Study on the Rotor Design Method for a Small Propeller-Type Wind Turbine Postprint

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Abstract

Numerical simulations were performed to predict the film cooling effectiveness on the flat plate with a three-dimensional discrete-hole film cooling arrangement. The effects of basic geometrical characteristics of the holes, i.e. diameter D, length L and pitch S/D were studied. Different turbulent heat transfer models based on constant and variable turbulent Prandtl number approaches were considered. The variability of the turbulent Prandtl number Pr-t in the energy equation was assumed using an algebraic relation proposed by Kays and Crawford, or employing the Abe, Kondoh and Nagano eddy heat diffusivity closure with two differential transport equations for the temperature variance k(theta) and its destruction rate epsilon(theta). The obtained numerical results were directly compared with the data that came from an experiment based on Transient Liquid Crystal methodology. All implemented models for turbulent heat transfer performed sufficiently well for the considered case. It was confirmed, however, that the two-equation closure can give a detailed look into film cooling problems without using any time-consuming and inherently unsteady models.

Full Text

Preamble

Study on the Rotor Design Method for a Small Propeller-Type Wind Turbine

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Small propeller-type wind turbines operate at low Reynolds numbers, which limits the selection of usable airfoil materials. Consequently, their design method-

ology remains insufficiently established, and their performance is often suboptimal. The ultimate goal of this research is to establish high-performance design guidelines and methods for small propeller-type wind turbines. To this end, we designed two rotors: Rotor A, based on the rotor optimum design method derived from blade element momentum theory, and Rotor B, featuring an extended tip chord length with a linearized chord length distribution. We examined the performance characteristics and flow fields of both rotors through wind tunnel experiments and numerical analysis. Our results revealed that while the maximum output tip speed ratio of Rotor B shifted to a lower value than that of Rotor A, its maximum power coefficient increased by approximately 38.7%. Both rotors exhibited large-scale separation on the hub side, which extended to the mid-span region in Rotor A. This difference in separation behavior significantly impacted the substantial decrease in Rotor A' s output relative to the design value and the corresponding increase in Rotor B' s output compared to Rotor A.

Keywords: Wind Turbine, Propeller-Type, Horizontal Axis, Blade Element Momentum Theory, Rotor Design

Introduction

In recent years, energy problems stemming from global warming and fossil fuel depletion have become increasingly serious, prompting expectations for effective utilization of natural energy sources. Wind power generation offers relatively low power generation costs, making it suitable for commercialization. Propeller-type wind turbines are currently the standard for power generation and have been progressively upsized for deployment not only in coastal and mountainous areas but also in offshore locations. Meanwhile, small propeller-type wind turbines are expected to become more common in urban and suburban environments; however, their widespread adoption remains limited due to insufficient power generation from poor startup characteristics and low power generation efficiency, damage from changes in wind direction and speed, and noise pollution from turbine rotation.

Blade element momentum theory [1,2] has been widely employed for rotor design of propeller-type wind turbines, enabling both optimum design [3-6] and performance analysis [7,8]. However, few reports provide detailed comparisons between optimal properties and actual flow fields of rotors designed using blade element momentum theory. Specifically, for small propeller-type wind turbines, the Reynolds number is low and usable airfoil materials are limited; thus, examining optimal properties and flow fields is particularly important. In recent years, reverse taper blades—where chord length increases toward the tip, contrary to conventional small propeller-type wind turbine designs based on blade element momentum theory—have been employed and experimentally shown to improve turbine performance within low rotational speed ranges [9,10]. However, the mechanism behind this performance improvement remains unclear. At present, a comprehensive design method for small propeller-type wind turbines has not been established.

In this research, our ultimate goal is to establish high-performance design guidelines and methods for small propeller-type wind turbines. Therefore, we designed rotors using the rotor optimum design method [11] based on blade element momentum theory, as well as rotors with extended tip chord lengths and linearized chord length distributions. The performance characteristics and flow fields of these two rotor types were examined through wind tunnel experiments and numerical analyses and compared with their design values.

Nomenclature

Symbol	Description
a	axial induction coefficient
a'	circumferential induction coefficient
A	wind-receiving area (m^2)
B	number of blades
C	chord length (m)
C_D	drag coefficient
C_L	lift coefficient
C_W	power coefficient
D	rotor diameter (m)
n	rotational speed (min ^{1})
P	static pressure (Pa)
Q	torque ($\mathrm{N \cdot m}$)
r	radius (m)
Re	local Reynolds number $= wC/v$
r_h	hub radius (m)
r_t	tip radius (m)
v	absolute velocity (m/s)
v_∞	absolute velocity at infinite distance (m/s)
w	relative velocity (m/s)
W	turbine output (W) $= Q\omega$
α	attack angle (°)
β	blade angle (°)
ε	drag-lift ratio
ζ	convection velocity ratio
λ	tip speed ratio $= r_t\omega/v_\infty$
ν	kinematic viscosity coefficient (m^2/s)
ρ	fluid density (kg/m^3)
ϕ	angle between rotation surface and relative velocity (°)
ω	rotational angular velocity (rad/s) $= 2\pi n/60$
Subscripts	
1	rotor inlet
2	rotor outlet

Symbol	Description
a	axial component
r	radial component
u	circumferential component
∞	infinite distance

Rotor Design Method

We designed rotors using the rotor optimum design method [11], which is based on blade element momentum theory. The summary is presented below [11].

Design specifications are as follows: v_∞ is the absolute velocity at an infinite distance, r_t is the tip radius of the rotors, r_h is the hub radius of the rotors, and B is the number of blades. We assumed the tip speed ratio (λ) and power coefficient (C_W) as follows:

$$C_W = \frac{W}{\frac{1}{2}\rho v_\infty^3 A}$$

where ρ is the fluid density and ω is the rotational angular velocity. Turbine output (W) and rotational speed (n) are confirmed using the equations as follows:

$$W = Q\omega$$
$$\omega = \frac{2\pi n}{60}$$

Because design calculations are performed for each radius, the blade is subdivided in the radial direction. The dimensionless radius (ξ) is defined as:

$$\xi = \frac{r}{r_t}$$

where r is the radius. In this study, design calculations were performed with $\xi = 0.14, 0.2, 0.3, 0.4, 0.5, 0.6, 0.7, 0.8, 0.9$, and 0.975. Because calculations cannot be performed when $\xi = 1$, $\xi = 0.975$ was used for the calculation instead.

The velocity triangle for an arbitrary radius is shown in Figure 1 [Figure 1: see original paper] [11]. The angle between the relative velocity (w) of the inflow into the blade element and the chord is defined as the attack angle (α), and the angle between the rotation surface of the rotor and the chord is defined as the blade angle (β). The angle between the rotation surface and the relative velocity (w) is defined as ϕ. In addition, vortices from blades of arbitrary radius form a spiral vortex surface. At this time, the slipstream was not expanded and was assumed to have the same diameter. We also assumed a condition in which the energy loss is minimized. Betz' s law [12] describes where the vortex surface flows without deformation. The locus of the vortex filament present in the vortex surface is shown in Figure 2 [Figure 2: see original paper] [11]. Here, w'_n is the convection velocity of the vortex filament and is perpendicular to the vortex surface.

In addition, v' is the apparent convection velocity in the axial direction of the vortex surface. We assume the convection velocity ratio (ζ) to be defined by the equation given below (ζ is constant regardless of the radius). In this research, we used $\zeta = 0.3$ as the initial value.

$$\zeta = \frac{v'}{w'_n}$$

where C_D is the drag coefficient. The power coefficient C_W can be written as:

$$C_W = 4\zeta(1-\zeta)^2$$

The value for ϕ is obtained from:

$$\tan\phi = \frac{1-\zeta}{\lambda\xi}$$

The loss coefficient of the blade tip (F) is obtained from the Prandtl equation [12] as follows:

$$F = \frac{2}{\pi}\cos^{-1}\left[\exp\left(-\frac{B(1-\xi)}{2\sin\phi_t}\right)\right]$$

where ϕ_t is the ϕ of the blade tip. We used ϕ when $\xi = 0.975$.

Next, we select the airfoil and obtain the lift coefficient (C_L) of the blade element from airfoil characteristics. The local Reynolds number (Re) is obtained from:

$$Re = \frac{wC}{\nu}$$

where C is the chord length and ν is the kinematic viscosity coefficient. We set the attack angle (α) of the blade element to obtain the drag-lift ratio (ε) using the local Reynolds number (Re) and airfoil characteristics by the equation as follows:

$$\varepsilon = \frac{C_D}{C_L}$$

Moreover, J_1 and J_2 are obtained from:

$$J_1 = \int_{\xi_h}^{1} \frac{(1-a)^2}{\sin^2\phi} F\xi d\xi$$

$$J_2 = \int_{\xi_h}^{1} \frac{(1-a)a'}{\sin\phi\cos\phi} F\xi d\xi$$

The obtained J_1 and J_2 as well as the assumed power coefficient (C_W) are used to calculate ζ from:

$$\zeta = \frac{J_1}{J_1 + J_2}$$

If the calculated ζ and the ζ assumed as the initial value do not match, we use the ζ obtained here and recalculate with ϕ in Equation (7). If ζ does not converge even after repeating this calculation, we change the assumed power coefficient (C_W), airfoil, or attack angle (α). If ζ still does not converge, the design specifications must be reviewed.

If ζ converges, we obtain axial induction coefficient (a) and circumferential induction coefficient (a') from:

$$a = \frac{J_1}{J_1 + J_2}(1-\zeta)$$

$$a' = \frac{J_2}{J_1 + J_2}\frac{\zeta}{\lambda\xi}$$

The relative velocity (w) is obtained using the equation as follows:

$$w = \frac{v_\infty(1-a)}{\sin\phi}$$

Chord length (C) is determined from:

$$C = \frac{8\pi r_t \zeta(1-\zeta)}{BC_L\lambda}$$

Blade angle (β) is obtained from:

$$\beta = \phi - \alpha$$

The obtained chord length distribution ($C(r)$) and blade angle distribution ($\beta(r)$) determine the blade shape, and an optimum rotor can be established.

Experimental Apparatus and Methods

Test Rotors

Schematics of the two types of test rotors used in this experiment are shown in Figure 3 [Figure 3: see original paper], and their specifications are listed in Table 1 . In addition, the chord length and blade angle distributions are illustrated in Figures 4 and 5. Rotor A was designed using the optimal design method [11] based on the blade element momentum theory described above. Its design specifications are shown in Table 2 .

We set the absolute velocity at infinite distance (v_∞) as 8 m/s, the rotor tip radius (r_t) as 0.25 m, the rotor hub radius (r_h) as 0.07 m, and the number of blades (B) as 3, and assumed a tip speed ratio (λ) of 5 and a power coefficient (C_W) of 0.4 for the design. For all radial positions, MEL031 [13] was used as the airfoil, and the attack angle (α) was 4°. The chord length at the tip was 33.3 mm.

However, giving consideration to strength, in Rotor B, the chord length was extended such that the maximum blade thickness at the tip matched that of Rotor A at 5.0 mm, and the chord length distribution was linearized. In addition, the blade angle of both rotors was the same.

Experimental Apparatus and Methods

An outline of the wind tunnel experimental apparatus is shown in Figure 6 [Figure 6: see original paper]. This open-type wind tunnel has air outlet dimensions of 800 mm × 800 mm. Downstream from the center of the wind tunnel outlet, the 1D position (500 mm) was designated as the standard position, and the wind velocity at this position was set as 8 m/s. Wind velocity was measured using a Pitot tube and an inclined tube manometer. Flow at 1D, a downstream position from the wind tunnel outlet, was confirmed to be almost uniform. In addition, test rotors were positioned at 1.25D (750 mm) downstream from the wind tunnel outlet.

The load of the rotor was controlled using a motor and an inverter. Rotational speed (n) and torque (Q) were measured with a magnetoelectric rotation detector and a torque detector, respectively. From these values, the turbine output (W) was obtained. Note that the torque obtained was corrected by measuring the torque without a rotor.

Numerical Analysis Methods and Conditions

We used the general-purpose thermal fluid analysis code ANSYS CFX 15.0 for the numerical analyses, and conducted three-dimensional incompressible steady flow analyses. The governing equations are the conservation of mass equation [14] and the conservation of momentum equation [14]. The SST (Shear Stress Transport) model [15] was adopted as the turbulence model. Air at 25°C was used as the working fluid.

The total computational region is shown in Figure 7 [Figure 7: see original paper]. The computational region mainly consisted of an external region, a middle region, and a rotor region. The external region was cylindrical with a diameter 10 times the outer diameter D of the rotor. Moreover, the lengths upstream and downstream were 10 times and 15 times the length of the outer diameter D of the rotor, respectively, and were measured from the center of the rotor. The numbers of computational grids for Rotors A and B were approximately 4,130,000 and 4,030,000 elements, respectively; the total numbers of grids were 6,020,000 and 5,920,000 elements, respectively. The boundary conditions were the flow velocity ($v_\infty = 8$ m/s) for the inlet boundary, arbitrary rotational speed for the rotor, and the static pressure ($P_\infty = 0$ Pa) for the outlet boundary. Regarding the wall surface boundary, the outer periphery of the external region was assigned as the slip condition and wall surfaces were assigned as the no-slip condition. Moreover, the boundaries between the rotating and static systems were joined using the frozen rotor [16] technique.

Results and Discussion

Comparison of Performance Characteristics

The relation between the tip speed ratio (λ) and the power coefficient (C_W) is shown in Figure 8 [Figure 8: see original paper]. First, experimental and calculated values for C_W of Rotor B were compared. Both C_W values reached their maximum at $\lambda = 4.5$. At this maximum output tip speed ratio ($\lambda = 4.5$), the experimental value was $C_W = 0.335$, whereas the calculated value was $C_W = 0.333$; these values were in good agreement. However, at low tip speed ratios ($\lambda = 2.5$) or high tip speed ratios ($\lambda = 6$), differences between these two values increased. At the maximum output tip speed ratio of Rotor B, the experimental and calculated values were in good agreement; thus, our calculated results are considered to be valid.

Next, let us consider the calculated C_W of Rotor A. The maximum value of this C_W (0.240) was obtained at the designed tip speed ratio ($\lambda = 5$); however, this value was significantly lower than the design value of $C_W = 0.4$. On the other hand, the calculated C_W of Rotor B had a maximum value that, while lower than the design value, was approximately 38.7% higher than that of Rotor A. Yet, the maximum output tip speed ratio of Rotor B was shifted slightly lower, as described previously. We will examine the significant decrease in Rotor A' s output compared to the design value and the increase in Rotor B' s output compared to Rotor A.

Comparison of Flow Fields

The radial direction distributions of the axial components (v_{a1} and v_{a2}) and the circumferential components (v_{u1} and v_{u2}) are shown in Figures 9 and 10, respectively, for the absolute velocity at the inlet/outlet of Rotors A and B based on numerical analysis. In both cases, the tip speed ratio was $\lambda = 5$. Figure 9 [Figure 9: see original paper] shows that when comparing v_{a1} of Rotor A with the design value, at any radial position, v_{a1} of Rotor A was significantly larger than the design value. On the hub side, v_{a1} of Rotor B was similar to that of Rotor A, but from the mid-span to the tip side, the value of Rotor B was smaller than that of Rotor A, and at around $0.8 \leq r/r_t \leq 0.9$, the values were almost the same as the design value. However, on the tip side, when $r/r_t > 0.9$, v_{a1} of Rotor B increased rapidly; the same was true for Rotor A. On the other hand, v_{a2} decreased compared to v_{a1}. Especially in Rotor B, the difference between v_{a1} and v_{a2} was large. Therefore, compared to Rotor B, the flow in Rotor A was less complete, meaning that the energy of wind was not sufficiently extracted.

Figure 10 [Figure 10: see original paper] shows that v_{u1} in Rotors A and B was almost 0, as designed, and at the rotor inlet, the flow was without rotation. However, v_{u2} had positive and negative values for both rotors, which were especially large on the hub side. When comparing v_{u2} of Rotors A and B, the values were almost the same on the hub side, but from the mid-span to the tip side, the value of Rotor B was higher than that of Rotor A. Differences in the v_a and v_u distributions of Rotors A and B likely arise from the fact that the chord length was mostly the same on the hub side but differed from the mid-span to the tip side.

The relative velocity (w_1) at the rotor inlets of Rotors A and B based on numerical analysis, the local Reynolds number (Re), and the radial direction distribution of the attack angle (α) are shown in Figures 11, 12, and 13, respectively. In all cases, the tip speed ratio was $\lambda = 5$. According to Figure 11 [Figure 11: see original paper], w_1 of Rotors A and B was almost the same and agreed well with the design value.

Therefore, as shown in Figure 12 [Figure 12: see original paper], Re of Rotor A agreed with the design value, increasing from the hub to the mid-span side and then decreasing as it approached the tip side. On the other hand, Re of Rotor B exhibited a similar value to that of Rotor A on the hub side but increased as it approached the tip side, reaching approximately 0.9×10^5. This is because the chord length of Rotor B from the mid-span to the tip side was larger than that of Rotor A. Figure 13 [Figure 13: see original paper] shows that α of Rotor A was larger than the design value at all radial positions and was especially large on the hub side. Meanwhile, α of Rotor B had a similar value to that of Rotor A on the hub side but was smaller than that of Rotor A from the mid-span to the tip side, and at $0.8 \leq r/r_t \leq 0.9$ on the tip side, the α value became similar to the design value.

Figures 14(a) and 14(b) show limiting streamlines on suction surfaces for Rotors A and B when $\lambda = 5$. A large-scale separation was observed on the hub side of both rotors. Especially in Rotor A, the separation extended to the mid-span side. For the MEL031 airfoil used in this research, the published lower limit of the Reynolds number characteristic is 1.0×10^5. Therefore, values for the lift coefficient and drag-lift ratio in Reynolds numbers below 1.0×10^5 were obtained through linear function extrapolation. However, because the local Reynolds number (Re) was extremely low (approximately 0.4×10^5) on the hub side of both rotors, the design values and actual values for the lift coefficient and drag-lift ratio deviated significantly. Thus, the axial component of the absolute velocity became large on the hub side, making the attack angle (α) much larger than the design value. Consequently, a large-scale separation was observed on the hub side of both rotors. Especially for Rotor A, Re was low from the mid-span to the tip side with a large attack angle (α); thus, the separation extended to the mid-span. The difference in the area of separation likely had an impact on the significant decrease in Rotor A' s output compared to the design value and the increase in Rotor B' s output compared to Rotor A.

Conclusions

We examined performance characteristics and flow fields of Rotor A (designed using the rotor optimum design method based on blade element momentum theory) and Rotor B (with increased tip chord length and linearized chord length distribution compared to Rotor A) through wind tunnel experiments and numerical analysis. The findings of our study are as follows:

1. In Rotor A, the maximum output tip speed ratio agreed with the design value, but the maximum power coefficient decreased significantly in comparison with the design value.

2. In Rotor B, the maximum output tip speed ratio was lower when compared with Rotor A, but the maximum power coefficient increased by approximately 38.7%.

3. A large-scale separation was observed in Rotors A and B on the hub side; however, the separation extended to the mid-span side in Rotor A. This difference in the extent of separation is believed to cause the significant decrease in Rotor A's output compared to the design value and the increase in Rotor B' s output compared to Rotor A.

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Note: Figure translations are in progress. See original paper for figures.