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A Theoretical Investigation of the Design of a Wind Powered Water Pumping-set for Bangladesh

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ABSTRACT

The aim of this study is to design a wind powered water pumping set suitable for irrigation in Bangladesh. First, considering the average wind velocity at Chittagong, the blade configuration of the windmill for twist and chord is determined by applying momentum theory and blade element theory assuming zero drag and no tilting or coning. Then considering the designed power of the turbine, a suitable piston pump has been designed. The behaviour of the pump and also that of the rotor coupled with the pump are studied.

LIST OF SYMBOLS

a	axial interference factor
a'	tangential interference factor
A	turbine disc area
B	number of blades
C	chord of the blade
C_D	blade element drag coefficient
C_L	blade element lift coefficient
C_{Ld}	design lift coefficoent
C_P	power coefficient
C_Q	torque coefficient
C_T	thrust coefficient
C_Q	starting torque coefficient
D	drag
D_p	diameter of the piston
d	diameter of leak hole
f	friction factor
L	lift force
P	turbine power
P_{hydro}	hydraulic power
Q	torque
R	rotor radius
S	stroke length of the pump
T*	number of pumps
T	thrust force
V_d	design wind velocity
V_∞	undisturbed wind velocity

Greek Symbols

α	angle of attack
α_T	tilt angle
λ_d	design tip speed ratio
ρ	air density
ρ_w	water density
θ_r	pitching angle
ϕ	angle of relative wind velocity
η	efficiency
η_{mech}	mechanical efficiency
η_{vol}	volumetric efficiency
∇_S	stroke volume ($\pi D_p^2 S/4$)
Ω	angular velocity of rotor
Ω^*	angular velocity of pump
β	coning angle
β_t	blade twist angle
γ	yawing angle

1. INTRODUCTION

Modified Blade Element Theory or Strip Theory which is used for the present analysis is the most frequently used theory for the design and performance analysis of a horizontal axis wind turbine. Throughout this study NACA 4418 airfoil section has been used for the blade section of the windmill. If a pump is coupled to a wind rotor which turns at a speed such that the mechanical power of the rotor must be equal to the mechanical power exerted by the pump. The working point can be found out by intersection of the rotor curve and the pump curve.

2. CALCULATION SCHEME

The calculation scheme for the present study is as follows :

a) The choice of basic parameters such as number of blades, the radius of rotor, the types of airfoil and the design tip speed ratio for the windmill.

b) The calculation of the blade twist and the chord at a number of stations along the length of the blade in order to produce maximum power at a given tip speed ratio by every section of the blade.

c) Design of a piston pump.

2.1 CHOICE OF ROTOR PARAMETERS

For the design of a wind rotor, a design tip speed ratio is to be chosen. The general rule is that for the lower design tip speed ratio a higher number of blades

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is chosen and for a higher design tip speed ratio a lower number of blades is chosen. Because choice of higher number of blades for high design tip speed ratio will lead to very small and thin blades which results in manufacturing problems.

To obtain the optimum configuration each blade of the rotor is divided into a number of radial stations. Four formulas [Islam (1986)] will be used to determine the blade twist and chord distribution along the length of the blade.

For local tip speed ratio

$$\lambda = \lambda_d \frac{r}{R} \qquad (2.1)$$

Relation for flow angle

$$\lambda_r = \frac{\sin\theta\,(2\cos\theta - 1)}{(1-\cos\theta)(2\cos\theta + 1)} \qquad (2.2)$$

where

$$\theta = \tfrac{2}{3}\tan^{-1}\frac{1}{\lambda_r}$$

For twist angle:

$$\beta_T = \phi - \alpha \qquad (2.3)$$

For chord :

$$C = \frac{8\pi r\,(1-\cos\theta)}{BC_{Ld}} \qquad (2.4)$$

The blade starting torque can be claculated by [1]

$$Q_{st} = \tfrac{1}{2}\rho v_\alpha^2 B \int_0^R C(r) C_l [\,90 - \beta_T(r)]\, r dr \qquad (2.5)$$

The rotor configuration is determined using the assumption of zero drag and without any tip loss. Each radial element is optimised independently by a continously varying chord and twist angle to obtain maximum energy extraction.

2.2 AERODYNAMIC FORCES :

After a wind turbine rotor is optimally designed, the aerodynamic forces and moments may be calculated. These forces and moments are obtained by applying the blade element and momentum theories. The different velocity components acting on a rotor blade element is shown in figure 1.

The components of relative velocity W can be expressed by [Kamal (1988)]

$$W_x = V_{\infty}\cos\gamma\cos\theta_k + V_{\infty}\sin\gamma\sin\theta_k\sin\alpha_T - \Omega r\cos\beta(1+a') \qquad (2.6)$$

$$W_y = V_{\infty}\{\sin\gamma\cos\alpha + \cos\beta(1-a)\} + \sin\beta\sin\theta_k\cos\gamma - \sin\gamma\sin\beta\sin\alpha + \cos\theta_k \qquad (2.7)$$

The local angle of attack a is defined as

$$\alpha = \phi - \beta_T = \tan^{-1}\frac{W_y}{W_x} - \beta_T \qquad (2.8)$$

The expressions for thrust, torque and power coefficients are given by [Kamal (1988)]

$$C_T = \frac{8}{\pi R^2}\cos^2\beta\cos^2\alpha_T\sin^2\gamma \int_0^{2\pi}\int_0^R \left(\frac{V_{\infty o}}{V_\infty}\right)^2 aF(1-aF)\left(1+\frac{C_D}{C_L}\tan\phi\right) r dr d\theta \qquad (2.9)$$

$$C_Q = \frac{8\Omega}{\pi R^3 V_\infty^2}\cos\alpha_T\cos\gamma\cos^4\beta \times \int_0^{2\pi}\int_0^R V_{\infty o} r^3 a' F(1-aF)\left(1-\frac{C_D}{C_L}\frac{1}{\tan\phi}\right) dr d\theta \qquad (2.10)$$

$$C_P = \frac{8\Omega^2}{\pi R^2 V_\infty^3}\cos\alpha_T\sin\gamma\cos^4\beta \times \int_0^{2\pi}\int_0^R V_{\infty o} r^3 a' F(1-aF)\left(1-\frac{C_D}{C_L}\frac{1}{\tan\phi}\right) dr d\theta \qquad (2.11)$$

3. DESIGN OF A ROTOR COUPLED PUMP

There exists a large variety of water lifting devices. The scope of this study is confined to pumps that can be driven by wind rotors with a focus on the reciprocating pumps. A reciprocating piston pump basically consists of a piston, two valves, a suction pipe and a delivery pipe. Sometimes air chambers are utilized to smooth the flow and to reduce shock forces.

3.1 Expressions of Force, Torque and Discharge

The force on the piston is equal to the weight of the water column acting upon it. The expression for this can be written as

$$F_p = \rho_\omega g H \tfrac{\pi}{4} D_p^2 \qquad (3.1)$$

Here H is the static head but later on the extra head required to cover the losses has to be added. The ideal torque which is sinusoidal can be expressed as

$$Q_{id} = \rho_\omega g H \tfrac{\pi}{4} D_p^2 * \tfrac{1}{2} S \sin\Omega t \quad \text{for } 0 < \Omega t < \pi$$

The expression for the average torque becomes

$$Q_{id} = \frac{1}{2\pi}\rho_\omega g H \tfrac{\pi}{4} D_p^2 \tfrac{1}{2} S \qquad (3.2)$$

The net power supplied by the rotor pump combination must be equal to the hydraulic power to lift the water.

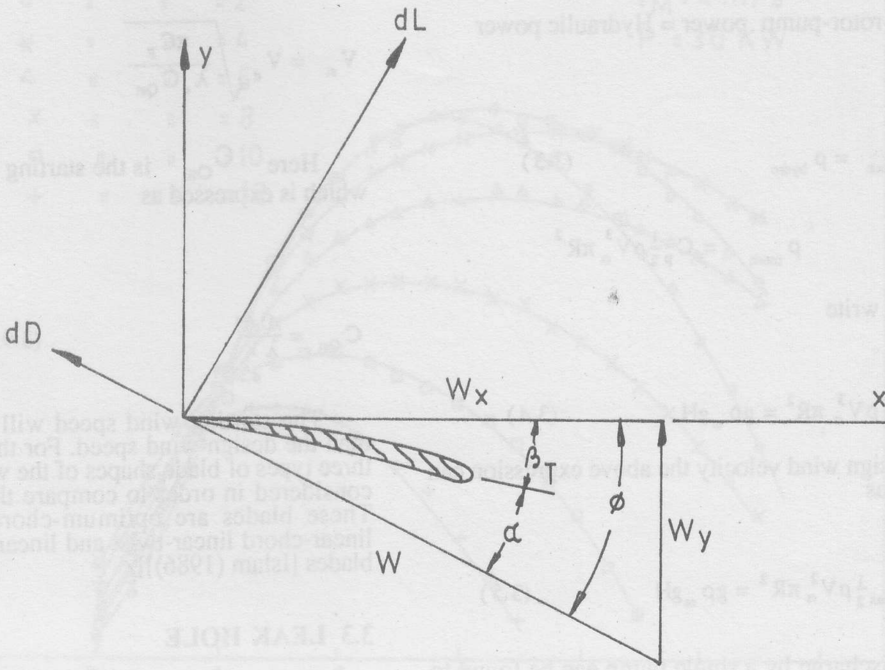

Figure 1 : Velocity diagram for rotor blade element.

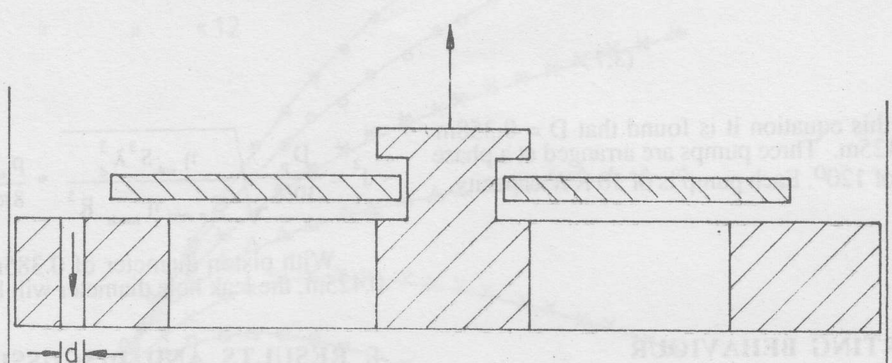

Figure 2 : Schematic drawing of a piston with a leakhole.

i. e. Total rotor-pump power = Hydraulic power

Now

$$\eta_{mech}\,\rho_{mech} = \rho_{hydro} \qquad (3.3)$$

Since $\rho_{mech} = C_p \frac{1}{2}\rho V_\alpha^3 \pi R^2$

So we can write

$$\eta_{mech} C_p \frac{1}{2}\rho V_\alpha^3 \pi R^2 = g\rho_\omega gH \qquad (3.4)$$

For the design wind velocity the above expression can be written as

$$\eta_{mech} C_{pmax} \frac{1}{2}\rho V_\alpha^2 \pi R^2 = g\rho_\omega gH \qquad (3.5)$$

The discharge by a single pump can be found to be

$$q = \eta_{vol} S \frac{\pi}{4} D_P^2 \frac{\Omega}{2\pi} T \qquad (3.6)$$

In this study, design wind velocity has been chosen to be 4m/s which is the average wind velocity at Chittagong and the highest in Bangladesh (Table - 1). For a design tip speed ratio of 8, the radius of the rotor is found to be 23.27 m. For optimum design the relationship between stroke and diameter of the pump is given by [Karassik et al]

$$\frac{S}{D_0} = 1.1 \qquad (3.7)$$

From this equation it is found that D = 0.358m and S = 0.425m. Three pumps are arranged at a phase difference of 120^0. Each pump is of 10 KW capacity.

3.2 STARTING BEHAVIOUR

The starting behaviour of a water pumping windmill is the static description in which the starting torque of the rotor is equal to the maximum torque required by the pump at the starting wind speed.

The starting wind velocity is given by [Jansen (1977)]

$$V_{st} = V_d \sqrt{\frac{\pi C_p}{\lambda_d C_{Qst}}} \qquad (3.8)$$

Here C_{Qst} is the starting torque co-efficient which is expressed as

$$C_{Qst} = \frac{0.6}{\lambda_d^2} \qquad (3.9)$$

The starting wind speed will usually be higher than the design wind speed. For the present analysis, three types of blade shapes of the windmill have been considered in order to compare the starting torque. These blades are optimum-chord optimum-twist, linear-chord linear-twist and linear -chord zero-twist blades [Islam (1986)]].

3.3 LEAK HOLE

A small hole is drilled in the piston of the pump in order to improve the starting characteristics of windmill equipped with a reciprocating pump. The effect of the leak hole is that at very low speeds i.e. when starting, all the water that could be pumped is leaking through the hole. this implies that the pressure on the piston is very low and as a result the starting torque required is low. If the speed is high, the quantity of water leaking through the hole is small compared to the normal output of the pump and the pump behaves as a normal piston pump. The schematic drawing of a piston with a leakhole is shown in figure 2.

The expression for leak hole diameter can be written as

$$d^2 = \frac{D_p^3}{30.8}\sqrt{\frac{\eta_{vol} S^3 \lambda_d^3}{C_{p\,max}\,\eta_{mech} R^5}} * \frac{\rho_\omega f}{8\pi\rho} \qquad (3.10)$$

With piston diameter of 0.385m and a stroke of 0.425m, the leak hole diameter will be 8.17mm.

4. RESULTS AND DISCUSSIONS

The effect of pitch angle on calculated power and thrust for the wind turbine can be seen in figures 3 and 4. Increased pitch angle reduces maximum power but can increase the power available at low tip speed ratios. At low tip speed ratio the power coefficient is strongly influenced by the maximum lift coefficient. From figure 4, it is found that teh rotor thrust

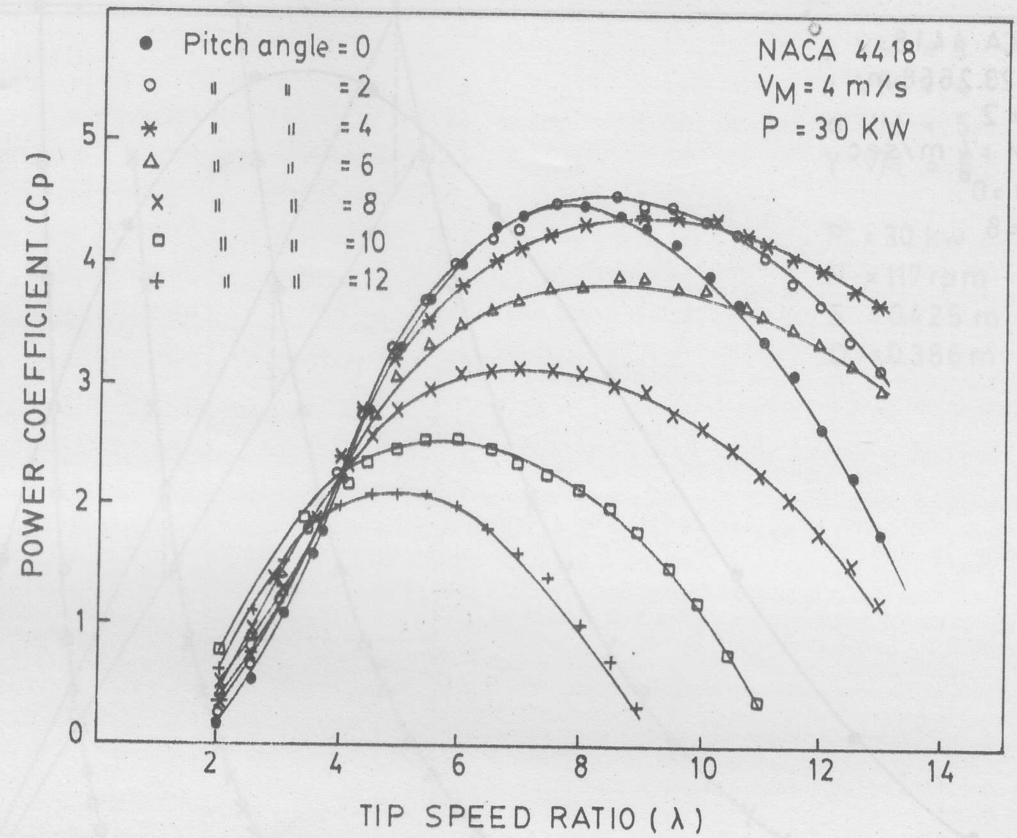

Figure 3 : Variation of power coefficient with tip speed ratio.

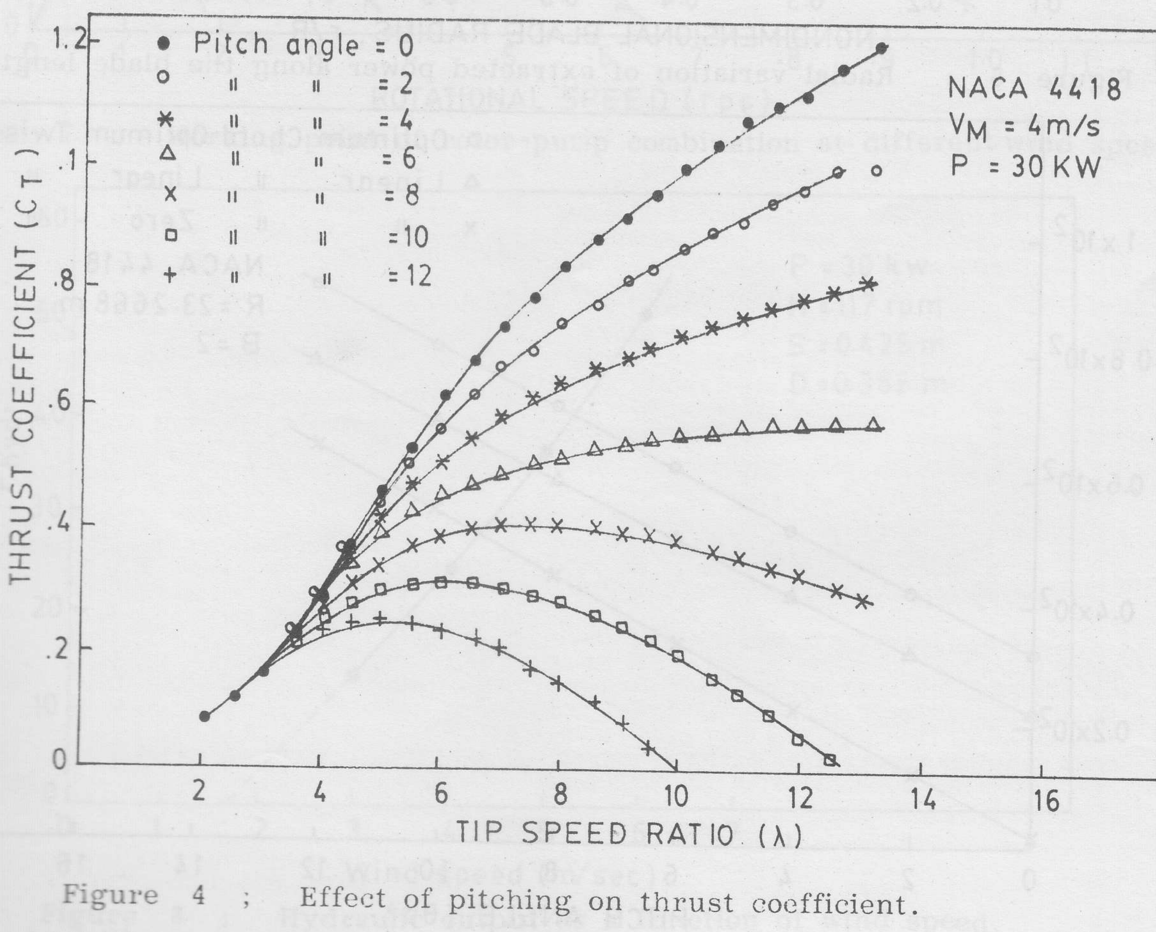

Figure 4 ; Effect of pitching on thrust coefficient.

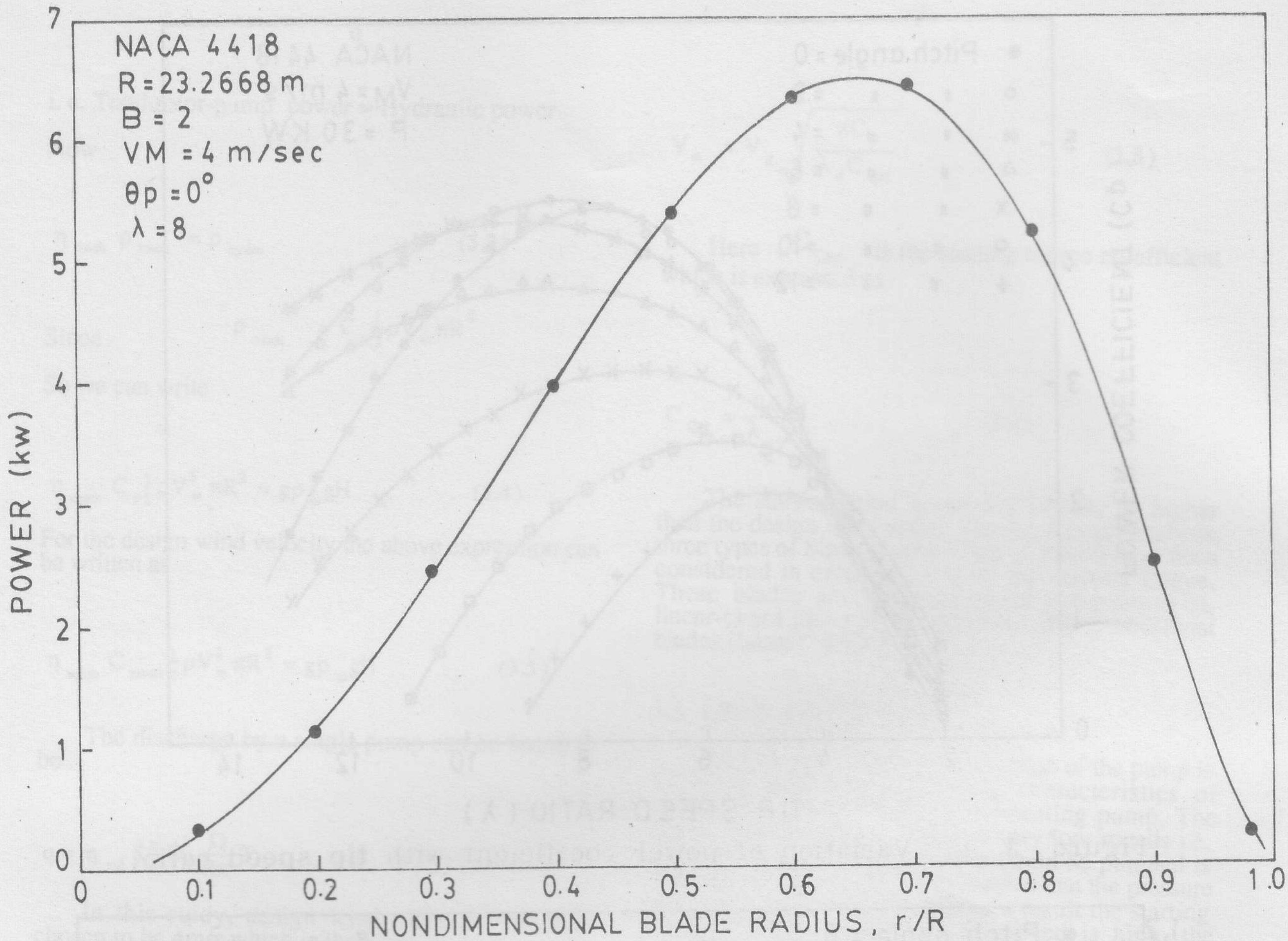

Figure 5 : Radial variation of extracted power along the blade length.

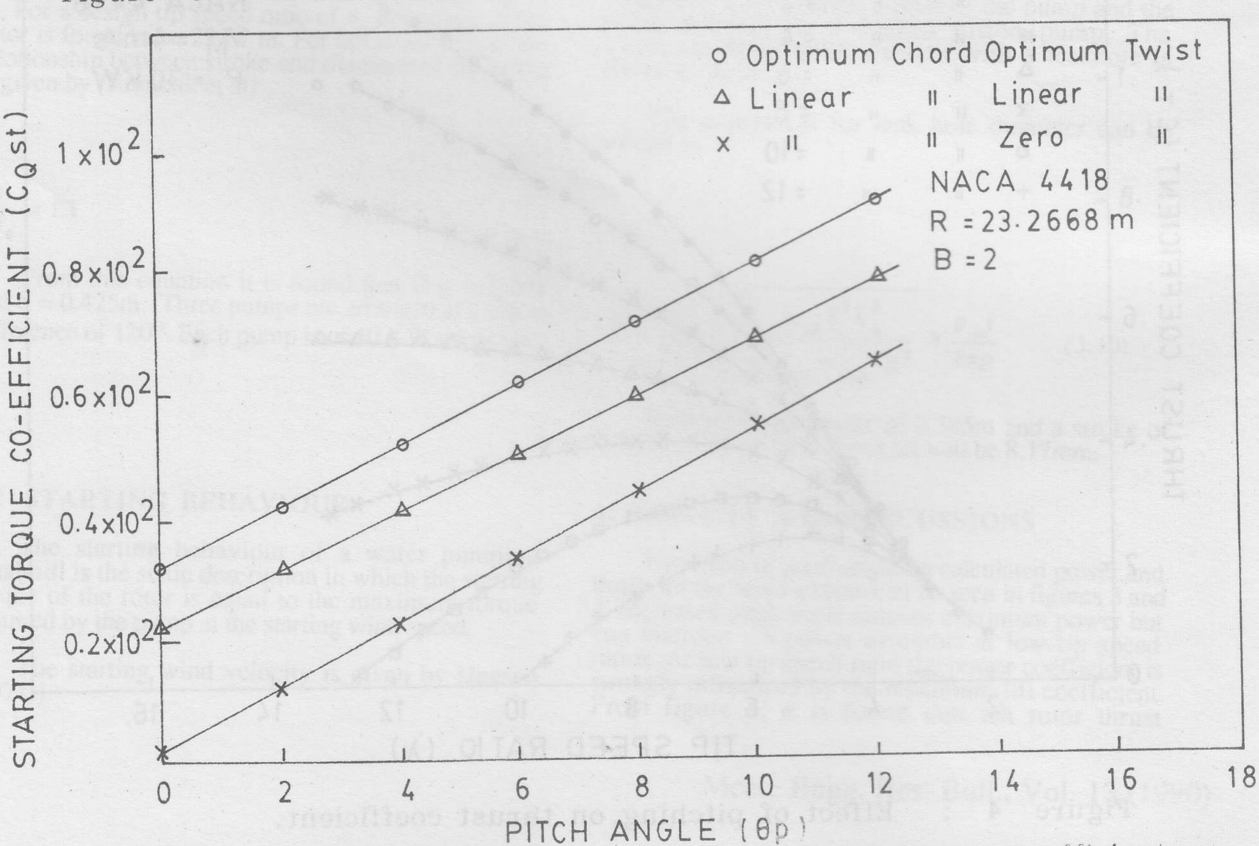

Figure 6 : Effect of blade shapes on starting torque coefficient.

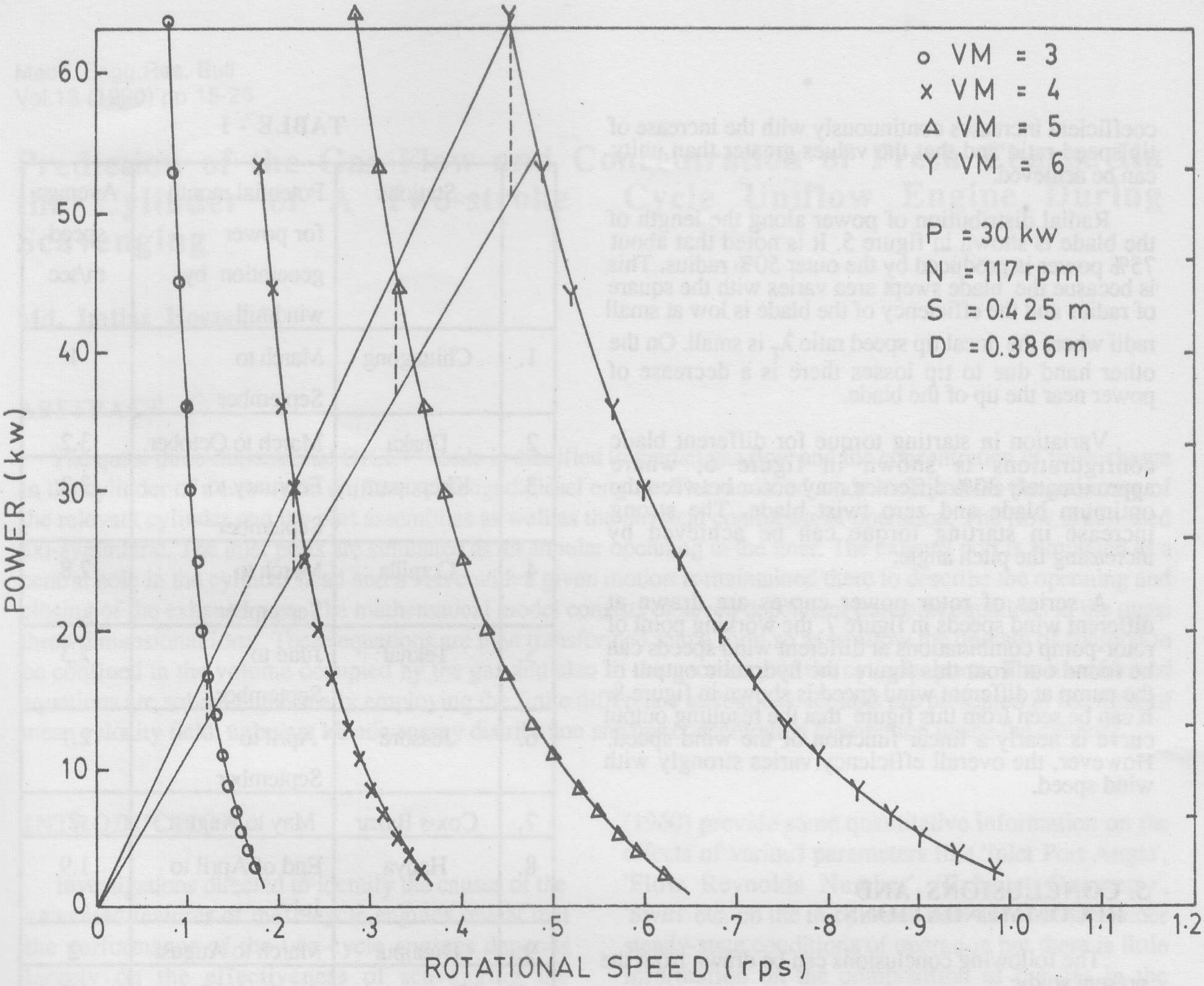

Figure 7 : Working point of rotor-pump combination at different wind speed.

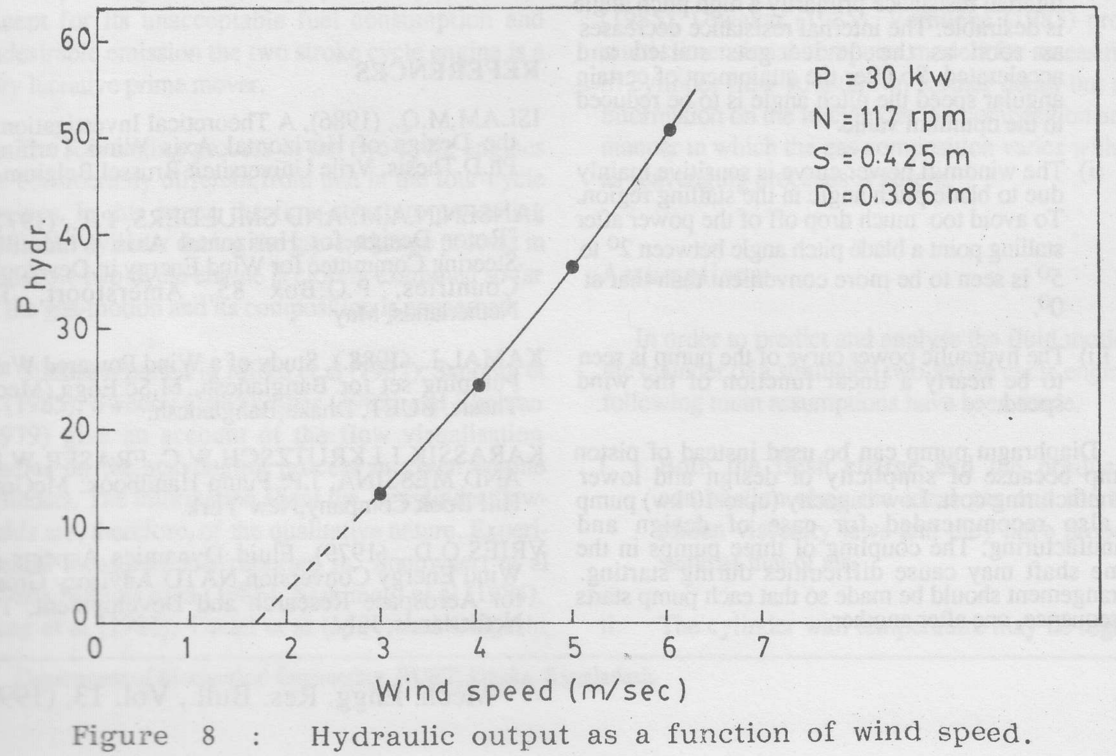

Figure 8 : Hydraulic output as a function of wind speed.

coefficient increases continuously with the increase of tip speed ratio and that the values greater than unity can be achieved.

Radial distribution of power along the length of the blade is shown in figure 5. It is noted that about 75% power is produced by the outer 50% radius. This is becasue the blade swept area varies with the square of radius and the efficiency of the blade is low at small radii where the local tip speed ratio λ_T is small. On the other hand due to tip losses there is a decrease of power near the tip of the blade.

Variation in starting torque for different blade configurations is shown in figure 6, where approximately 30% differnce may occur between the optimum blade and zero twist blade. The strong increase in starting torque can be achieved by increasing the pitch angle.

A series of rotor power curves are drawn at different wind speeds in figure 7. the working point of rotor-pump combinations at different wind speeds can be found out from this figure. the hydraulic output of the pump at different wind speed is shown in figure 8. It can be seen from this figure that the resulting output curve is nearly a linear function of the wind speed. However, the overall efficiency varies strongly with wind speed.

5. CONCLUSIONS AND RECOMMENDATIONS

The following conclusions can be drawn from the present study:

i) To start a low speed rotor that has a high internal resistance primarily a high pitch angle is desirable. The internal resistance decreases as soon as the device gets started and accelerated. So after the attainment of certain angular speed the pitch angle is to be reduced to the optimum value.

ii) The windmill power curve is sensitive mainly due to blade pitch angle in the stalling region. To avoid too much drop off of the power after stalling point a blade pitch angle between 2^o to 5^o is seen to be more convenient than that at 0^o.

iii) The hydraulic power curve of the pump is seen to be nearly a linear function of the wind speed.

Diaphragm pump can be used instead of piston pump because of simplicity of design and lower manufacturing cost. Low capacity (upto 10 kw) pump is also recommended for ease of design and manufacturing. The coupling of three pumps in the same shaft may cause difficulties during starting. Arrangement should be made so that each pump starts in sequence, one after another.

TABLE - I

	Stations	Potential month for power generation by windmill	Average speed m/sec
1.	Chittagong	March to September	4
2.	Dhaka	March to October	3.2
3.	Khepupara	February to September	3.2
4.	Comilla	March to September	2.8
5.	Teknaf	June to September	2.3
6.	Jessore	April to September	2.1
7.	Cox's Bazar	May to August	2
8.	Hatiya	End of April to July	1.9
9.	Dinajpur	March to August	2
10.	Rangamati	April and May	1.9

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