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THE AERODYNAMIC DESIGN OF A 5kW WIND TURBINE

P.D. CLAUSEN, P. EBERT, S.G. KOH and D.H. WOOD

Department of Mechanical Engineering
University of Newcastle, NSW 2308, AUSTRALIA

ABSTRACT

This paper describes the design of the blades and tail fin of a prototype 5 kW wind turbine. The blades were designed to give optimal performance at a high tip speed ratio. A finite element analysis of a rotating blade was done to allow minimisation of stress levels from centrifugal and aerodynamic forces. The tail fin was designed to counteract the torque extracted by the blades and to turn the turbine out of the wind after rated wind speed.

NOTATION

$C_p =$	$W/\frac{1}{2}\rho U_0^3 \pi r_t^2$, power coefficient
$C_t =$	$F/\frac{1}{2}\rho U_0^2 \pi r_t^2$, thrust coefficient
F	axial thrust on blades
r_t	tip radius
U_0	wind velocity
W	power extracted from the wind
$\lambda =$	$\Omega r_t/U_0$ tip speed ratio
ρ	density
Ω	rotational speed of the blades

INTRODUCTION

Man has extracted power from the wind for centuries. In the last few decades machines capable of producing a MW or so of power have been developed. These machines generally operate in "wind farms" in windy locations and are connected to the main electricity grid. Less attention has been given to machines producing only a few kilowatts of power suitable for remote area power systems. This paper describes aspects of the design of a 5 kW wind turbine for domestic power supply in remote areas which are often associated with low wind speeds.

This turbine has been designed to have low speed starting torque and will operate at maximum performance for a range of wind speeds by varying Ω to maintain λ at its optimal value. To maximise its efficiency at low wind speed, the generator will be connected to the blades through a 10.5:1 step-up gearbox. The turbine will have another unusual feature. The generator will be fixed at the top of the support tower on the axis of the wind turbine pivot and connected to the gearbox by a right-angle mitre-box. This arrangement obviates the need for slip rings to remove power from the wind turbine, however a counter-torque equal in magnitude to the torque supplied to the generator, must be supplied by the tail fin. A schematic of this wind turbine is shown in figure 1. The two 2.5 m long blades will be cast from moulds that were manufactured using the Departments' numerically controlled milling machine.

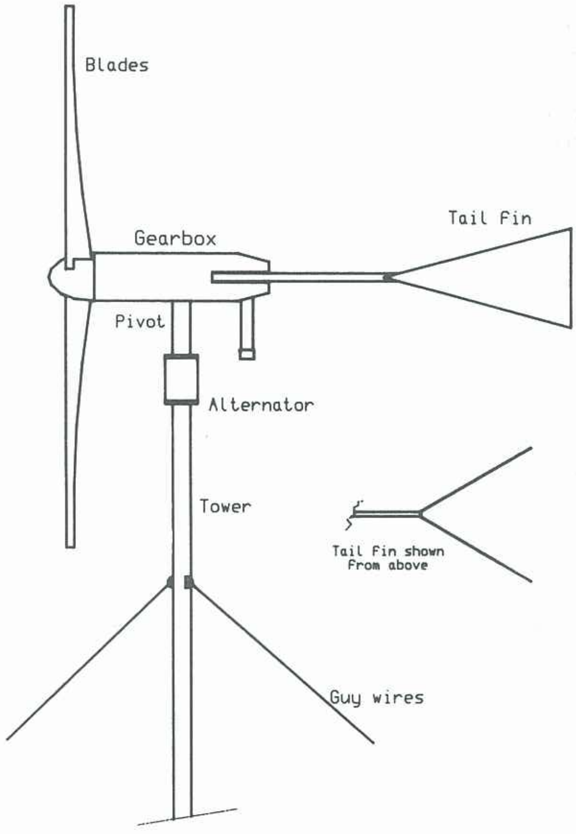

Figure 1. Schematic of the wind turbine

The paper concentrates on the design of two parts of the prototype wind turbine; the blades and the tail fin. Firstly the aerodynamic design of the blades are discussed. Next a detailed finite element model of the blade will be described and the results for a fibreglass blade discussed. Finally the design of the tail fin is discussed. All the design decisions were based on analysing the turbine's behaviour in steady flow. After installation of the prototype at the end of this year, field testing of around 18 months will be used to assess the main effects of unsteadiness.

BLADE AERODYNAMIC DESIGN

It is hoped to achieve good low wind speed performance largely by designing the blades to have optimum performance at a high tip speed ratio, λ. This together with the constant λ control strategy and the step-up gearbox, should allow the generator to run at the highest possible shaft speed for all wind conditions. The design was based on the turbine tested in the field and in a wind tunnel by Anderson et al. (1982), which achieved a maximum Cp of 0.44 at λ = 10. For our machine, this corresponds to a shaft speed of 400 rpm at a wind speed, U_0, of 10 m/sec.

Two main computer programs were used in the design. The first is based on traditional blade element theory (BET) which subdivides the blades into a number of radial blade elements. In turn, these elements are assumed to behave as aerofoils, whose lift and drag coefficients can, therefore, be used in determining the blade's performance. Coupled to BET is a model for the expanding and rotating wake that was developed by Koh & Wood (1991a, 1991b). The geometry of the wake is determined partly by balancing the thrust and torque acting on each blade element against the angular and axial momentum flux in the flow over that element. The axial velocity used in these equations depends on the pitch of vortices comprising the wake. Since the velocity appears in the energy equation, this dependence emphasises the relationship between the vortex geometry and turbine performance. As shown by Koh & Wood (1991a), the Betz limit can be derived from a special case of the wake model. The velocities induced at the blades were calculated from the Biot-Savart law using the technique described (as Method 1) by Wood & Meyer (1991). Approximately 90 - 95% of the cpu time required by the program was expended in these calculations. Figure 2 shows a comparison between the thrust and power coefficients predicted by the computer program and the measurements of Anderson et al. (1982). The computer program was then used to modify the aerofoil section, and the chord and twist distribution of the original blade to slightly improve the overall performance (The twist is the angle between the chord and the plane of rotation).

The second program combines the wake model with a boundary integral or "panel" method for the blades. The latter was described by Wood (1991) and does not, at present, contain any allowance for the effects of the blade boundary layer. Therefore, the panel code gives an upper limit on blade pressure distribution which was then used with the finite element analysis described in the next Section.

BLADE STRESS ANALYSIS

The blade design calculations gave the chord and twist for each blade element. The relative positioning of these elements along a chord line is, however, arbitrary as long as the radial continuity of the blade is maintained. What is needed is a criteria for positioning the blade elements; the logical one is to minimise the stress in the blade induced by the highest non-fluctuating load - in this case caused by the centrifugal force.

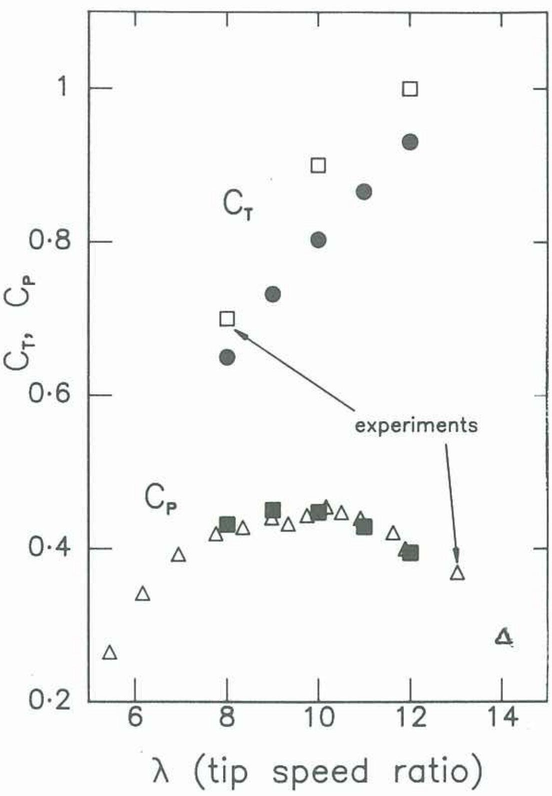

Figure 2. Comparison of predicted power and thrust coefficients with the measurements from Anderson et al. (1982)

The process of locating and adjusting the blade elements to minimise stress was done using STRAND6, a finite element analysis software package. The finite element approach was taken as it combined simplicity in creating a model of a complex three dimensional surface and accuracy in calculating the blade deflection and stresses.

In addition to the stacking problem, there were two other problems to be addressed by the finite element analysis; where to place the fibres in the fibreglass blade to optimise material use, and to check that the natural frequency of vibration of the blade is well outside the normal operating speed range of the blade.

The stacking of the blade elements was investigated by assuming the blade was constructed from a solid isotropic material. This allowed the blade elements, constructed using 20 node isoparametric brick elements, to be moved with relative ease without loosing continuity in the surface. As a reasonable starting point, the quarter chord positions of each blade element were aligned along the blade. The next reasonable guess was to align the centre of area of each profile. This resulted in a 25% reduction in stress when compared to the quarter chord aligned model. Slight shifting of the elements about this position resulted in an increase in blade stress levels. It was found later that these results could be used when the blades were constructed from an anisotropic material such as timber or a fibreglass shell blade filled with a foam core. As a consequence, the blade elements were stacked about their centre of area.

The next stage was to create a finite element model of the fibreglass blade. The fibreglass skin of the blade was modelled using 4 node plate elements and the foam core modelled using 8 node tetrahedral brick elements. The plate elements were sized and orientated to be similar to, or a subset of, the panels in the panel code. This was done to allow the calculated surface pressure to be applied directly to the blade. The model was initially solved with isotropic plate properties to find the distribution of stress in the shell of the blade. The orientation and type of glass fibre was then determined from these results; the plate properties were then altered to the orthotropic properties of fibreglass. This model allowed the tip deflections for the blade to be determined. A plot of von Mises stress for the final blade is shown in figure 3 below. The maximum stress within the blade were found to be well below the breaking values for fibreglass.

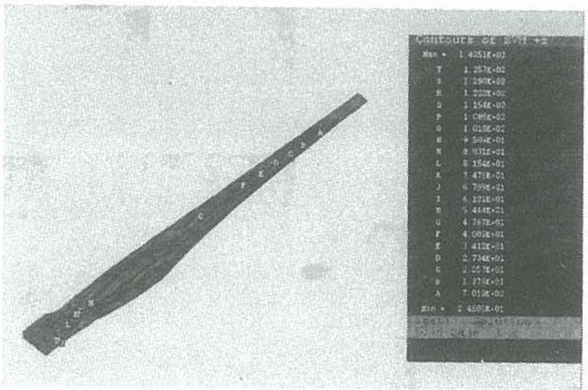

Figure 3. Contours of Von Mises stress for centrifugal and aerodynamic forces at λ = 10 and U_0 = 10 m/sec Stresses in MPa.

The maximum predicted streamwise deflection and lateral deflection in the plane of rotation was 4 mm and 2 mm respectively for the blade subject to centrifugal forces only. The radial deflection of the blade tip was found to be negligible. Detailed examination of the model showed the blade was twisting along its length. Adding the surface pressures to the finite element model increased the streamwise deflection to about 160 mm and the lateral deflection to about 35 mm for λ = 10 at 10 m/sec.

The natural modes of vibration of the blade are primarily a function of the stiffness and mass of the blade. It was found for this particular blade that the first mode of vibration occurred at a equilivant rotational speed of 500 rpm, 100 rpm above the design speed at 10 m/sec. It is likely that this difference was sufficiently large to prevent resonance.

TAIL FIN DESIGN

Wind turbine tail fins are traditionally used to point the turbine into the wind using a single fin. We combined this duty with the requirement that the tail fin turn the blades out of the wind after the design speed and so reduce the blade stresses induced by high wind speeds and prevent the power output exceeding 5 kW. The first design considered consisted of two delta wings joined at their apexes. One wing was fixed, with the other free to move on a hinge against a pretensioned spring. The pretensioning prevented the hinged wing from moving until the rated wind speed was reached. For higher wind speeds the hinged wing would collapse, and the tail fin would find a new equilibrium position by yawing the turbine out of the wind.

A computer program was written to model this process, and wind tunnel testing was carried out on a number of models to verify the predicted results and to investigate dynamic response. Model combinations of different planform wings and springs were tested at different wind speeds, with wing aspect ratios from 3.07 to 0.7. Flow visualisation was used to test for stall problems associated with vortex breakdown.

The results showed that the yawing capacity was restricted to the stall angle of the wing; the best configuration had a wing with an aspect ratio of 1.1 which gave about 35° of yaw. As a consequence, this tail fin would limit the power to 5 Kw only if the wind speed remained below 12.5 m/sec.

A second design with both wings fixed is currently under development. The counter-torque required by the generator will be developed from an offset of the delta wings, as indicated in figure 1. Producing this counter-torque, at least in steady flow, is straightforward for the reason that constant λ operation should require a counter-torque proportional to U_0^2 - a proportionality which, apart from Reynolds number effects, can be produced easily from virtually any lifting body. A collapse mechanism on the tail fin support arm will be used to turn the wind turbine out of the wind for all wind speeds greater than rated. When the wind speed drops, the mechanism resets itself and no manual resetting is required. A computer program modelling this tail fin has been written and the results suggest that it can work although wind tunnel and field testings are required before a definite conclusion can be reached.

SUMMARY AND CONCLUSIONS

This paper has described the design of the blades and tail fin of a 5 kW prototype wind turbine. The aerodynamic design of the blades should ensure good low wind speed performance by having optimum performance at high tip speed ratio, and a 10.5:1 step-up gearbox to allow the generator to run at high shaft speeds. The detailed finite element analysis of the blade structure allowed minimisation of the stresses from centrifugal and aerodynamic forces, mainly by positioning the aerofoil sections making up the blades along their centre of areas. The orientation and type of glass fibres were determined from this analysis. Also the natural frequency of vibration of the blade was computed to be greater than the operating speed of the turbine. Two tail fin designs were described. The second design will supply the counter-torque required by the positioning of the generator on the turbine pivot axis and progressively yaw the blades out of the wind at high wind speeds.

At present it is not possible to perform a detailed analysis of the unsteady behaviour of the wind turbine. Much of the prototype testing period will, therefore, concentrate on measuring the effects of unsteadiness on the turbine's aerodynamics, dynamics and stress levels. For example it may turn out that the unsteady counter-torque supplied by the tail fin does not always match the torque from the blades. Concurrently we are developing computational models to be used in conjunction with those measurements to improve the prediction of unsteady behaviour.

ACKNOWLEDGMENTS

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