Energy Conversion and Management 172 (2018) 277–284

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

The dead band in the performance of cross-flow turbines: Effects of Reynolds number and blade pitch

M. Somoano, F.J. Huera-Huarte*

Department of Mechanical Engineering, Universitat Rovira i Virgili (URV), 43007 Tarragona, Spain

ARTICLE INFO

Keywords:
Cross-flow turbines
Darrieus rotors
H-rotors
Straight-bladed turbines
Vertical axis wind turbines
Blade pitch

ABSTRACT

The performance of a three straight bladed Cross-Flow or Vertical Axis Turbine based on symmetrical profiles, with a chord-to-diameter ratio of 0.16 has been studied in detail. The research is focused on the effects of Reynolds number and blade pitch angle, not only in the performance itself, but in the appearance or not of a dead band region in which aerodynamic loading cannot overcome resistive loads. Experiments were carefully designed in order to cover the transitional region were this dead band is incipient, for Reynolds numbers in the range from 3 to $5{\cdot}10^5$.

Results show how the performance is highly dependent on pitch angle at all Reynolds numbers investigated. Efficiencies over 25% are reached at tip speed ratios (blade to free stream velocity ratio) slightly larger than 2, for pitch angles in the range 4–8°. The appearance of the dead band is conditioned by the combination of Reynolds number and pitch angle.

1. Introduction

Cross-Flow Turbines (CFT), also known as vertical axis wind turbines or Darrieus rotors [1], are characterized by having their shaft perpendicular to the incoming flow and are designed for Tip Speed Ratios (TSRs) greater than one [2]. The TSR is a dimensionless number for the relationship between the tangential speed of the blade due to its rotation and the free stream velocity

$$\lambda = \frac{\omega D}{2u_\infty} \tag{1}$$

where ω is the rotational speed, D the turbine diameter, and u_∞ the free stream velocity. The interest on straight bladed CFTs has been growing over the last decades because they have several advantages over Axial-Flow Turbines (AFTs), namely: omni-directionality, lower maintenance costs, better behaviour in unsteady and skewed wind conditions with complex flow patterns such as high wind velocities and turbulent wind flows [3–5].

Performance curves result from plotting the power coefficient as a function of TSR, and are widely used to represent the efficiency of turbines. The power coefficient is the ratio of the mechanical power to the power available in the free stream, a dimensionless form of the power output which indicates the efficiency of the turbine, and is defined as

$$C_P = \frac{T\omega}{\frac{1}{2}\rho u_\infty^3 LD} \tag{2}$$

where T is the net torque applied on the shaft, ρ is the fluid density, and L the blade span. The shape of this performance curve in the case of cross-flow turbines shows three clearly distinct and well defined regions as a function of the tip speed ratio [6]. In the first region (R1), for the lowest TSRs, the power coefficient grows monotonically. For intermediate TSRs, in the second region (R2), the C_P exhibits its maximum values. In the third region (R3), for highest TSRs, the C_P drops abruptly. For sufficiently low or high λ, the performance of this type of turbines is considerably lower than that typical of region R2. Besides, it is also well known that the higher the turbine solidity $\left(\sigma = \frac{Nc}{\pi D}\right)$, which depends not only in the diameter but also in the number of blades (N) and the chord (c), the lower the optimal tip speed ratio that maximizes its performance [7–9]. Important parameters with great influence on performance of turbines are the pitch angle between the foil centreline and the line tangent to the circular trajectory of the blade at its mid-chord (β) and the Reynolds number based on the turbine diameter ($Re_D = u_\infty D/\nu$, with ν being the kinematic viscosity of the fluid).

In order to obtain the performance curve of a three straight bladed CFT, Bachant and Wosnik [10] towed their model in a water tank. The turbine had blades with NACA-0020 profile and a chord-to-diameter ratio (c/D) of 0.14. Their experiments were all conducted at the same speed, with a constant Re_D of $1.0{\cdot}10^6$. Tip speed ratio was prescribed and

* Corresponding author.
E-mail address: francisco.huera@urv.cat (F.J. Huera-Huarte).

https://doi.org/10.1016/j.enconman.2018.06.087
Received 11 April 2018; Received in revised form 12 June 2018; Accepted 23 June 2018
0196-8904/ © 2018 Elsevier Ltd. All rights reserved.

held constant during each run, by using a servo motor coupled to the turbine shaft. They showed a maximum power coefficient of 0.26 at a tip speed ratio of 1.9. By changing Re_D in the range from $3.0{\cdot}10^5$ to $1.3{\cdot}10^6$ [11], they observed a drastic improvement in the dimensionless power output of the CFT.

Bravo et al. [12] used a three straight bladed CFT model with NACA-0015 blades and a c/D of 0.16 to carry out experiments in a full-scale wind tunnel for a range of diameter Reynolds number from $1.0{\cdot}10^6$ to $2.7{\cdot}10^6$. They measured the torque with a load cell and devised an active closed-loop speed control system, that accurately regulated the rotational velocity of the rotor. They controlled the speed because the system was not self-regulating at the lowest TSR of the performance curve, before reaching the maximum power output. With a constant load the CFT was not able to keep its angular rotating speed ω, so it tended to slow down until stopping or headed to higher speeds. At all Re_D investigated, the maximum power coefficient occurred at a tip speed ratio $\lambda \approx 1.6$ and for $Re_D \geqslant 1.3{\cdot}10^6$ the performance curve became essentially Re independent. Remarkably, they did not obtain their maximum C_P, with a value of 0.3, at the highest Reynolds number, but at $Re_D = 1.7{\cdot}10^6$. Fiedler and Tullis [13] conducted identical experiments but at lower Reynolds numbers, $8.3{\cdot}10^5 < Re_D < 1.8{\cdot}10^6$, showing consistent results. They repeated the measurements with different preset pitch angles at $Re_D = 1.7{\cdot}10^6$, taking into account that a forward shift of mount-point from the centre of the chord towards the leading edge corresponds to an effective toe-in pitch change of

$$\beta = tan^{-1}\left(\frac{2x}{D}\right) \tag{3}$$

where x is the distance between these two points along the blade chord. With a $\lambda = 1.6$, the authors obtained a better performance as they changed β from 6.4° toe-in to 5.3° toe-out, passing through 2.5° toe-in and 1.4° toe-out.

In the work by Takao et al. [14], the authors used a three straight bladed CFT, but this time based on asymmetric NACA-4518 profiles, with a c/D ratio of 0.17. The experiments were performed in an open-jet wind facility at $Re_D = 3.2{\cdot}10^5$, and a maximum power coefficient $C_P = 0.11$ was obtained at a tip speed ratio $\lambda = 1.7$. Li et al. [15] investigated a four straight bladed CFT with a chord-to-diameter ratio $c/D = 0.13$ and NACA-0021 blades. With a pitch angle of 8° toe-out, the turbine had the highest power output. In addition to a torque meter, the authors used a six-component balance to measure the loads on the turbine at a $Re_D = 1.1{\cdot}10^6$. The power coefficient slowly rose up to $C_P \approx 0.15$ and then from $\lambda = 1.7$, it decreased rapidly. Li et al. [16] repeated the previous experiment on a two straight bladed CFT model with a c/D ratio of 0.13, changing the pitch angle in 2° steps from $\beta = 4°$ toe-out to $\beta = 8°$ toe-out. The turbine performance increased up to $C_P = 0.21$ at $\lambda = 2.2$ when adjusting the blade pitch from 4° to 6° toe-out, but decreased when moving β from this last value to 8° toe-out. Moreover the optimal tip speed ratio that maximized its performance was slightly increased as the pitch angle became higher.

El-Samanoudy et al. [17] with their four straight bladed CFT based on NACA-0024 airfoil with a chord-to-diameter ratio of 0.19 in a wind open-jet facility at a $Re_D = 4.3{\cdot}10^5$, also observed that the maximum dimensionless power output began to decrease as soon as the pitch was configured with toe-out positions. Using load cells to measure the tension of a rope in contact with a grooved pulley attached to the shaft, they found a maximum power coefficient of 0.25 at $\lambda = 1.3$ for a β of 10° toe-out. The maximum C_P for both $\beta = 0°$ and $\beta = 20°$ toe-out, was smaller and took place at lower TSR. Strom et al. [18] tested straight bladed CFTs with two and four NACA-0018 blades with a c/D ratio of 0.24 in a water flume facility at $Re_D = 1.2{\cdot}10^5$, covering the range of pitch angles $0° < \beta < 12°$ toe-out with increments of 2°. They operated a servo motor by setting up constant velocities and measured the turbine torques with a six-axis load cell. In both cases the best scenario in terms of performance took place with $\beta = 6°$, obtaining the maximum C_P with a value of 0.18. The optimal tip speed ratio ($\lambda = 1.8$) for the two-bladed CFT was higher, but in both cases the performance curves had two clearly defined peaks. The region between the two peaks, where the torque can be negative was named *dead band* by Baker [19].

Araya and Dabiri [20] investigated a three straight bladed CFT with its blades based on NACA-0018 profile and a chord-to-diameter ratio $c/D = 0.33$, in a water channel facility. Their goal was to prove that motor-driven and flow-driven turbines have similar hydrodynamics. They defined the rotor *neutral curve* as the $\lambda(Re_D)$ curve that shows the upper bound of tip speed ratios for flow-driven cases, i.e. the maximum λ values that the rotor reaches if flow-driven. Their empirical curve was shifted to lower TSRs than that would be theoretically possible in the presence of only airfoil lift and drag forces because of the inherent losses of the turbine system, and was monotonically increasing except from around $Re_D \approx 6{\cdot}10^4$ where there was an abrupt rise. Moreover they obtained good agreement between the torque measurements of motor-driven and flow-driven cases using a rotary torque sensor at a diameter Reynolds number $Re_D = 7.8{\cdot}10^4$, when operating on the back side of the performance curve (for λ larger than those at the maximum power output) where CFTs operate with constant angular velocities. Edwards et al. [21] tested a three straight bladed CFT (NACA-0022 with a ratio $c/D = 0.06$) in a wind tunnel with an open section at a $Re_D = 3.1{\cdot}10^5$. They used a motor that could be engaged/disengaged via an electromagnetic clutch, to drive the turbine up to high speeds passing through the dead band.

The objective of the work presented here, is to describe the effect of Reynolds number and blade pitch on the efficiency of cross-flow turbines. Although this has been investigated in the past [13,16–18], there are no published works that cover the transitional region in which the aerodynamic torque of the blades overtakes the resistive torques, with cases clearly showing the appearance of the dead band. We demonstrate how small variations of the pitch angle can change the shape of the performance curves, and therefore the behaviour of this type of turbines. Moreover we put in perspective these results, with recent measurements that show the flow dynamics inside the rotor of this type of machines.

2. Experimental set-up

The experiments were carried out in the Boundary Layer Wind Tunnel (BLWT) of the Laboratory for Fluid-Structure Interaction (LIFE), at Universitat Rovira i Virgili (URV) in Tarragona. The facility is an open-circuit blower tunnel, with an axial fan that forces the air into a settling chamber. The settling chamber is composed of an array of three honeycombs that reduce turbulence intensity, and supply a more uniform flow to a three-dimensional contraction before the working section. Wind speeds up to 12 m/s can be reached. The wind tunnel has a total length of 23 m and a working section of 1.84×1.22 m^2 with 14 m of optical access. The model was placed centred in the working section in order to avoid wall effects, and to have a uniform flow profile. The wind tunnel calibration in the test-section previous to the experiments presented here, resulted in a turbulence intensity of 2.4% in the worst case scenario, i.e. at the highest Reynolds number in the experiments.

The turbine model consisted of a three straight bladed cross-flow turbine with its blades based on the NACA-0015 profile, printed in PolyLactic Acid (PLA). The blades had a chord of 0.12 m and a total span of 0.75 m with an inner core made of an aluminium 8 mm square tube, that provided very high flexural stiffness to the blades. They were connected to a horizontal stainless steel shaft with a diameter of 25 mm, through six aluminium struts located at 1/5 and 4/5 of the blade span. The turbine diameter (D) was 0.75 m leading to a chord-to-diameter ratio (c/D) of 0.16 and a span-to-diameter ratio (L/D) of 1. The parts connecting the aluminium square tube that acted as the core of the blades and the two struts, were also printed in PLA and allowed the blades to be oriented with the desired fixed pitch angle. Experiments were conducted by setting up 9 different pitch angles β, taking into account the effective toe-in pitch in Eq. (3) for a forward shift of the

Download English Version:

https://daneshyari.com/en/article/7157971

Download Persian Version:

https://daneshyari.com/article/7157971

Daneshyari.com