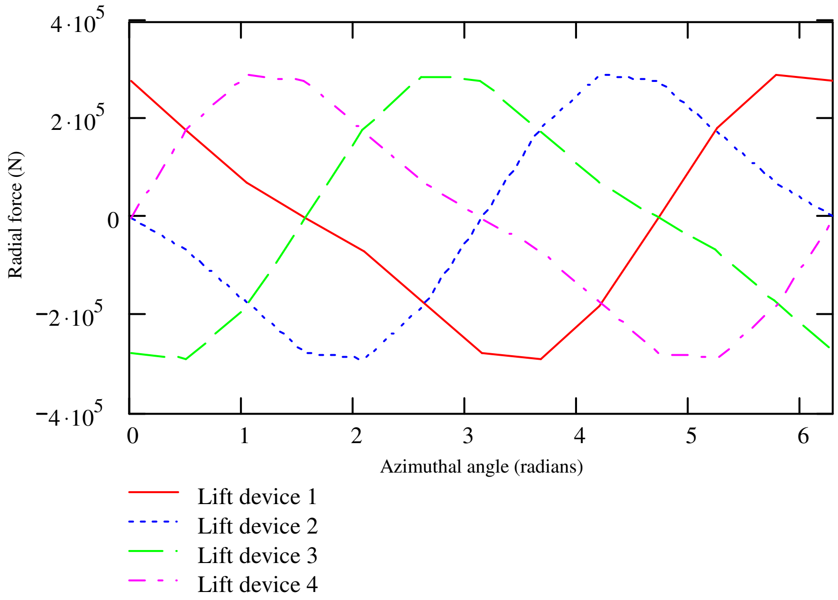
Total lift over depth

$$\text{liftt} := \sum_{iz} \text{lift}_{iz}$$



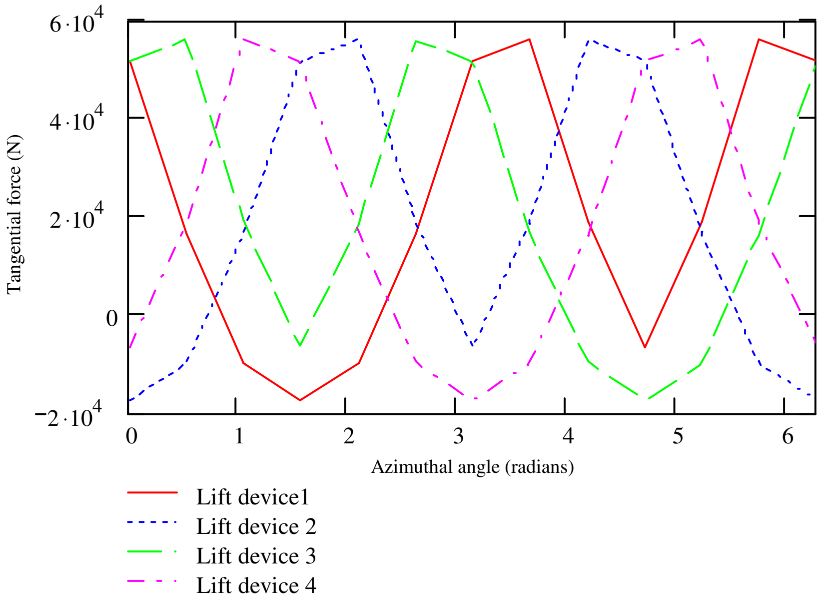
Total radial force over depth

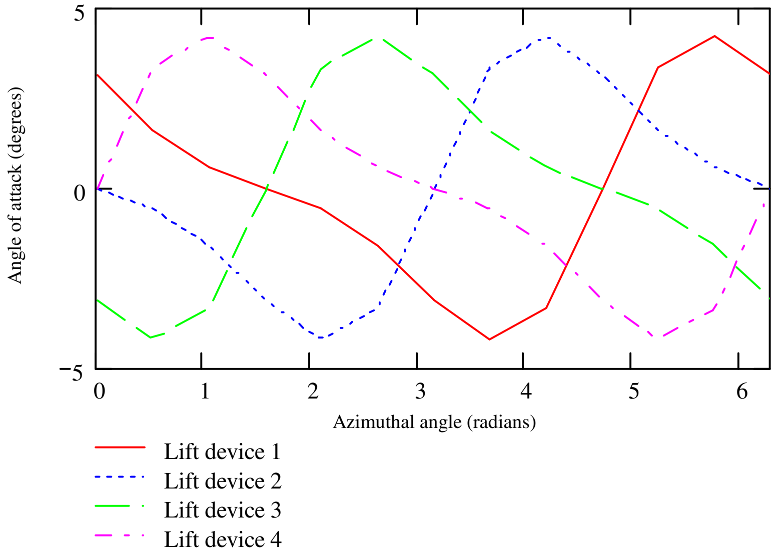
$$Rradt := \sum_{iz} Rrad_{iz}$$



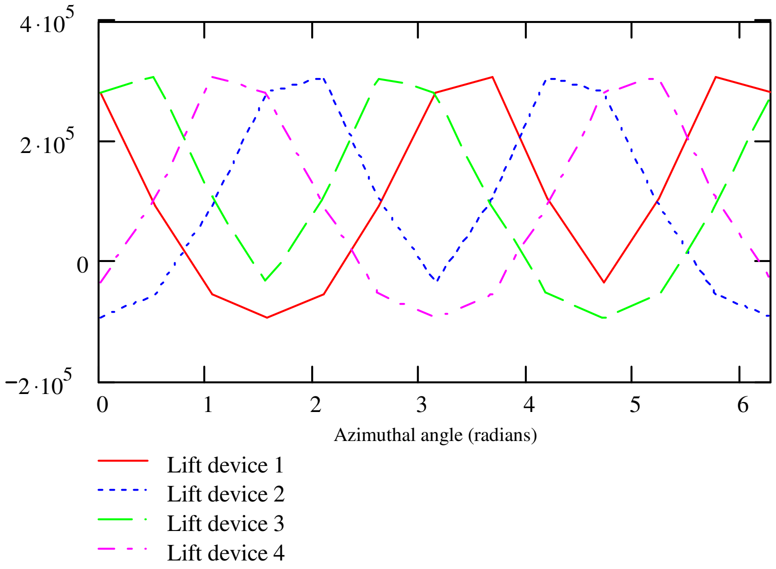
Total tangential force over depth

$$Rtangt := \sum_{iz} Rtang_{iz}$$





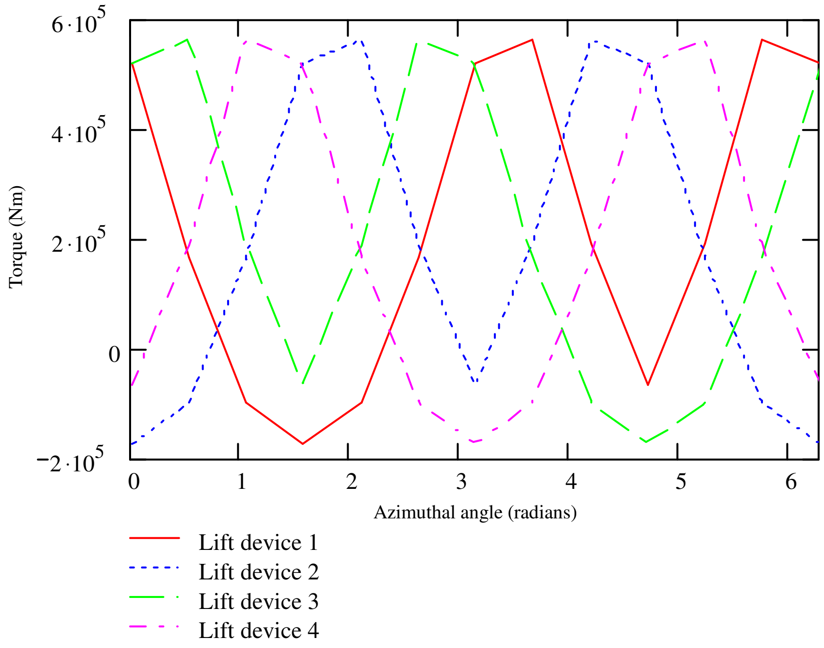
Angle of attack (degrees)
5
0
−5
0
1
2
3
4
5
6
Azimuthal angle (radians)
Lift device 1
Lift device 2
Lift device 3
Lift device 4

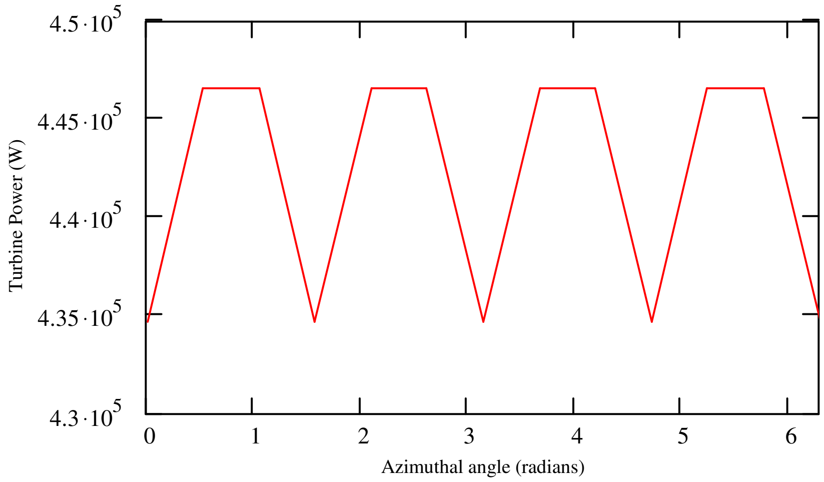


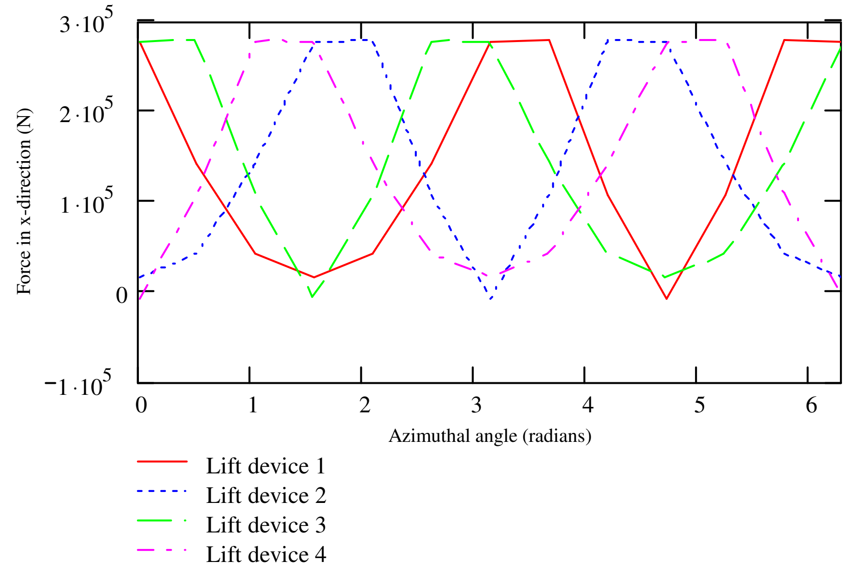
Power (W)
$4 \cdot 10^5$
$2 \cdot 10^5$
0
$-2 \cdot 10^5$
0
1
2
3
4
5
6
Azimuthal angle (radians)
Lift device 1
Lift device 2
Lift device 3
Lift device 4

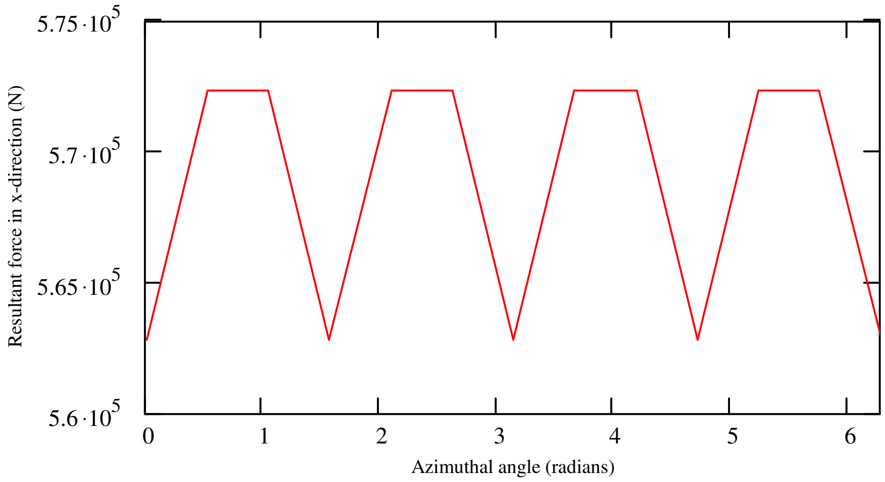
Total torque over depth for each lift device

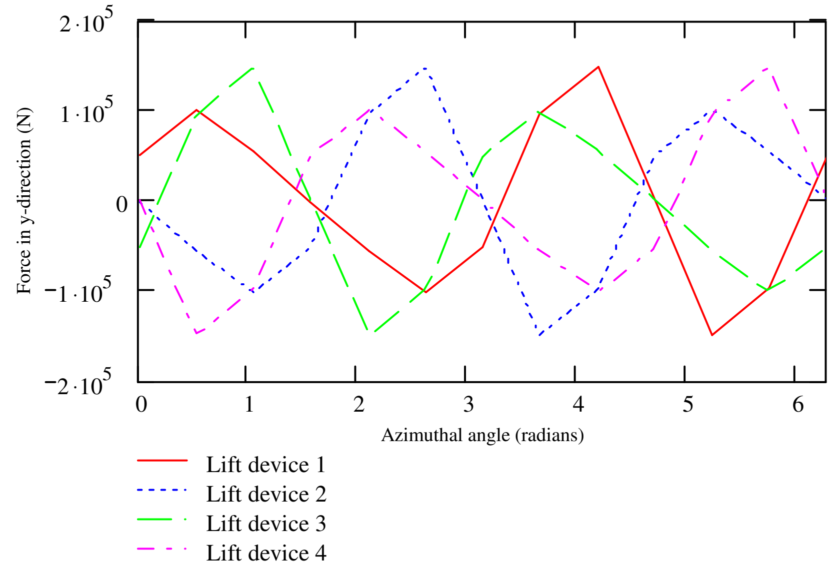
$$\text{torque} := \sum_{iz} \text{Torque}_{iz}$$

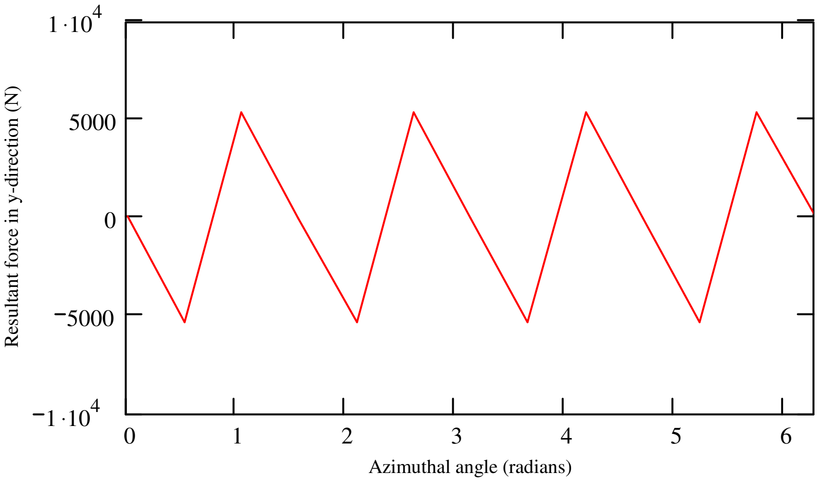





$3 \cdot 10^5$
$2 \cdot 10^5$
$1 \cdot 10^5$
0
$-1 \cdot 10^5$
Force in x-direction (N)
0
1
2
3
4
5
6
Azimuthal angle (radians)
Lift device 1
Lift device 2
Lift device 3
Lift device 4


$5.75 \cdot 10^5$
$5.7 \cdot 10^5$
$5.65 \cdot 10^5$
$5.6 \cdot 10^5$
Resultant force in x-direction (N)
0
1
2
3
4
5
6
Azimuthal angle (radians)


$2 \cdot 10^5$
$1 \cdot 10^5$
0
$-1 \cdot 10^5$
$-2 \cdot 10^5$
Force in y-direction (N)
0
1
2
3
4
5
6
Azimuthal angle (radians)
Lift device 1
Lift device 2
Lift device 3
Lift device 4


$1 \cdot 10^4$
5000
0
-5000
$-1 \cdot 10^4$
Resultant force in y-direction (N)
0
1
2
3
4
5
6
Azimuthal angle (radians)

4 FUTURE DEVELOPEMENTS

The model currently does not calculate the power required to generate lift from either the Flettner rotor or the Turbsail devices. This can result in power outputs in excess of the power available from the tidal current passing through the swept area of the turbine. Future development of the model must include the power input required to spin the Flettner rotors or to provide suction for the Turbosails™ so that the net power can be calculated. The model must also account for the efficiency of energy extraction from the tidal current by the turbine and should also allow for losses during shaft torque to electricity conversion. The behaviour of the power take-off is crucial to optimisation of the machine.

The model must be developed further to include the bending moments and shear forces on the turbine to allow sizing of the supports and the resulting losses due to drag forces.

Once the model can provide a suitable description of the machines behaviour using the current quasi-steady assumptions it will be appropriate to develop the model to allow investigation of the transient and unsteady behaviour.

5 EXAMPLES

Example 1: 0.5 MW Hydrofoil, radius 5m, height 10m

Data

Environmental Data

Surface current velocity	$VcSurf := 2\frac{m}{s}$
Sea water density	$density := 1025\frac{kg}{m^3}$
Sea depth	$wDepth := 50m$
Number of depth increments	$nzh := 5$

Turbine Data

Number of arms (blades, rotors or whatever)	$narm := 4$
Overall radius	$radius := 5m$
Overall height	$height := 10m$
Rotation speed / current speed	$rcratio := 3.5$
Turbine type	$itype := 1$
Phase angle increment	$ntheta := 12$
Solidity	$sol := 0.15$
Interference factor	$a := 0.3$

itype=1 for hydrofoil
itype=2 for Flettner rotor
itype=3 for Turbosail^{TM}

hydrofoil characteristics

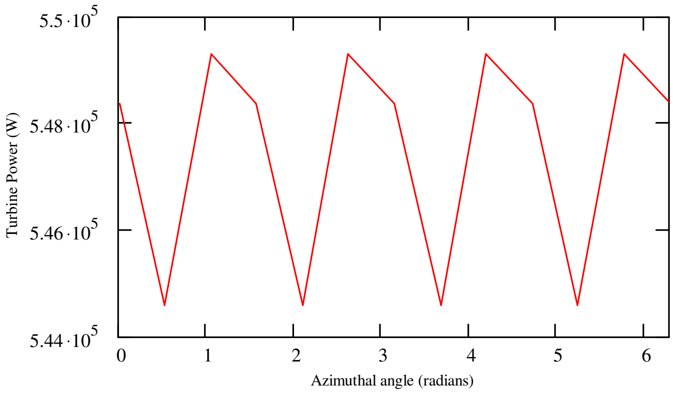
Lift coefficient constant	$cl0 := 4.297$
Drag coefficient constants	$cd0 := 0.01$
	$cd1 := 0.05$
Pitch Angle	$pitchAng := 0.0$

Flettner rotor characteristics

Spin speed / rotation speed	$scratio := 2$

Turbosail characteristics

Suction velocity	$csuck := 0.04$
Stall Angle	$stallAng := 406.8 csuck\cdot(1 + 14.87 csuck)$



$$P_{mean} = 5.475\times 10^5\ W$$

Example 2: 5 MW Hydrofoil, radius 25m, height 25m

Data

Environmental Data

Surface current velocity	$\mathrm{VcSurf} := 2\frac{m}{s}$
Sea water density	$\mathrm{density} := 1025\frac{kg}{m^3}$
Sea depth	$\mathrm{wDepth} := 50m$
Number of depth increments	$\mathrm{nzh} := 5$

Turbine Data

Number of arms (blades, rotors or whatever)	$\mathrm{narm} := 4$
Overall radius	$\mathrm{radius} := 25m$
Overall height	$\mathrm{height} := 25m$
Rotation speed / current speed	$\mathrm{rcratio} := 3.5$
Turbine type	$\mathrm{itype} := 1$
Phase angle increment	$\mathrm{ntheta} := 12$
Solidity	$\mathrm{sol} := 0.12$
Interference factor	$a := 0.3$

itype=1 for hydrofoil
itype=2 for Flettner rotor
itype=3 for Turbosail^{TM}

hydrofoil characteristics

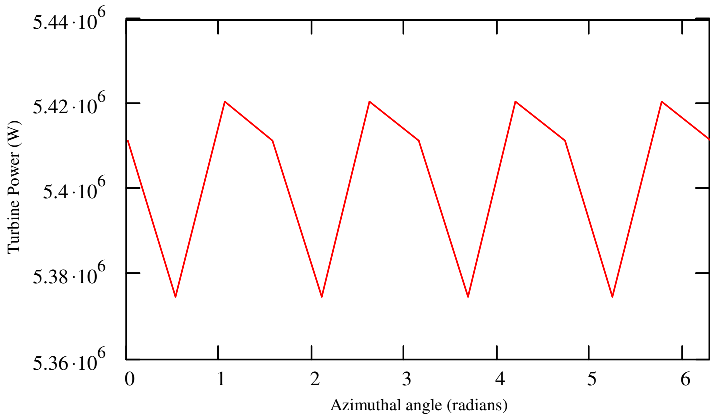
Lift coefficient constant	$\mathrm{cl0} := 4.297$
Drag coefficient constants	$\mathrm{cd0} := 0.01$
	$\mathrm{cd1} := 0.05$
Pitch Angle	$\mathrm{pitchAng} := 0.0$

Flettner rotor characteristics

Spin speed / rotation speed	$\mathrm{scratio} := 2$

Turbosail characteristics

Suction velocity	$\mathrm{csuck} := 0.04$
Stall Angle	$\mathrm{stallAng} := 406.8\,\mathrm{csuck}\cdot(1 + 14.87\,\mathrm{csuck})$



$$P_{mean} = 5.403\times 10^{6}\,W$$

Example 3: 0.5 MW Flettner rotor, radius 6m, height 10m

Data

Environmental Data

Surface current velocity	$\mathrm{VcSurf} := 2\frac{m}{s}$
Sea water density	$\mathrm{density} := 1025\frac{kg}{m^3}$
Sea depth	$\mathrm{wDepth} := 50m$
Number of depth increments	$\mathrm{nzh} := 5$

Turbine Data

Number of arms (blades, rotors or whatever)	$\mathrm{narm} := 4$
Overall radius	$\mathrm{radius} := 6m$
Overall height	$\mathrm{height} := 10m$
Rotation speed / current speed	$\mathrm{rcratio} := 3.5$
Turbine type	$\mathrm{itype} := 2$
Phase angle increment	$\mathrm{ntheta} := 12$
Solidity	$\mathrm{sol} := 0.04$
Interference factor	$a := 0.3$

itype=1 for hydrofoil
itype=2 for Flettner rotor
itype=3 for Turbosail^{TM}

hydrofoil characteristics

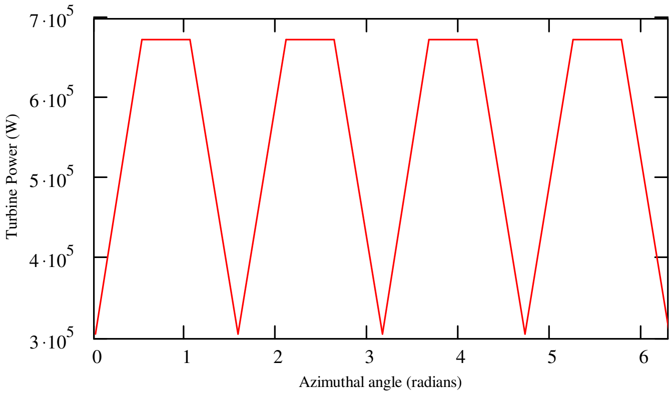
Lift coefficient constant	$\mathrm{cl0} := 4.297$
Drag coefficient constants	$\mathrm{cd0} := 0.01$
	$\mathrm{cd1} := 0.05$
Pitch Angle	$\mathrm{pitchAng} := 0.0$

Flettner rotor characteristics

Spin speed / rotation speed	$\mathrm{scratio} := 2$

Turbosail characteristics

Suction velocity	$\mathrm{csuck} := 0.04$
Stall Angle	$\mathrm{stallAng} := 406.8\,\mathrm{csuck}\cdot(1 + 14.87\,\mathrm{csuck})$



$P_{mean} = 5.312\times 10^5\ W$

Example 4: 5 MW Flettner rotor, radius 25m, height 25m

Data

Environmental Data

Surface current velocity	$VcSurf := 2\frac{m}{s}$
Sea water density	$density := 1025\frac{kg}{m^3}$
Sea depth	$wDepth := 50m$
Number of depth increments	$nzh := 5$

Turbine Data

Number of arms (blades, rotors or whatever)	$narm := 4$
Overall radius	$radius := 25m$
Overall height	$height := 25m$
Rotation speed / current speed	$rcratio := 3.5$
Turbine type	$itype := 2$
Phase angle increment	$ntheta := 12$
Solidity	$sol := 0.04$
Interference factor	$a := 0.3$

itype=1 for hydrofoil
itype=2 for Flettner rotor
itype=3 for Turbosail^{TM}

hydrofoil characteristics

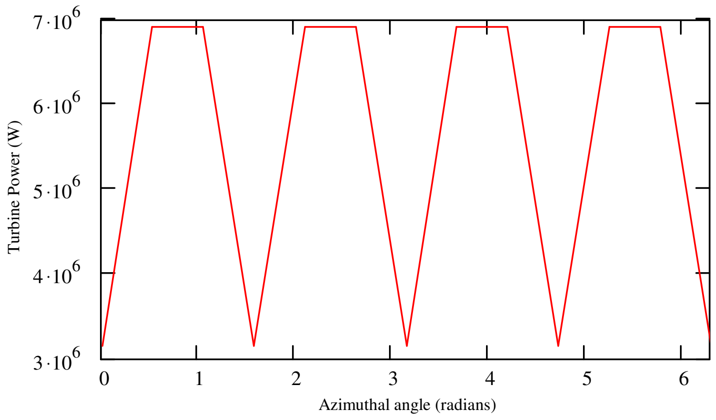
Lift coefficient constant	$cl0 := 4.297$
Drag coefficient constants	$cd0 := 0.01$
	$cd1 := 0.05$
Pitch Angle	$pitchAng := 0.0$

Flettner rotor characteristics

Spin speed / rotation speed	$scratio := 2$

Turbosail characteristics

Suction velocity	$csuck := 0.04$
Stall Angle	$stallAng := 406.8 csuck\cdot(1 + 14.87 csuck)$



$P_{mean} = 5.461\times10^{6}\,W$

Example 5: 0.5 MW Turbosail™ , radius 6m, height 10m

Data

Environmental Data

Surface current velocity	$VcSurf := 2\frac{m}{s}$
Sea water density	$density := 1025\frac{kg}{m^3}$
Sea depth	$wDepth := 50m$
Number of depth increments	$nzh := 5$

Turbine Data

Number of arms (blades, rotors or whatever)	$narm := 4$	
Overall radius	$radius := 6m$	
Overall height	$height := 10m$	
Rotation speed / current speed	$rcratio := 3.5$	
Turbine type	$itype := 3$	itype=1 for hydrofoil itype=2 for Flettner rotor itype=3 for Turbosail^{TM}
Phase angle increment	$ntheta := 12$	
Solidity	$sol := 0.08$	
Interference factor	$a := 0.3$	

hydrofoil characteristics

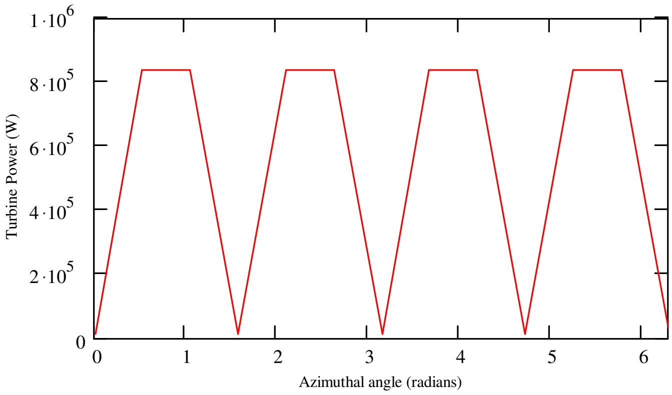
Lift coefficient constant	$cl0 := 4.297$
Drag coefficient constants	$cd0 := 0.01$
	$cd1 := 0.05$
Pitch Angle	$pitchAng := 0.0$

Flettner rotor characteristics

Spin speed / rotation speed	$scratio := 2$

Turbosail characteristics

Suction velocity	$csuck := 0.065$
Stall Angle	$stallAng := 406.8\, csuck \cdot (1 + 14.87\, csuck)$



$$P_{mean} = 5.182\times10^5\,W$$

Example 6: 5 MW Turbosail™, radius 25m, height 25m

Data

Environmental Data

Surface current velocity	$VcSurf := 2\frac{m}{s}$
Sea water density	$density := 1025\frac{kg}{m^3}$
Sea depth	$wDepth := 50m$
Number of depth increments	$nzh := 5$

Turbine Data

Number of arms (blades, rotors or whatever)	$narm := 4$
Overall radius	$radius := 25m$
Overall height	$height := 25m$
Rotation speed / current speed	$rcratio := 3.5$
Turbine type	$itype := 3$
Phase angle increment	$ntheta := 12$
Solidity	$sol := 0.08$
Interference factor	$a := 0.3$

itype=1 for hydrofoil
itype=2 for Flettner rotor
itype=3 for Turbosail^{TM}

hydrofoil characteristics

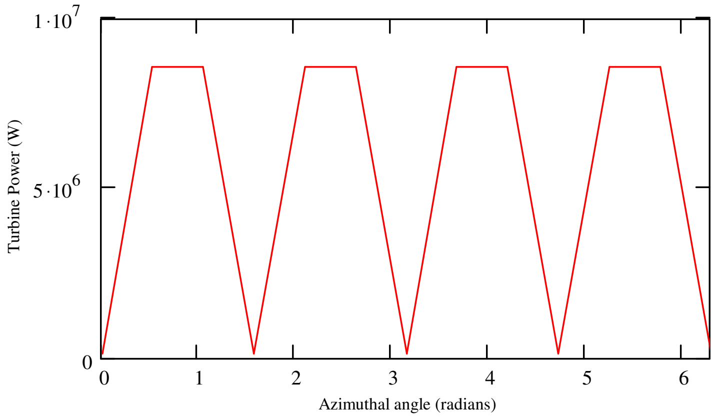
Lift coefficient constant	$cl0 := 4.297$
Drag coefficient constants	$cd0 := 0.01$
	$cd1 := 0.05$
Pitch Angle	$pitchAng := 0.0$

Flettner rotor characteristics

Spin speed / rotation speed	$scratio := 2$

Turbosail characteristics

Suction velocity	$csuck := 0.065$
Stall Angle	$stallAng := 406.8\,csuck\cdot(1 + 14.87\,csuck)$



$P_{mean} = 5.328\times 10^{6}\,W$

6 REFERENCES

4. Glauert, H.,"The Elements of Aerofoil and Airscrew Theory", C.U.P. (1926)
5. Malavard, L., "Un nouveau propulseur eolien de navire", Comptes Rendus de l'Academie des Sciences, (1984), 57-72.
6. Swanson, W.M., The Magnus effect - a summary of investigations to date, J. Basic Eng. (T ASME), 83, (1961), 461-470.
7. Templin, R.J., "Aerodynamic performance theory for the NRC VAWT", Report LTR-LA-160, National Research Council, Canada, (1974)
8. Templin, R.J. & Rangi, R.S., "Vertical axis wind turbine development in Canada", Proc. Inst. EE (A) 130(1983), p. 555-561.