Performance Prediction of H-Type Darrieus Turbine by Single Stream Tube Model for Hydro Dynamic Application

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Abstract

A Hydrodynamic model that can accurately predict the performance of a device can be used to optimize the design parameters more rapidly and at a much lower cost than carrying out the design studies using scale model tests. Tidal turbine models are principally dependent on the Hydrodynamic models developed for wind turbines. These models are crucial for deducing optimum design parameters and also for predicting the performance before fabricating the Darrieus hydro turbine. The method of analysis used is Single stream tube model in that induced velocity is assumed as a constant and it is obtained by actuator disc theory and blade momentum theory. Single stream tube model can predict the coefficient of performance easily before experiment of the turbine. In this paper Comparison of coefficient of power and co-efficient of torque by single stream tube model and experimental result, and its strengths and weaknesses are discussed.

Keywords: Low head power generation, Darrieus turbine, Aerodynamic model, VAWT.

1. Introduction

The Darrieus vertical axis wind turbine concept attracted considerable research interest in the late 1970s and 1980s, but has never competed successfully with horizontal axis wind turbines. In recent years there has been growing interest in Darrieus straight blade hydro turbine for low head application which is based on Darrieus wind turbine.

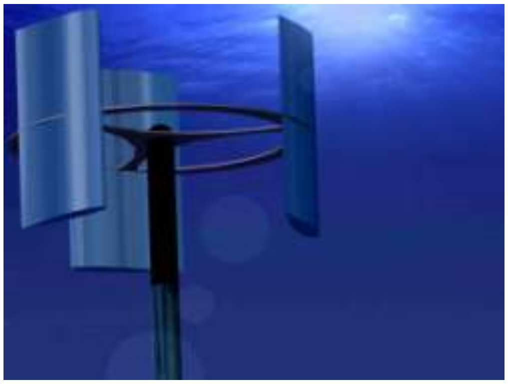

Figure: 1: Darrieus Hydro Turbine

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In 1974 Templin proposed the single stream tube model which is the first and most simple prediction method for the calculation of Aerodynamic performance characteristics of curve blade Darrieus Vertical axis wind turbine [1].Stream tube models are momentum models based on Glauert's Blade element theory[2]. In stream tube models the change in fluid momentum in the flow direction is equated to the stream wise forces on the aerofoil blades.

In this model the entire turbine is assumed to be enclosed within a single stream tube. The objective of the model is to determine the performance coefficient of rotor.

2. Nomenclature

A Projected frontal area of turbine

C Blade chord

C_d Blade drag coefficient

C_{dor} Reference zero-lift-drag coefficient

C_D Turbine overall drag coefficient $= F_D / \rho A V_\infty^2$

C_{DD} Rotor drag coefficient $= F_D / \rho A V_a^2$

C_l Blade lift coefficient

C_n Normal force coefficient

C_P Turbine overall power co-efficient= $P_o / \rho A V_\infty^3$

C_T Turbine overall torque coefficient= $T_B / \rho A V_\infty^2 R$

C_t Tangential force coefficient

D Blade drag force

F_n Normal force

F_t Tangential force

H Height of turbine blade

L blade lift force

N Number of blade

R Turbine radius

Re Local Reynolds number $= \rho W C / \mu$

V_a Induced velocity

V_C or V_t Chordal velocity component

V_n Normal velocity component

V_W Wake velocity

W Relative velocity

ρ Density of fluid

θ Azimuth position of blade

α Angle of attack

3. Axial Induction Factor

The axial induction factor, 'a' is the fractional decrease in water velocity between the free stream and the rotor plane, so it is defined as [11],

$$a = \frac{Nc}{2\pi R} \frac{R\omega}{V_\infty} \sin\theta \qquad (1)$$

We can define induction factor by reference [5],

$$a = \frac{V_\infty - V_a}{V_\infty} \qquad (2)$$

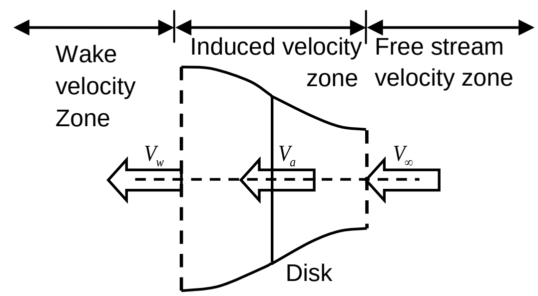

Figure: 2: Actuator disc model for Darrieus rotor.

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The value for V_a can be obtained by Gluert Actuator Disk theory [2], the expression of the uniform velocity through the rotor is,

$$V_a = \frac{V_\infty + V_W}{2} \tag{3}$$

From equation (1) and (2) we can find induced and wake velocity,

$$V_a = V_\infty(1-a) \tag{4}$$

$$V_w = V_\infty(1-2a) \tag{5}$$

3. Blade Angle of Attack and Relative Velocity

The flow velocities in the upstream and downstream sides of the Darrieus rotor are constant as seen in figure 3. From this figure one can observe that the flow is considered to occur in the axial direction. The tangential velocity (or chordal velocity) component V_t (or V_c) in tangential direction of blade profile and the normal velocity component V_n is normal to blade profile. Tangential velocity (V_t) and Normal velocity (V_n) of blade which is given by [3],

$$V_t = R\omega + V_a \cos\theta \tag{6}$$

$$V_n = V_a \sin\theta \tag{7}$$

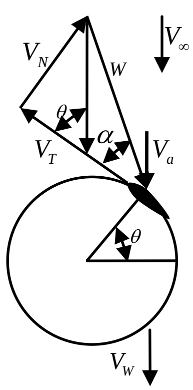

Figure: 3: Flow velocities of Darrieus hydro turbine.

Angle of attack α is angle between relative velocity W and tangential velocity V_t is obtained from the following expressions:

$$\alpha = \tan^{-1}\left(\frac{V_n}{V_t}\right) \tag{8}$$

Substituting the values of (6) and (7) in equation (8), And Non-dimensional zing the equation

$$\alpha = \tan^{-1}\left[\frac{\sin\theta}{(R\omega / V_a) + \cos\theta}\right] \tag{9}$$

The relative flow velocity (W) can be obtained as,

$$W = \sqrt{V_t^2 + V_n^2} \tag{10}$$

Inserting the values of V_t and V_n , and non-dimensional zing, one can find velocity ratio as,

$$\frac{W}{V_a} = \sqrt{\left(\frac{R\omega}{V_a}\right)^2 + 2\left(\frac{R\omega}{V_a}\right)\cos\theta + 1} \tag{11}$$

Local relative water dynamic pressure *(q)* is given by:

$$q = \frac{1}{2}\rho W^2 \tag{12}$$

4. Blade Element Force and Drag Co-Efficient

Since the disk acts as a drag device, the source of drag must be a pressure difference across the disk and this drag manifests itself as thrust loading along the axis normal to the disk. Rewriting N2 in terms of momentum:

$$\sum F_x = \dot{m}(\Delta V) \tag{13}$$

According to glauert's theory [2] the velocity through a wind mill disk V_a is the arithmetic mean of the undisturbed velocity V_∞ and the velocity in the wake. The turbine drag D is given by:

$$D = m(V_\infty - V_a) \tag{14}$$

$$D = 2\rho A V_a (V_\infty - V_a) \tag{15}$$

The disk drag coefficient C_{DD} [4] based on dynamic pressure and the disk area is defined as:

$$C_{DD} = \frac{D}{\frac{1}{2}\rho A V_a^2} \tag{16}$$

And from equation (13) and (14)

$$C_{DD} = 4\left(\frac{V_\infty}{V_a} - 1\right) \tag{17}$$

$$\frac{V_\infty}{V_a} = 1 + \frac{1}{4}C_{DD} \tag{18}$$

For structural design purpose, a more convenient drag co-efficient C_D is based on the ambient dynamic pressure [4]:

$$C_D = \frac{D}{\frac{1}{2}\rho V_\infty^2 A} \tag{19}$$

$$C_D = C_{DD}\left(\frac{V_D}{V_\infty}\right)^2 \tag{20}$$

$$C_D = \frac{C_{DD}}{\left(1 + \frac{1}{4}C_{DD}\right)^2} \tag{21}$$

For a given turbine geometry and rotational speed, turbine power and rotor drag are calculate using the blade element theory [4]. To calculate the blade element forces, the local Hydrodynamic angle of attack and the local relative dynamic pressure are required. The resultant velocity vector, relative vector, relative to the moving blade element, is resolved into two perpendicular components; one parallel to the span wise and other in blade profile plane as shown in figure 4. The velocity component parallel to the blade span wise has no effect on the blade element Hydrodynamic forces.

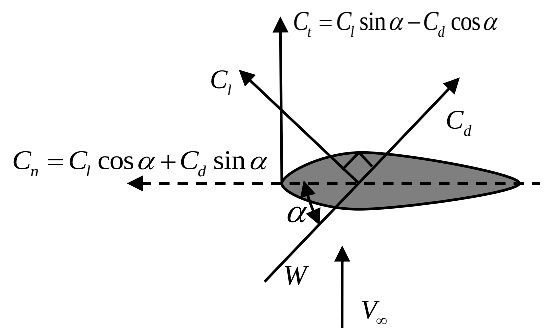

Figure: 4: Normal and Tangential force coefficient on blade

Assuming that the instantaneous local blade element lift and drag coefficients C_l and C_d are function of angle of attack in steady flow, the normal force coefficient C_n and tangential force coefficient C_t are given by following expressions:

$$C_t = C_l \sin\alpha - C_d \cos\alpha \tag{22}$$

$$C_n = C_l \cos\alpha + C_d \sin\alpha \tag{23}$$

The net tangential and normal forces defined as:

$$F_t = C_t \frac{1}{2}\rho C H W^2 \tag{24}$$

$$F_n = C_n \frac{1}{2}\rho C H W^2 \tag{25}$$

Blade element of chord C is subjected to an elemental normal force dN and a forward thrust force dT given by the relation:

$$dF_n = C_n q C H \tag{26}$$

$$dF_t = C_t q C H \tag{27}$$

The elemental drag at any blade position 'θ' as shown in figure 5 is:

$$dD = (dF_n \sin\theta - dF_t \cos\theta)d\theta \tag{28}$$

The total drag value is obtained by integration on a full revolution ($0 \le \theta \le 2\pi$). Thus the total drag of a rotor with N blades of constant chord C is given by the relation:

$$D = \frac{NCH}{2\pi} \int_0^{2\pi} q\left(C_n \sin\theta - C_t \cos\theta\right) d\theta \tag{29}$$

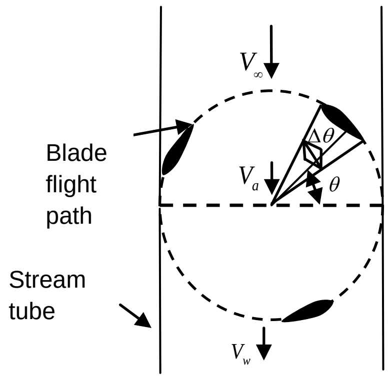

Figure 5: Blade at θ azimuth position.

Drag co-efficient is given by,

$$C_D = \frac{D}{\frac{1}{2}\rho V_\infty^2 A} \tag{30}$$

Put the value of drag D from equation (29).

$$C_{DD} = \frac{NC}{\pi D \rho V_\infty^2} \int_0^{2\pi} q\left(C_n \sin\theta - C_t \cos\theta\right) d\theta \tag{31}$$

For numerical integration of the expression of above equation is divided into a finite number of surface element having equal increments of azimuth angle θ.

5. Rotor Torque and Power Co-Efficient

The rotor torque is produced by only the tangential component of the force on the rotor blade element and in its elemental form, for a single blade element height '*H*' and at $d\theta$ azimuth position of the blade the rotor torque is given by:

$$dT_S = CHRqC_t d\theta \tag{32}$$

Blade torque varies with blade azimuthally angle θ. The total torque for rotor is obtained by integration on a full revolution ($0 \leq \theta \leq 2\pi$).

$$T_B = \frac{NCHR}{2\pi} \int_{\theta=0}^{2\pi} qC_t d\theta \tag{33}$$

Torque co-efficient is defined as:

$$C_T = \frac{T_B}{\frac{1}{2}\rho V_\infty^2 AR} \tag{34}$$

From equation (34) and (33),

$$C_T = \frac{NC}{2\pi\rho R V_\infty^2} \int_0^{2\pi} qC_t d\theta \tag{35}$$

The shaft power is then,

$$P = \omega T_B \tag{36}$$

For straight blade vertical axis rotor of diameter *D* and height *H* only one part of the total wind kinetic energy flow is converted into useful shaft power. Maximum possible power of the swept rotor area is:

$$P_{\max} = \frac{1}{2}\rho V_\infty^3 A \tag{37}$$

The power coefficient C_p can be defined as the ratio of actual power *P* given by equation to the maximum value $P_{\max}$ given by [5],

$$C_p = \frac{P}{P_{\max}} \tag{38}$$

Put value of $P_{\max}$ and *P* in above equation,

$$C_p = \left(\frac{NC\omega}{2\pi\rho V_\infty^3}\right) \int_0^{2\pi} qC_t d\theta \tag{39}$$

Coefficient of power for rotor is obtained by integration on a full revolution ($0 \leq \theta \leq 2\pi$). The relation with coefficient of torque and coefficient of power are given by:

$$C_P = \lambda C_T \tag{40}$$

We can find co-efficient of power from above equation for a given turbine geometry and for each specified value of tip speed ratio $R\omega / V_a$

5. Experimental Setup

The set-up consists of a Darrieus rotor and structure. The structure is fabricated using studs and mild steel plates. Geometric details of the rotors considered in the present study are given in Table. 1. Darrieus rotor geometry detail is given. Calculation of performance coefficient is based on data for both blade profiles.

Table: 1: Darrieus rotor geometry detail

Sr No.	Blade Profile	D (mm)	H (mm)	C (mm)	V (m/s)
1	NACA 0018	250	150	100	0.46
2	NACA 4415	370	150	100	0.395

The hydrofoils used in turbine rotors are made from rapid prototype machine. Fibber Reinforced Plastic (FRP) is used to make hydrofoils. The mild steel plates are held in place by means of washers and nuts for easy replacement of different diameter rotors using same structure. Rotors are mounted on two bearings (UC 204, NTN make). A self-aligned bearing is used as upper side support, to avoid unwanted frictional torque arising due to minor misalignment between upper and lower rotor support shaft. A rope brake dynamometer type arrangement is used for measuring torque and subsequently power developed by Darrieus rotor. The accuracy of spring balance used is 10gms. The ultrasonic flow meter used to measure volumetric flow rate of water though the channel. Velocity of water is calculated using the continuity equation by using cross sectional flow area. Cross sectional flow area is measured by scale with least count of 1 mm. Details of the open channels used in the present study are given in Table.

Table 2: Details of the open channels used

Channel	Channel Width (mm)	Water level (mm)	Average velocity (m/s^2)	Blade Profile used
1	750	300	0.46	NACA 0018
2	750	300	0.395	NACA 4415

To take care about same frictional condition, bearings are always washed in diesel before mounting in experimental set-up. This exercise is carried out at the start of the experiment to maintain same mechanical efficiency for every experimental result. Bearings are periodically lubricated by lubricating oil. Single unit rotor experiments are performed with constant velocity given in table.2 for above NACA profiles are used for study effect tip speed ratio on co-efficient of power and co-efficient of torque. Water level is maintained such that the rotor is always in submerged condition (i.e., just above the top plate of the rotor). To measure the torque, the rotor is loaded gradually to record spring balance reading and angular speed of the rotor. Mostly, each result is repeated thrice.

5 Results and Discussion

5.1 Angle of Attack

Here Angle of attack α for Darrieus rotor blade is calculated for different tip speed ratio between 1 and 2 for different azimuth angle of blade between 0 to 2π. Blade profile of rotor is taken as NACA0018 for calculation of angle of attack and Geometry detail and velocity is given in table 1. Figure 6 shows the variation in angle of attack α as the azimuthally position θ of blade for different tip speed ratio. The angle of attack become smaller as the tip speed ratio increases; this can be appreciated in the following graph of the angle of attack variation of a blade in a full revolution.

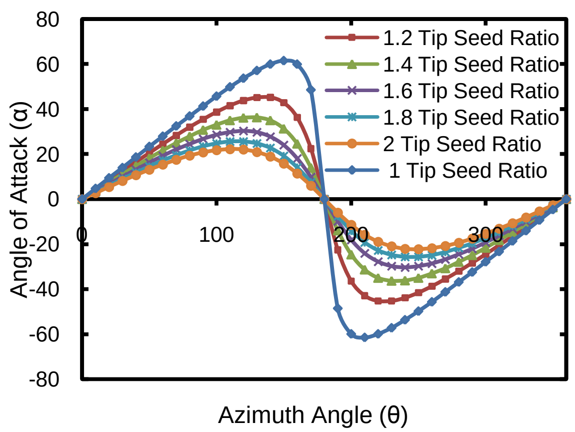

Figure: 6: Variation of Angle of attack for Different Azimuth angle for Different Tip Speed ratio.

5.2 Lift Co-efficient

Figure 7 shows lift coefficient versus different angle of attack for different Reynolds number. Here lift coefficient drawn for Blade azimuth position from 0 to 180 degree for blade profile NACA0018. For blade azimuth position from 30 to 180 degree there is no change in the lift of the blade is shown in figure 7.

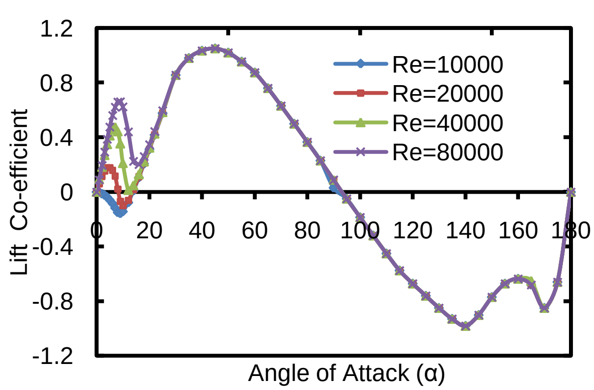

Figure: 7: Variation of Lift Co-efficient for different Angle of attack for Different Reynolds Number.

Lift coefficient for azimuth position of blade of 0 to 30 degree there is more variation in the lift of the blade which is shown in figure 8. When the Reynolds number increases there is increment in the lift coefficient of the blade.

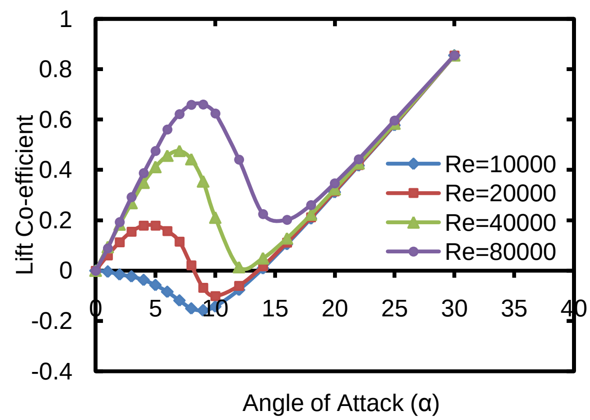

Figure: 8: Variation of Lift Co-efficient for different Angle of attack for Different Reynolds Number.

5.3 Drag co-efficient

Figure 9 shows Drag coefficient versus different angle of attack for different Reynolds number. Here, drag coefficient drawn for Blade azimuth position from 0 to 2π degree for blade profile NACA0018.

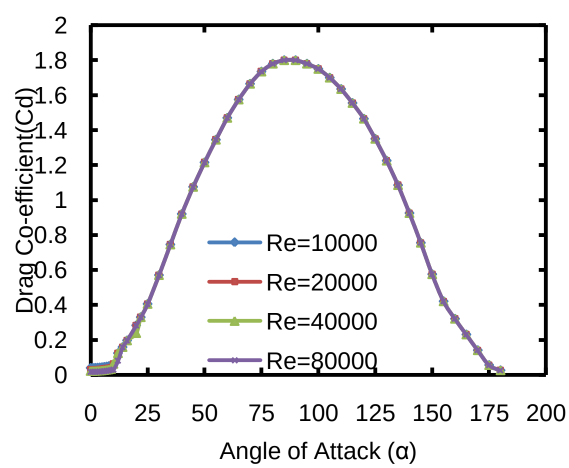

Figure: 9: Variation of Drag Co-efficient for different Angle of attack for Different Reynolds Number.

It is noted that Blade azimuth position from 15 to 180 degree there is no change in the drag of the blade but at azimuth position of blade for 0 to 15 degree there is more variation in the drag of the blade. When the Reynolds number increases there is increment in the drag coefficient of the blade that shown in figure 10.

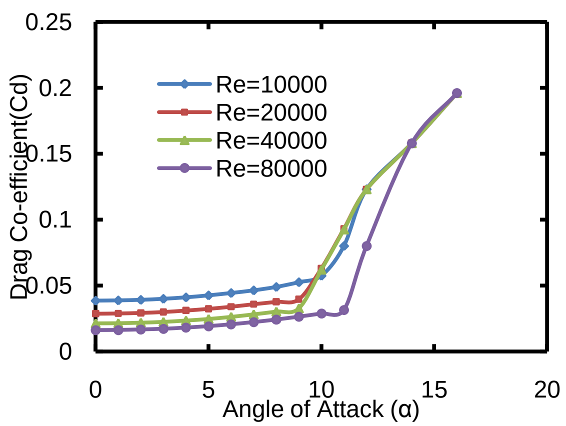

Figure: 10: Variation of Drag Co-efficient for different Angle of attack for Different Reynolds Number.

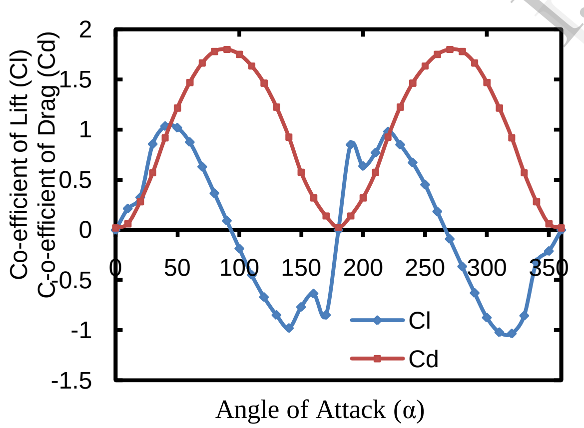

Figure: 11: shows the comparison of the lift and drag coefficient with reference to the angle of attack

Lift and drag coefficient for blade profile NACA0018 for Reynolds number of 40000 is drawn with reference to the angle of attack. As the angle of attack increases, lift and drag coefficient also increase but after 50 degree angle of attack lift is decreasing and drag is increasing as shown in figure10.

5.4 Comparison between Tangential and Normal force coefficient

Here, the variation of tangential and normal force coefficient with reference to the angle of attack is drawn for bale azimuth position between 0 to π degree and Reynolds number of 40000 for blade profile is NACA0018.

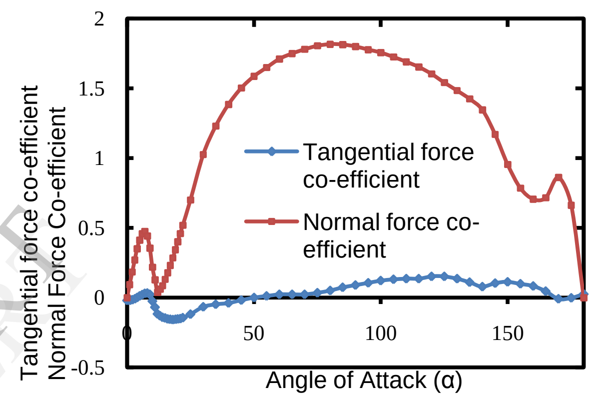

Figure: 12: Comparison of Variation of Tangential and Normal force coefficient for variation of Angle of attack.

5.5 Torque coefficient

Experiment is done for torque coefficient for different tip speed ratio for two different NACA profile. The graph of torque coefficient for two NACA profile, NACA0018 and NACA4415, with reference to the tip speed ratio and with their experimental result is shown. Geometrical data and velocity for both blade profiles are given in table 1. Torque coefficient is accurately predicted by single stream tube model for lower tip speed ration but when the tip speed ratio increases there is more variation in the experimental and analytical result which is shown by figure 13 and figure 14.

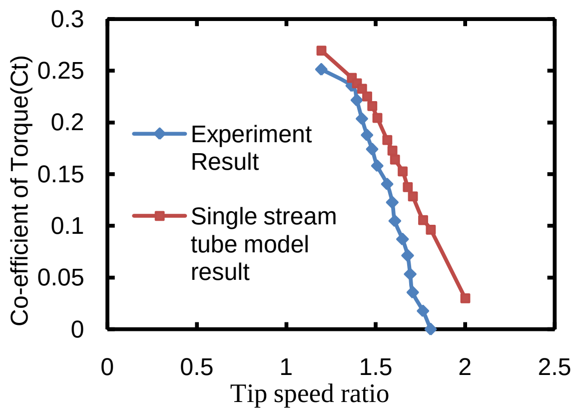

Figure: 13: Comparison of Experimental and Single stream tube result of for NACA0018

In figure 13 and 14 Variation in torque coefficient with reference to tip speed ratio and their comparison with the experimental result is shown.

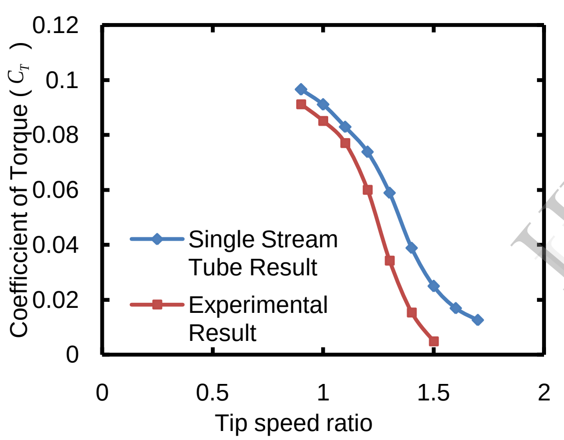

Figure: 14: Comparison of Experimental and Single stream tube result of for NACA4415

5.6 Power coefficient

Figure 1.15 and 1.16 shows the comparison of power coefficient between experimental result and analytical result with reference to tip speed ratio for two NACA profile, NACA0018 and NACA 4415. Power coefficient is accurately predicted by single stream tube model for lower tip speed ration but when the tip speed ratio increases there is more variation in the experimental and analytical result.

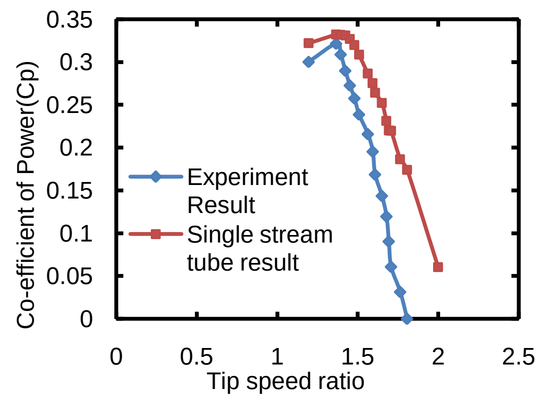

Figure: 15: Comparison of Experimental and Single stream tube result of for NACA0018

Here, maximum power coefficient developed for blade profile NACA0018 and NACA4415 is 0.3322 and 0.091241042 accordingly, but their experimental result is 0.3215 and 0.0845 accordingly which is more than single stream tube result.

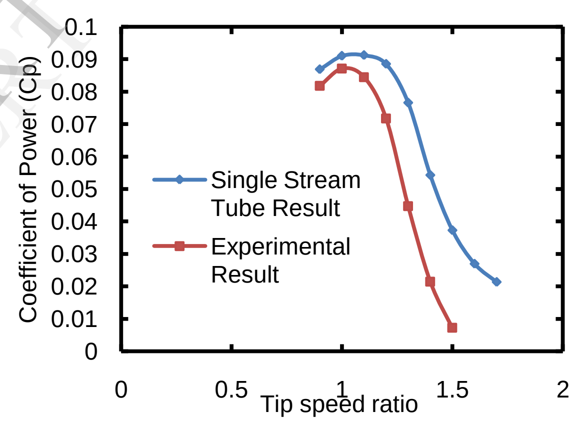

Figure: 13: Comparison of Experimental and Single stream tube result of for NACA4415

6. Conclusion and future work

Single stream tube model is simple prediction of coefficient of performance of rotor. Time require for calculation is less. This model can predict the overall performance of low tip speed ration of rotor and a lightly loaded turbine but according to the inquest, it always predicts higher power than the experimental results. It does not predict the water velocity variations across the rotor. These variations gradually increase with the increase of the blade solidity and tip speed ratio. Single stream tube model does not take into account the difference in the induced velocities between the upstream and downstream halves of the rotor or any difference in velocities across the rotor such as those due to water shear. The main drawback of these models is that they become invalid for large tip speed ratios and also for high rotor solidities because the momentum equations in these particular cases are inadequate.

In future work we can develop another model based on single stream tube model which can be accurately predict the performance coefficient for darrieus rotor for higher tip speed ratio and higher solidity rotor.

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