

Performance prediction of horizontal axis marine current turbines

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Abstract. In this study, hydrodynamic performance of a 400 mm diameter horizontal axis marine current turbine model was tested in a cavitation tunnel with 1.21 m x 0.8 m cross-section for over a range of tip speed ratios. Torque and thrust data, as well as cavitation visualizations, for certain operating conditions were acquired. Experimental results indicated that the turbine can be exposed to significant amount of sheet and cloud cavitation over the blades along with vortex cavitation at the blade tips. Inception and distribution of cavitation along the blades of the model turbine were then modelled numerically for design operating conditions using a vortex lattice method. The method was also applied to a turbine tested previously and obtained results were compared with the data available. The comparison between simulation results and experimental data showed a slight difference in terms of span-wise extent of the cavitation region. The cloud and tip vortex cavity observed in experiments cannot be modelled due to the fact that the VLM lacks the ability to predict such types of cavitation. Notwithstanding, the use of such prediction methods can provide a reasonably accurate approach to estimate, therefore take the hydrodynamic effects of cavitation into account in design and analysis of marine current turbines.

Keywords: marine current turbine; cavitation; vortex lattice method

1. Introduction

In recent years, energy generation from renewable sources such as wind, sun, marine and tidal currents etc. has increased very rapidly than before. The large potential possessed by marine currents is regarded as a predictable and sustainable resource for wide scale generation of electrical power in a safe, clean and economic manner. Marine current turbines (MCTs) that exploit underwater currents for power generation are one of the most promising renewable energy systems utilized in oceans and seas (Baltazar *et al.* 2011). It is noted by Fraenkel (2002) that MCTs are technically feasible and current resource is large enough to make a major contribution towards meeting the future energy demand by providing regular and predictable energy as mentioned by Charlier (2003) and Bahaj *et al.* (2007).

Experimental studies on marine current turbines have been of high importance due to the fact

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that the useful information and detailed data they provide for the hydrodynamic design of MCTs and validation of numerical models. Comprehensive information from cavitation tunnel tests and towing tank measurements of a model turbine is given in Bahaj *et al.* (2007). The results of this experimental investigation provided an insight into the effect on performance of changes in the tip immersion of the rotor and possible areas of cavitation inception. An experimental investigation on cavitation, noise and slipstream characteristics of MCTs was conducted in Wang *et al.* (2007). The section (foil) characteristics are also very important to design the blades of MCTs. Some measurements and predictions of forces, pressures and cavitation characteristics on 2-D sections suitable for MCTs were given in Molland *et al.* (2004). Wake studies, on the other hand, for a model MCT can also be found in Myers and Bahaj (2007). The classical blade element momentum (BEM) theory and a boundary element method were applied to optimize a cavitating marine current turbine in Usar and Bal (2011).

Most of the theories and methods originally developed for marine propellers can also be applied to marine current turbines. Introduced for the analysis of fully wetted marine propeller flows by Kerwin and Lee (1978) vortex lattice method is one of the most widely used numerical methods to analyze the performance of cavitating marine current turbines by employing a robust arrangement of singularities and control point spacing to produce accurate results (Kinnas *et al.* 2012). The method was extended to analyze the super-cavitating propellers subject to steady flow and the ability of searching for the mid-chord cavitation was implemented in Kinnas *et al.* (1998). The effect of hub and wake alignment, including the effect of shaft inclination as well as the unsteady wake alignment was developed in order to determine the accurate location of the wake in He *et al.* (2001). The method was further developed for the prediction and investigation of performance characteristics of podded propellers in Bal and Güner (2009). The method was also applied successfully for analysis of optimum cavitating ship propellers and to improve open water propeller performance in Bal (2011a, b). The effect of cavitation on hydrodynamic performance of a marine current turbine was estimated using VLM along with a boundary element and blade element momentum method in Usar and Bal (2015) and Usar (2015).

Present study includes the results of cavitation tunnel tests on a 400 mm diameter model of a three bladed horizontal axis marine current turbine. The tests were carried out in The Emerson Cavitation Tunnel of Newcastle University with a measuring cross-section of 1.21 m x 0.8 m to find out the power coefficient of the turbine in its full operable range of tip speed ratios as well as to observe cavitation behaviour of the turbine for three different cavitation conditions. The corresponding torque and thrust values for 3 m/s inflow velocity were measured, hence power and thrust coefficients were calculated over a range of tip speed ratios.

Distribution of sheet cavitation along the blades of the model turbine was then modelled using a vortex lattice method. In order to assess the capability of the cavitation prediction method, the marine current turbine tested previously in Bahaj *et al.* (2007) was also analysed for a particular operating condition and the results were compared with the experimental data available.

2. Model test setup

2.1 Cavitation tunnel

The tests of the model turbine were conducted in Newcastle University's Emerson Cavitation Tunnel (ECT). The tunnel is a medium size propeller cavitation tunnel as shown in Fig.1, with the

following test section particulars presented in Table 1.

The tunnel is fully enclosed and the velocity was measured using a pitot tube. Tunnel static pressures were measured on the sidewall of the test section. The latest full details of the ECT can be found in (Atlas 2011).

2.2 Description of the model

The tested stream turbine was a three-bladed and stall-regulated turbine designed to operate in a tidal stream speed of 3 m/s with a shaft speed of 12 rpm and a diameter of 20m in full scale (Wang *et al.* 2007). The turbine was scaled to a pitch-controllable model of 400 mm in diameter with a scale ratio of 1 to 50. The blade section S814, developed by D.M. Somers as shown in Figure 2, was adopted for the blade sections along radius (Janiszewska 1996).

The turbine model was manufactured in such a way that pitch values of the blades can be set in a range of -50 to 150 degrees (design pitch angle is zero degrees at R0.7). Main particulars of the stream turbine model are given in Table 2.

Table 1 Emerson Cavitation Tunnel test section particulars

Length	3.1m
Breadth	1.22m
Height	0.81m
Flow speed range	0.5 - 8 m/s
Absolute pressure range	7.6 – 106 kN/m ²
Cavitation number range	0.5 – 23

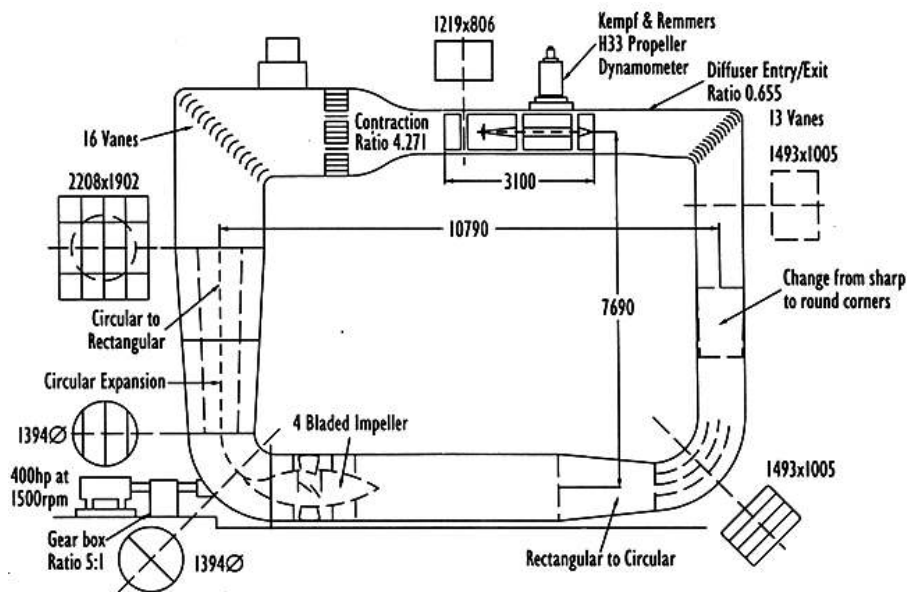


Fig. 1 Dimensional sketch of Emerson Cavitation Tunnel

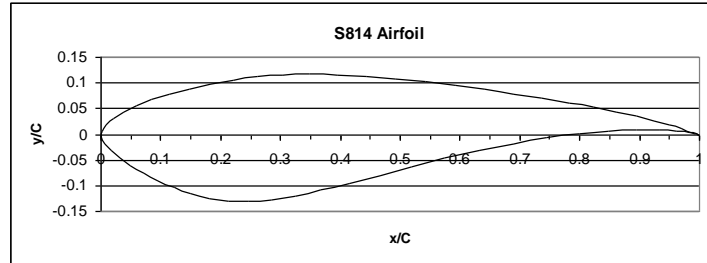


Fig. 2 S814 Airfoil profile

Table 2 Main particulars of the stream turbine model

r/R	0.2	0.25	0.3	0.35	0.4	0.45	0.5	0.6	0.7	0.8	0.9	1
chord (mm)	64.35	62.21	60.06	57.91	55.76	53.62	51.47	47.18	42.88	38.59	34.29	30
twist angle (deg)	27	20	15	11	7.5	5.5	4	2	0.5	-0.4	-1.3	-2

2.3 Test rig layout

The model turbine was mounted on a vertically driven Kempf & Remmers type H33 dynamometer as shown in Fig. 3. This dynamometer was used to measure the thrust and torque of the model turbine. On top of the dynamometer, a three-phase DC motor was mounted to drive a propeller or to absorb the power generated by the turbine and control the rotation rate.



Fig. 3 Marine current turbine model mounted on Kempf & Remmers type H33 dynamometer

The dynamometer was calibrated fully wired with slip rings and dummy weight in place before testing. The responses of the dynamometer were recorded directly from a voltmeter, amplified and acquired on a computer. The water flow speed was acquired using the tunnel systems.

3. Test programme

The objectives of the tests were to derive a set of MCT performance characteristics by measuring power and thrust performance over a range of tip speed ratios under certain conditions. The cavitation observation tests required reduced tunnel pressures to simulate cavitation conditions. The range of required tip speed ratios was achieved with constant water velocity by varying the rotor RPM. For the cavitation observation tests, the water velocity and test section pressure were set at the required values.

3.1 Power extraction tests

To simulate the same loading conditions as that of its full scale, the model turbine must be tested at the same tip speed ratio which is calculated from the following equation

$$\lambda = \frac{\Omega R}{U} = \frac{2\pi n R}{U} \quad (1)$$

where, U is tunnel water speed (m/s); R is the turbine radius (m), Ω is rotational speed (rad/s) and n is rotation rate (revs/s) of the turbine model. The angle of attack or pressure distribution at each blade section of a turbine blade is going to be the same between model and full scale turbine as long as they have the same tip speed ratios (Wang *et al.* 2007).

During the experiment, water speed in tunnel was maintained constant whilst rotational speed of turbine model was controlled to vary in obtaining certain tip speed ratios. At each set of rotational speed, the torque and thrust of the turbine model were measured. Then the power and thrust coefficients were calculated using the equations below

$$C_P = \frac{2\pi n Q}{0.5\rho\pi(D/2)^2 U^3} = \frac{16nQ}{\rho D^2 U^3} \quad (2)$$

$$C_T = \frac{T}{0.5\rho\pi(D/2)^2 U^2} = \frac{8T}{\rho D^2 U^2} \quad (3)$$

here, ρ is the density of water (kg/m^3), D is the diameter of the rotor (m), Q and T are the measured torque (Nm) and the thrust (N) of the turbine, respectively.

3.2 Cavitation observation tests

Though the rotor and propeller are totally different in their application, their hydrodynamic principles are exactly the same. Thus two dynamic similarity laws - the similarities in pressure of the interested point and angle of attack of the blade - should be fulfilled in model testing. Just like the tip speed ratio similarity, the similarity in the pressure is presented by a dimensionless coefficient (i.e., cavitation number), which is defined as:

$$\sigma = \frac{P - P_v}{0.5\rho V^2} \quad (4)$$

where, P is pressure value on the interested point (Pa); P_v is the vapour pressure of the water (Pa); ρ is the density of water (kg/m^3) and V is the velocity at the reference point (m/s). In order to reduce the scale effect on cavitation, the air contents dissolved in the water had to be maintained at the required level

$$\frac{\alpha}{\alpha_s} = \frac{P}{P_0} \quad (5)$$

where, α is the air content dissolved in the water in the pressure (P) of test condition; α_s is the air content dissolved in the water open to atmosphere (P_0) in the saturated condition.

4. Experimental results

4.1 Power extraction tests

The torque and thrust of the rotor are measured in a range of tip speed ratios – at which the rotor will extract energy from water - by means of varying the rotation rate of the rotor through a motor. Torque and thrust of the turbine model were measured at each rpm, then power and thrust coefficients were calculated using Eqs. (2) and (3) respectively. Tests were conducted at 3 m/s of constant water speed and repeated for five times and the results of repeated tests for coefficient of power, C_p , are shown in Fig. 4. Since the power extraction efficiency of a turbine is of higher importance, test results for the thrust generated will neither be presented nor discussed in this paper.

The difficulty of setting and keeping the RPM and exact water velocity in tunnel at each run was the main source of uncertainty while calculating the coefficient of power. Errors caused by data acquisition equipment and calibration error were treated as random uncertainty. Using error analysis techniques, data uncertainty is calculated and presented for 90% confidence interval and presented as error bars.

4.2 Cavitation observation tests

During the cavitation observation tests, dissolved air content of the tunnel water was 35%. The cavitation observations of the model turbine operating at three different test conditions as well as main dimensions and the corresponding test parameters of both the model and full scale turbine are presented in Figs. 5-7. At first cavitation condition below, only very slight thin tip vortexes were observed.

The observed cavitations in the above test conditions resulted in cloudy sheet cavitation covering the leading edge region from 0.5 – 0.9R with a maximum cavity length about 15-20% of the chord length from leading edge as shown in the Fig. 6.

During this particular cavitation observation test, strong cloudy sheet cavitation covered the leading edge region of the blades from 0.5 – 0.9R. The maximum cavity length of the sheet cavitation was about 30% of the chord from the leading edge. At the trailing edge of the sheet

cavity closure slight cloud cavitation, which was bursting into a strong tip vortex, could be observed as shown Fig. 7.

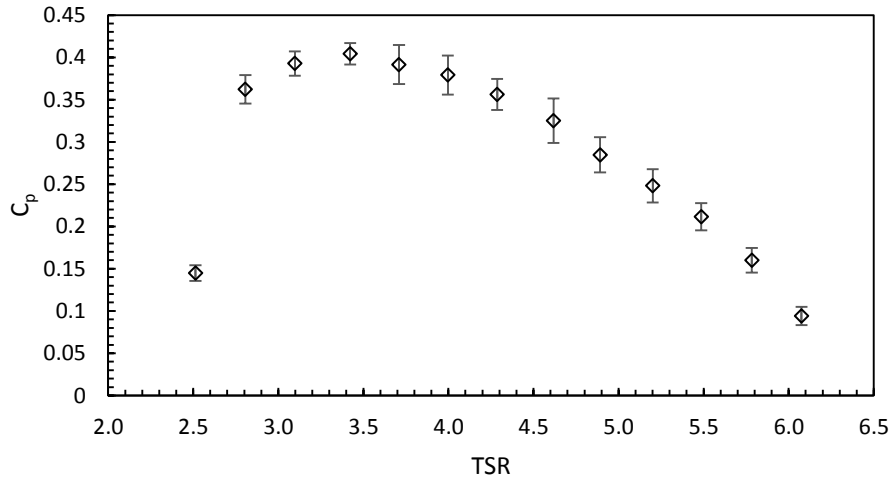


Fig. 4 Measured values of power coefficients for $v=3$ m/s tunnel velocity

In full scale:

Current speed (m/s) : 2.565
 Cavitation number : 2.349
 RPM : 12
 Diameter (m) : 20
 Shaft immersion (m) : 11
 Tip speed ratio : 4.9

In model scale:

Current speed (m/s) : 1.467
 Cavitation number : 2.349
 RPM : 342.9
 Diameter (m) : 0.4
 Tunnel Vacuum (mm-hg) : -435
 Tip speed ratio : 4.9

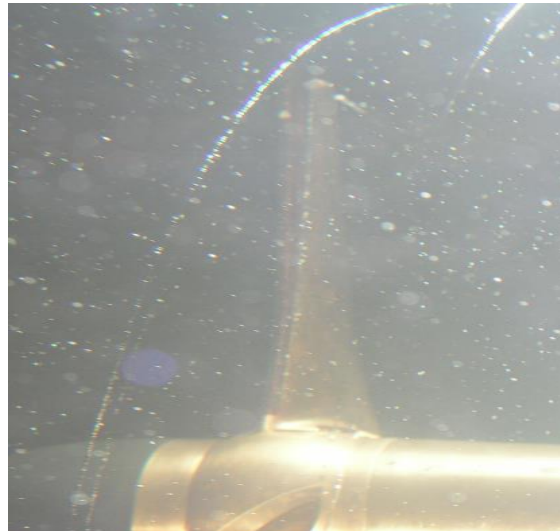


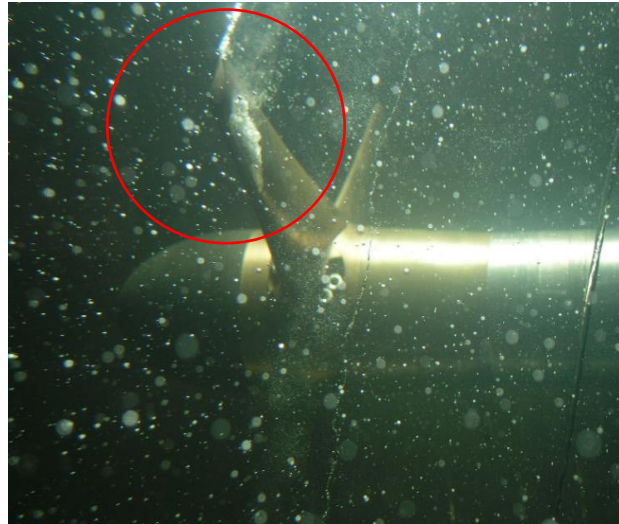
Fig. 5 Cavitation observed at 2.565 m/s current speed and $\sigma=2.349$

In full scale:

Current speed (m/s)	: 3.5
Cavitation number	: 2.232
RPM	: 12
Diameter (m)	: 20
Shaft immersion (m)	: 11
Tip speed ratio	: 3.591

In model scale:

Current speed (m/s)	: 2.0
Cavitation number	: 2.232
RPM	: 342.9
Diameter (m)	: 0.4
Tunnel Vacuum (mm-hg)	: -435
Tip speed ratio	: 3.591

Fig. 6 Cavitation observed at 3.5 m/s current speed and $\sigma=2.232$ ***In full scale:***

Current speed (m/s)	: 4.66
Cavitation number	: 2.061
RPM	: 12
Diameter (m)	: 20
Shaft immersion (m)	: 11
Tip speed ratio	: 2.7

In model scale:

Current speed (m/s)	: 2.66
Cavitation number	: 2.061
RPM	: 342.9
Diameter (m)	: 0.4
Tunnel Vacuum (mm-hg)	: -435
Tip speed ratio	: 2.7

Fig. 7 Cavitation observed at 4.66 m/s current speed and $\sigma=2.061$

5. Mathematical analysis

5.1 Vortex lattice method

The geometry of a turbine blade in the vortex lattice method is defined by the distribution of vortices and sources on the blade mean camber and its assumed wake surface in the same way as

that of marine propellers. The jump of the tangential velocity both at the camber surface and in the trailing wake sheet is represented by the vortex distribution. The blade thickness is represented by the source distribution. By resolving the vorticity vector on the mean camber surface into two arbitrarily assigned components, the vortex distribution is divided into span-wise and chord-wise components on the turbine blade surface. The chord-wise component is referred as trailing vorticity which sheds into the wake. Applying the kinematic and dynamic boundary conditions at certain control points, the unknown strengths of the singularities can be determined by solving an algebraic equation system. The details of the VLM can be found in Lee (1979).

The three dimensional coordinate system used in the method is defined with the positive x being the downstream direction. The y axis is attached to the key blade normal to the x axis and the z axis follows the right hand system as shown in Fig. 8.

The cylindrical coordinate is defined as follows

$$x = x \quad (6)$$

$$r = \sqrt{y^2 + z^2} \quad (7)$$

$$\theta = \tan^{-1} \frac{z}{y} \quad (8)$$

The mid-chord line of the turbine blade is defined by the radial distributions of skew, $\theta_m(r)$, and rake, $x_m(r)$. The coordinates of the leading and trailing edges are constructed by passing a helix of pitch angle $\phi(r)$ through the mid-chord line.

$$x_{l,t}(r) = x_m(r) \mp \frac{c(r)}{2} \sin\phi(r) \quad (9)$$

$$\theta_{l,t}(r) = \theta_m(r) \mp \frac{c(r)}{2} \cos\phi(r) \quad (10)$$

$$y_{l,t}(r) = r \cos\theta_{l,t}(r) \quad (11)$$

$$z_{l,t}(r) = r \sin\theta_{l,t}(r) \quad (12)$$

where $c(r)$ is the chord length at radius r , and the subscripts l and t denote the leading and trailing edges, respectively.

The camber distribution $f(r,s)$ is defined as the normal distance between the sectional mean line and the nose-tail helix on the cylindrical surface of radius r , where s is the chord-wise coordinate with 0 at the leading edge and l at the trailing edge. The thickness $t(r,s)$ is added symmetrically normal to the camber line. The elements used on turbine blades and their wakes in VLM model are given in Fig. 9. 20 vortex lattices ($N = 20$) were used along the chord-wise direction and 18 vortex lattices ($M = 18$) were used along the radius of the blades. The details of the blade geometry definition can be found in Kerwin and Lee (1978).

5.2 Numerical simulation of cavitation on MCT blades

Making use of the data available from the comprehensive study of Bahaj *et al.*, a 800mm diameter model of a horizontal axis marine current turbine operating in 1.73 m/s free stream velocity is modelled via vortex lattice method to achieve the distribution of cavitation over the turbine blades. The standard geometry has a pitch angle at the blade root equal to 15 degrees, corresponding to 0 degree pitch setting at the tip. In the present work, the 20 degrees pitch angle

setting is considered. The detailed geometry of the marine current turbine is given in (Bahaj *et al.* 2007).

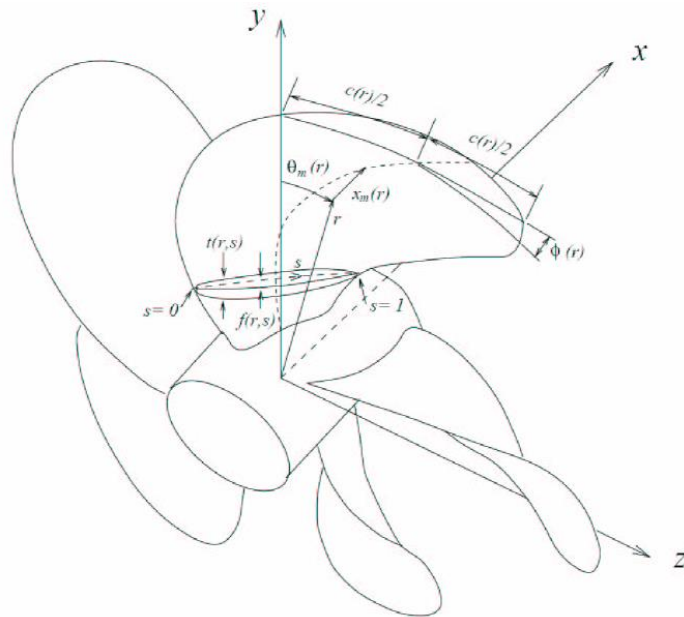


Fig. 8 Coordinate systems and geometrical notations in VLM (Greeley & Kerwin 1982)

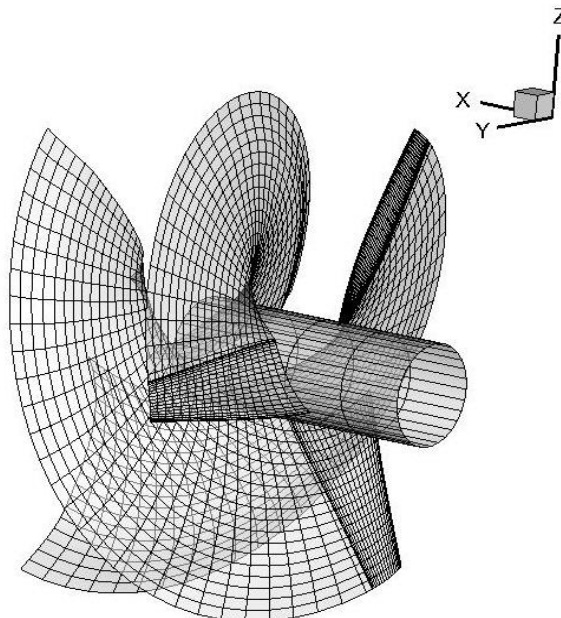


Fig. 9 Elements used on the turbine blades and their wakes in VLM

The distribution of cavitation over the turbine blades was modelled for the case where the cavitation number is 0.64, free stream velocity is 1.4m/s and rpm is 250 (TSR=7.5). Simulation results are compared with the experimental observations and shown in Fig. 10. The sheet cavitation covering the span-wise region between 0.65R to 0.9R along the blades was predicted accurately for this test condition. A satisfactory agreement between experimental result and simulation result has been obtained.

The cavitation developments on the turbine model used in the present work (section 4.2) are simulated for operating conditions defined and presented in Figs. 6 and 7, and the simulation results are compared with the cavitation observations, as shown in Figs. 11 and 12, respectively. The simulations seem to predict larger cavity region than the experimentally observed values. A possible reason of this is that in experimental observations there seems to be a mixture of cloud type of cavitation with the sheet type while the numerical simulations can only predict the sheet type of cavity.

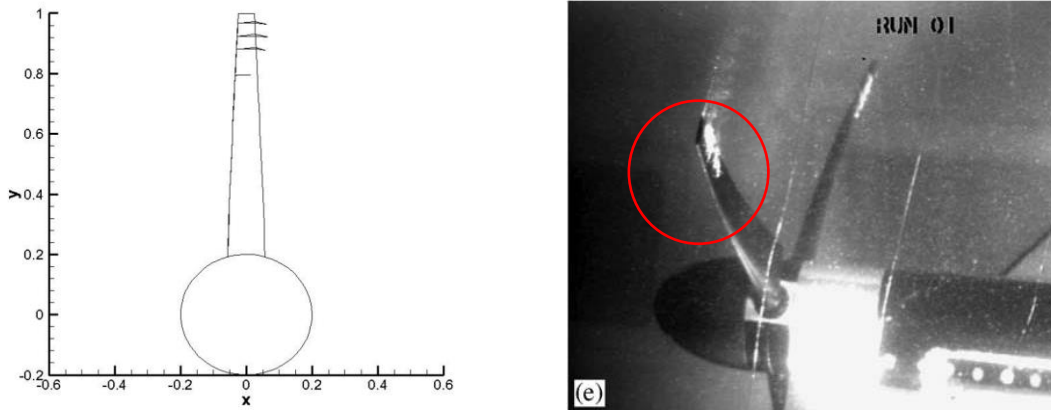


Fig. 10 Comparison of cavitation simulations and observation results for 1.4 m/s free stream velocity, TSR=7.5 and $\sigma=0.64$

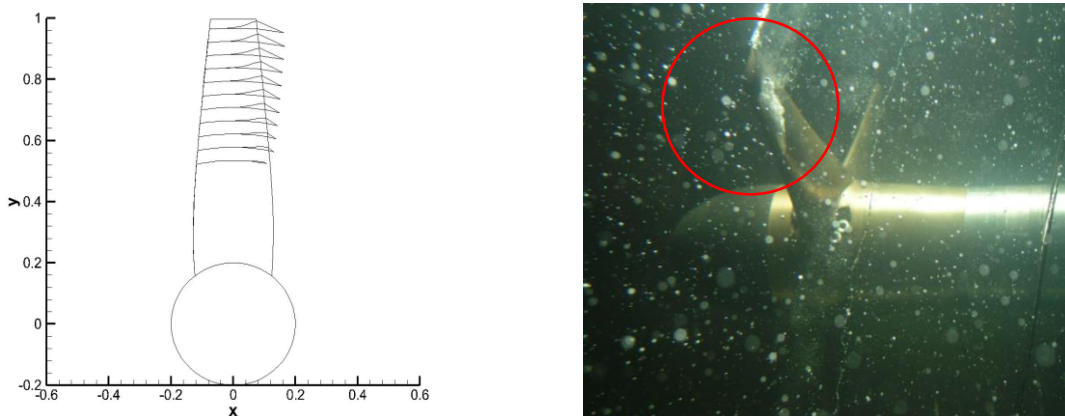


Fig. 11 For 2.0 m/s free stream velocity, TSR=3.591 and $\sigma=2.232$, comparison of mathematical simulation and observation of distribution of cavitation over the turbine blade

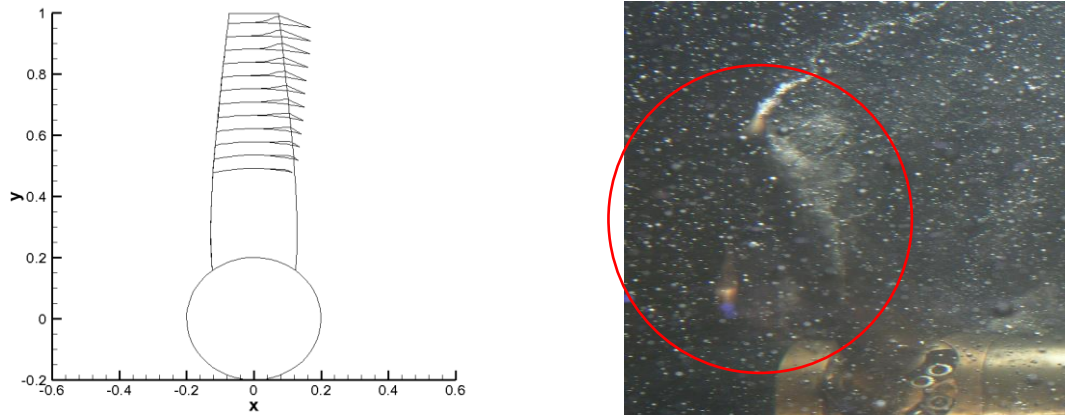


Fig. 12 For 2.66 m/s free stream velocity, TSR=2.7 and sigma: 2.061, comparison of mathematical simulation and observation of distribution of cavitation over the turbine blade

6. Conclusions

This paper presents useful experimental data for the power and cavitation performance of a 3-bladed, horizontal axis current turbine tested in a medium size cavitation tunnel. The cavitation performance data were used to validate a vortex lattice method based numerical tool to predict the sheet cavitation on a turbine. Based on the investigation it was found that:

- Despite some scatter caused during the setting of the shaft speed and tunnel incoming flow velocity, repetitive measurements indicated a peak efficiency for the tested turbine at a TSR of 3.5 with a maximum power coefficient of 0.41. The uncertainty analysis of the measured performance data for the torque and thrust varied between %7.3 to %24.9 and %1.9 to %4.7 respectively for %90 confidence interval.
- Cavitation observations for three different operating conditions showed that, the turbine blades can be exposed to a significant amount of cavitation depending on the severity of the operating conditions. As the free stream velocity increased the model turbine tested suffered from sheet and cloud cavitations, at a region along the blade radius of 0.5 to 0.9, accompanied with a strong tip vortex cavitation.
- Distributions of the cavitation along the blades of the model turbine tested by (Bahaj *et al.* 2007b) was modelled using a vortex lattice method. Mathematical simulations accurately predicted the observation test results, at which the sheet cavitation covered the span-wise region of the blade from 0.65R to 0.9R.
- Cavitation simulations of the model turbine tested in this study predicted larger cavity regions than the experimentally observed ones. A possible reason of this is that the cloud type of cavitation was predicted as sheet type due to restriction of the numerical model.

Due to the difficulties in simulating cavitation both experimentally and mathematically, investigation of the effect of cavitation on hydrodynamic performance of MCT's is rather

challenging. VLM model needs to be improved to simulate tip vortex cavity and parametric cavitation tunnel experiments are needed to accurately validate mathematical predictions.

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