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FEASIBILITY STUDY OF PULTRUDED BLADES FOR WIND TURBINE ROTORS

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ABSTRACT

In work performed under subcontract* to the National Renewable Energy Laboratory (NREL), a preliminary design study and proof-of-concept field test were conducted to evaluate the feasibility of using pultruded blades for wind turbine rotors. A 400 kW turbine was selected for the design study, and a scaled 80 kW rotor was fabricated and tested as a demonstration of the concept. To examine the feasibility of pultruded blades, several issues were addressed, including power performance, tower strikes, yaw stability, stall flutter, fatigue, and rotor cost. Results showed that with proper design, rotors using pultruded blades demonstrate acceptable fatigue life and stable yaw behavior without tower strikes. Furthermore, blades using this technology may be manufactured for approximately half the cost of conventional blades. Field tests of the scaled rotor provided experimental data on power performance and loads while verifying stable yaw operation.

INTRODUCTION

The primary incentive for pursuing pultruded blade technology is to reduce wind turbine cost of energy (COE). The initial blades, manufactured in 1980 by Morrison Molded FiberGlass (now Strongwell, Inc.) for the United Technologies Research Center, demonstrated the low cost of this method. Production requires little labor, and the cost of the materials is primarily for fiberglass and resin. This method demonstrated that blades could be made for less than \$4/lb when nearly all wind turbine blades manufactured by hand-lay-up methods cost approximately \$10/lb.

As a result of their constant cross section, pultruded blades are more flexible than conventional blades. In the past, this resulted in aeroelastically active and extremely flexible blades, which in turn led to operational problems. The history of turbines employed with this technology during the 1980s was not good. Problems included tower strikes, upwind running, and stall flutter. Because of the potential for achieving significant reductions in COE, it is logical to pursue solutions to the operational problems.

PS Enterprises (PSE) was awarded a subcontract under NREL's Next Generation Innovative Subsystems Program to conduct a feasibility study of using pultruded blades for utility-scale wind turbine rotors. The work consisted of the preliminary design of a 400 kW rotor and a field test of a dynamically-scaled version of that design.[1] In addition, several experts in the industry were employed to study specific technical issues related to pultruded blades and to compare cost to that of conventional designs.[2]

PULTRUDED BLADE TECHNICAL ISSUES

Propellers, helicopter rotors, and wind turbine blades have the potential to produce most of their thrust and torque near their tips, where the dynamic pressure is highest. Because the moment arm is greatest at this point, a structure inboard to carry this moment is needed. Tapering a blade to have a larger chord inboard would achieve this end.

From the standpoint of aerodynamic efficiency, it is undesirable to have higher lift outboard than inboard on a blade. This situation creates a nonuniform inflow that produces vorticity in the wake, which causes induced losses. The optimum blade would yield maximum thrust and torque per kilowatt and produce constant inflow. Again, tapering a blade to have a larger chord inboard would achieve this end. Adding twist would also assist in the goal of achieving constant inflow.

Pultruded blades are manufactured using constant cross section over their full length. Thus, they are unable to exploit the inherent advantages, both structurally and aerodynamically, of the tapered and twisted blade described above. To achieve a structurally sound blade, the inboard area of the pultrusion must be reinforced to reduce the otherwise high stress at the blade-hub interface. Rotors with straight untwisted blades, have been shown to produce about 12% less kilowatt hours (kWh) than a rotor of equivalent power rating with full twist and taper.[2] This paper, however, will show that much of this performance penalty is recovered by adding a shaped aerodynamic fairing over the inner 40% of the blade.

Because a pultruded blade has higher mass outboard compared to a tapered blade of similar solidity, its natural bending frequency will be lower. This results in a more dynamically active blade, which can create operational problems such as tower strikes and stall flutter. Hansen examined yaw stability and tower strikes in some detail.[3] His general conclusion was that, through proper design, pultruded blades could be free of these troubles.

STALL FLUTTER

Stall flutter is a phenomenon experienced mostly in rotating helicopter, propeller, and wind turbine rotors.[4] For these devices, the interaction between the flow field and the blade torsional motion can sometimes lead to unstable torsional oscillations in stalled flow. This type of flutter does not involve flatwise motion, as with classical flutter, and is generally not influenced by the center of gravity location, except when the location effects the torsional natural frequency. There is some evidence that the location of the elastic axis has an effect, in that more aft locations are detrimental. A stall-flutter index, reduced velocity (RV), has been established, which is the ratio of 75% of the blade tip speed (ΩR) divided by the product of the first-mode torsional frequency (ω_T) and the semi-chord (c/2). The higher the reduced velocity, the more susceptible the blade is to flutter, and the risk increases the more the static-stall angle is exceeded. Figure 1 shows an approximate flutter boundary as a function of reduced velocity and blade angle of attack above the static-stall angle at the 75% radius position. The boundary was determined from propeller experiments. Normal rotary wing practice is to design for a reduced velocity of 1.5 or lower. This criterion was used for the 400 kW design studied under the NREL subcontract.

BLADE DESIGN FOR 400 kW ROTOR

The 400 kW rating can be achieved at various combinations of rotor diameter, tip speed, solidity, and blade pitch. The objective is to achieve the lowest total blade weight per kWh. Previous studies have demonstrated that total blade weight (W) decreases as the number of blades (b) increases when rotor solidity, blade material density, thickness ratio (t/c), and skin-thickness-to-chord ratio (t_S/c) remain the same.[5] That is, the higher the aspect ratio (R/c) for a given radius, the lower the weight per unit rotor swept area.

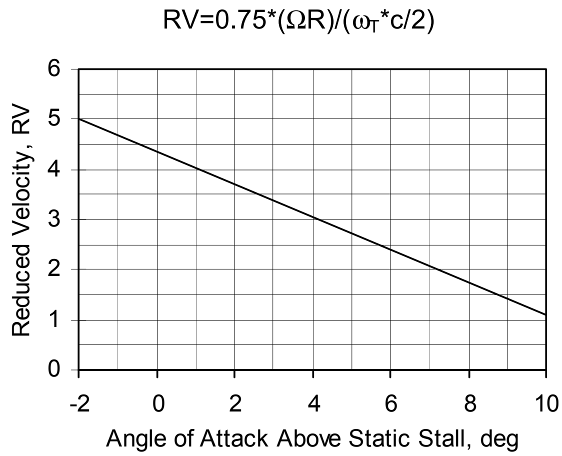

Figure 1. Stall-flutter boundary.

The maximum aspect ratio is determined by either of the two design constraints, reduced velocity or static stress. A hurricane wind of 62 m/s was selected as the static design condition. The minimum blade dimensions are calculated using the above criteria, and they can be determined before the radius is selected. Stress and stall-flutter are independent of rotor diameter for a given airfoil shape, except for the secondary influence of Reynold's number on drag coefficient and stall.

As shown in Figure 1, the stall-flutter boundary is a function of both tip speed and torsional frequency. Torsional frequency is a function of blade material properties, aspect ratio, and skin thickness. A 16% thick airfoil was selected for this study. A sketch of the cross section is shown in Figure 2. The root pads, which interface with the rotor hub, are also shown. The darkened leading-edge cavity represents high-density ballast used to control the center-of-gravity location.

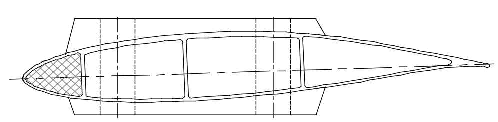

Figure 2. S813 Airfoil with root pads.

The torsional frequencies were calculated over a range of aspect ratios and skin thicknesses. The reduced velocities were then determined at various tip speeds. The combinations that produce the maximum allowable reduced velocity of 1.5 are shown in Figure 3. Except for tip speed, all parameters are nondimensional. For the same chord, torsional frequency is indirectly proportional to radius. Therefore, reduced velocity is directly proportional to tip speed and radius, and the relationship between tip speed and chord-to-radius ratio (c/R) is linear, as shown in Figure 3.

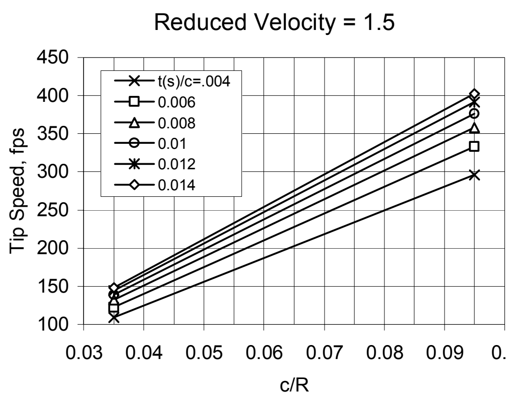

Figure 3. Maximum tip speed at stall flutter boundary.

In calculating the blade stress at the design condition, with the rotor parked in a 62 m/s steady wind, an iterative 20-segment beam analysis was used to determine the stabilized deflected position of the blade. A steady-state drag coefficient of 1.6 was used in the analysis.[6] To reduce the stress rise at the blade-hub interface, the inner one-third of the blade requires flatwise stiffening, either through a buildup of fiberglass on the upper and lower surfaces, or by adding an inboard fairing as described later in this paper.

Ultimate- and fatigue-strength tests of pultruded blade material were performed at Montana State University.[7] The fiberglass-vinylester coupons had ultimate strength in excess of 80,000 pounds per square inch (psi), with fatigue strength dropping to about 40% of ultimate at 4 million cycles. With safety factors for loads (1.35), materials (1.10), and stress concentrations (1.75), the maximum static design stress is 30,000 psi. The design process to achieve the lowest weight blade is an iterative procedure wherein aspect ratio, skin thickness, and tip speed are varied to satisfy the stall-flutter criterion (Figure 3) and the static stress limit of 30,000 psi. If blade-tip noise is a concern, the process can start by selecting the tip speed and then proceeding to the other parameters. The latter approach was taken for this design, with a tip speed of 55 m/s (180 fps). The non-dimensional values t_s/c and (c/R) that satisfy the design constraints can be found before determining the rotor dimensions. The weight per unit length of a blade is proportional to the product of material density, chord, and airfoil thickness. For any specified material, if the airfoil and skin thicknesses relative to the chord are maintained, it follows that blade weight changes as the square of the chord. Thus, for a given radius and total blade area, a rotor with more blades of smaller chord would weigh less than a rotor with fewer blades of larger chord. For example, a rotor with four blades would weigh half that of a two-blade rotor with twice the chord. The procedure is to enter Figure 3 at a tip speed of 180 fps and select the combination of skin-thickness and c/R that result in a static stress of 30,000 psi. In this illustration, the values of t_s/c and c/R are 0.012 and 0.044, respectively.

Many combinations of blade number and diameter will yield a peak power of 400 kW. The PROP93 computer code was used to determine these combinations,[8] which are shown in Figure 4 for several values of blade pitch. As the number of blades (solidity) is increased, a smaller diameter is required to achieve 400 kW.

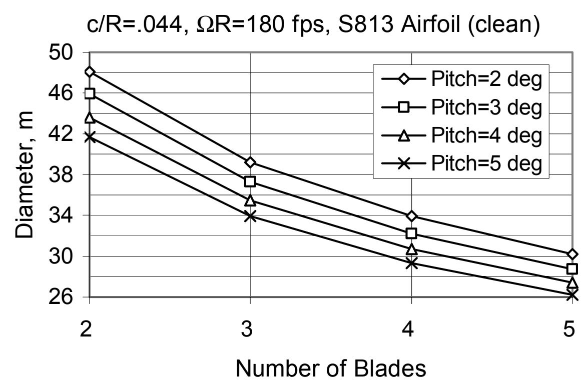

Figure 4. Diameter and number of blades required for 400 kW peak power.

Figure 5 shows the annual energy production (AEP) for these design cases assuming a 5.8 m/s wind site and a 40-meter hub height. The rotor diameters to achieve 400 kW peak power would increase if rough airfoil data were used instead of clean data. For example, for the 5-blade, 2^{o} pitch case, the diameter would increase from 30 meters to 33 meters.

Figure 6 shows the rotor weight, excluding hub, for the 400-kW designs depicted in Figure 4. The values given are the total of all blades, including the buildup of fiberglass on the upper and lower surfaces for root stiffening.

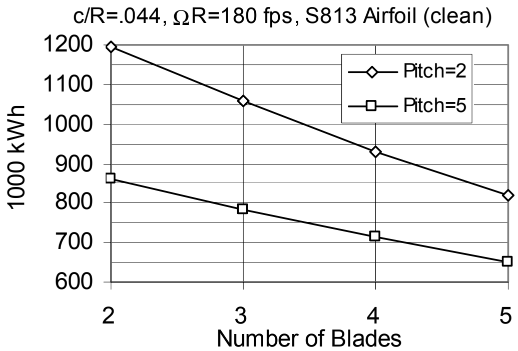

Figure 5. Annual energy production at 5.8 m/s.

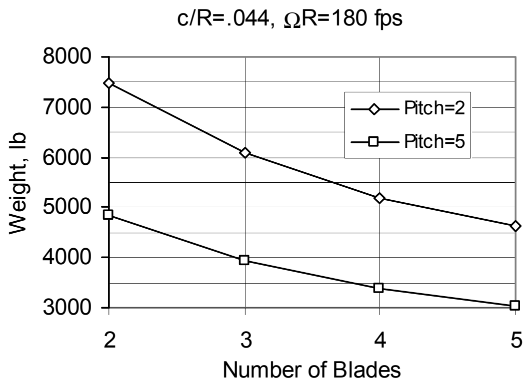

Figure 6. Rotor weight for 400 kW peak power.

Combining Figures 5 and 6 produces the specific rotor weight shown in Figure 7. The results, in pounds per kWh, which can be related almost directly to cost per kWh, show that increasing the number of blades generally reduces the rotor weight (and thus, cost) per kWh. The minimum occurs for five blades at a pitch angle of 5^o. Figure 6 indicates this configuration also has the lowest rotor weight of 3000 pounds. From Figure 4, this point corresponds to a diameter of 26.2 meters, and from Figure 5, an annual energy production of 650,000 kWh is indicated.

Although this optimization procedure showed that the smallest rotor resulted in the lowest *rotor* COE, this choice normally would not produce the lowest *system* COE. Because the rotor represents only one component of the turbine's total initial capital cost, the minimum rotor COE may not produce the minimum system COE.

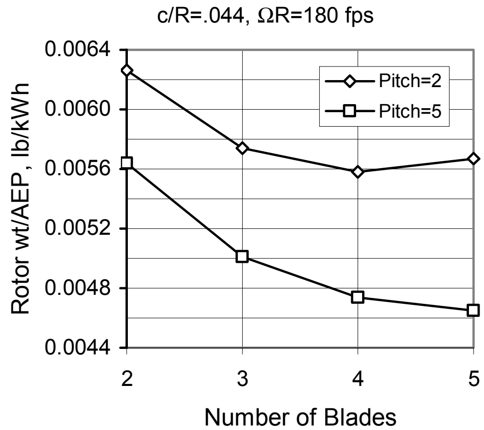

Figure 7. Specific rotor weight.

Usually, selecting a larger diameter (higher cost) rotor with greater energy production would result in a lower system COE. To this end, a 5-blade rotor with a diameter of 33 meters was selected for the remainder of the study. Its specifications are summarized in Table 1.

Table 1. Rotor Specifications

Diameter	33 m
Solidity	0.07
Number of blades	5
Blade chord	0.726 m
Blade c/R	0.044
Blade skin, t(s)/c	0.012
Airfoil	S813
Tip Speed	180 fps
Pitch	2 deg
Rated power	400 kW
Single blade weight	1204 lbs

HYBRID BLADES

The substantial advantages of pultruded blades provide ample incentive to explore methods of mitigating their lower aerodynamic efficiency. Therefore, a concept was investigated that added an inboard (root) fairing and a modular blade-tip extension to the straight pultruded blade. This concept, which we call the hybrid blade, is shown in its modular components in Figure 8.

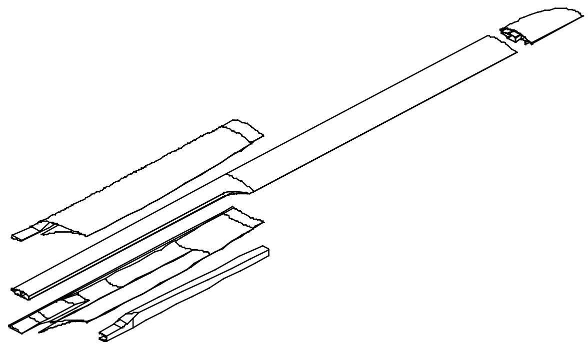

Figure 8. Hybrid blade with fairing and modular tip.

The twisted and tapered aerodynamic tip, with a span of approximately two chord lengths, may have an airfoil different from the basic pultrusion. Furthermore, its design may be highly refined to produce low blade-tip noise. Although its cost per pound is likely to be far greater than the basic pultrusion, its weight is so low as to have little detrimental effect on the cost advantage of pultruded blades. Aerodynamic studies indicate that the modular tip improves annual energy capture by 1-2% over the straight pultruded blade.

A companion element to the modular tip is a twisted and tapered inboard fairing having an airfoil section well suited to the inboard 30%-40% of the blade span. Preliminary studies indicate the root chord of the fairing can be twice that of the basic pultrusion, and a twist of 16^{o}-18^{o} is easily achievable. A planform modification such as this can recover 6%-8% of the performance loss suffered by a straight pultruded blade compared to an optimally twisted and tapered blade.

The use of an inboard fairing to improve aerodynamic performance is somewhat obvious, but the present study went beyond that application to investigate its use for structural reinforcement as well.

For the inboard fairing to provide a net benefit, it must have lower cost than the value of the energy production benefit it provides. The structural concept shown in Figure 8 was motivated by the desire to lower cost by minimizing both part count and fairing area.

The concept permits a tightly integrated geometry with direct bonds between the pultrusion and fairing noses, and between the lower corners of the pultrusion and the C-shaped shear web. Thus, only one web is needed to complete the structural unification of the fairing and pultrusion. The whole fairing assembly is composed of only three parts: upper surface shell, lower surface shell, and C-web. This same tight geometric integration, which allows a minimization of part count, also serves to minimize the area of the fairing, so the improvement in aerodynamic efficiency and energy capture can be obtained at low cost. Another attractive feature is a minimization of reinforcement that must be applied to the pultrusion, because the fairing can carry the blade flatwise bending moment over most of its length. Near the inboard end of the pultrusion, where the fairing ends, reinforcement is included by adding skin thickness to the fairing. This allows the same root fitting design, both with and without fairing, and the close mechanical coupling adds blade torsional stiffness. All of these factors played a part in choosing this concept from the many considered.

Figure 9 shows that the trailing edge of the pultrusion is cut away in the region of the root fairing. This is necessary to achieve the tight geometric integration discussed previously and to minimize the size and area of the fairing. The portions of the upper and lower fairing shells that contain the root-region reinforcing for the pultrusion are also evident.

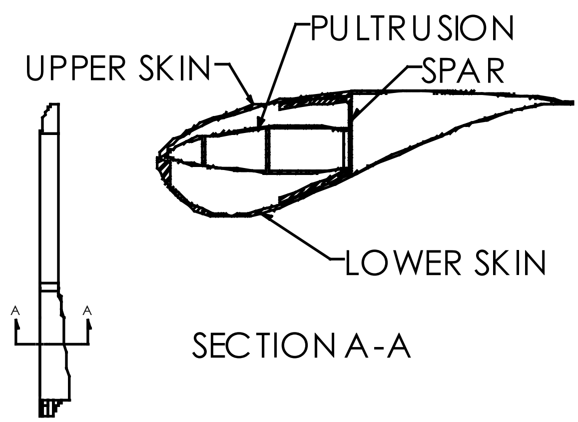

Figure 9. Cross-section of root fairing and pultrusion.

All pultruded material and molded parts require surface preparation prior to bonding. The installation sequence for the fairing is expected to begin with face-bonding the C-web to the aft face of the cut away pultrusion. With appropriate jigging to control alignment, the pultrusion and C-web would then be bonded into the lower fairing shell. Because the adhesive bonds at the inboard and outboard ends of the fairing are thought to be critical, this choice allows open inspection of those bonds during the assembly sequence. Completion of the fairing installation occurs with bonding of the upper shell, again with appropriate jigging to control alignment, and with the bonding of a closure rib, which is not shown in Figure 9. There are several noteworthy features of this concept:

- Structurally active, integrated spar caps located forward in the fairing shells are composed primarily of unidirectional fiber. Their function is to carry the inboard moment by forming an efficient beam section together with the C-web.
- Aft of the C-web, the shells are light and compliant to keep major structural loads concentrated near the components that are designed to carry them. Panels would have cored construction to provide needed panel strength and buckling stability, and the skins would be primarily double-bias material to provide high torsional stiffness.
- The pultrusion and fairing are bonded together over their full spanwise extent, so they bend together. This allows the fairing to carry inboard blade bending moment and eliminates issues of fairing movement relative to the pultrusion within it.
- To simplify fabrication, the twist and taper of the fairing are nearly linear.
- Because the fairing is large compared to the pultrusion, it should need less material to carry the inboard bending moments.
- The tightly integrated geometry permits the use of a single shear web–a simple, low-cost design.
- The configuration forms a triangular box from the pultrusion, spar cap, and C-web, thereby providing good structural support for the heavily loaded low-pressure-side spar cap.

Naturally, this structural concept presents some design challenges. Therefore, a number of issues must be addressed before a hybrid blade can be fabricated and tested. The purpose of developing the preliminary structural design described herein is to ensure that subsequent weight and cost estimates are based upon a plausible configuration and to facilitate the preparation of cost estimates by potential vendors.

PERFORMANCE

Of particular concern in considering pultruded blades is the aerodynamic performance penalty suffered because of the lack of twist and taper. Aerodynamic performance is discussed in terms of power coefficient (Cp), which is the ratio of power extracted to power available in the wind. Figure 10 shows Cp results from the PROP93 computer code.[8] It compares the power coefficient of the basic pultruded blade to the fully twisted and tapered blade and to the pultruded hybrid design discussed in the previous section. The blade pitch for each rotor was adjusted to achieve similar peak powers as shown in Figure 11. For the twisted-tapered and hybrid rotors, the inner 40% of the blades used high-lift airfoils having a maximum lift coefficient (Clmax) of 1.5. The outboard airfoil, as well as that of the basic pultrusion, had a Clmax of 1.1.

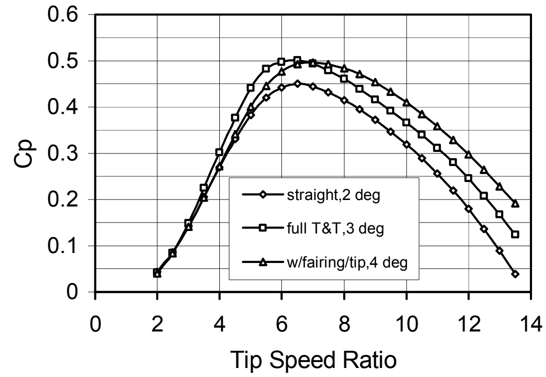

Figure 10. Cp comparison.

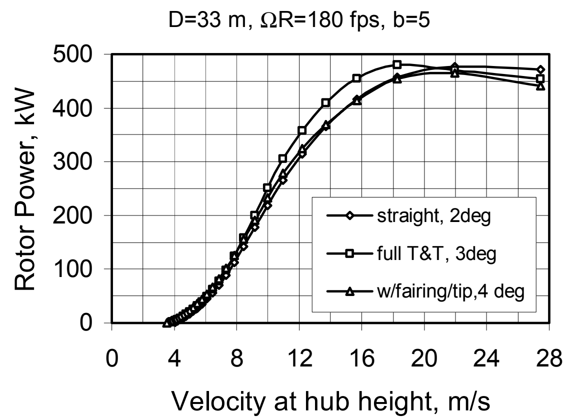

Figure 11. Power curve comparison.

The energy production corresponding to these power curves is shown in Figure 12. At a 6 m/s site, the annual energy production of the straight pultruded rotor and the hybrid rotor are 88.7% and 94.7%, respectively, of the fully twisted/tapered rotor.

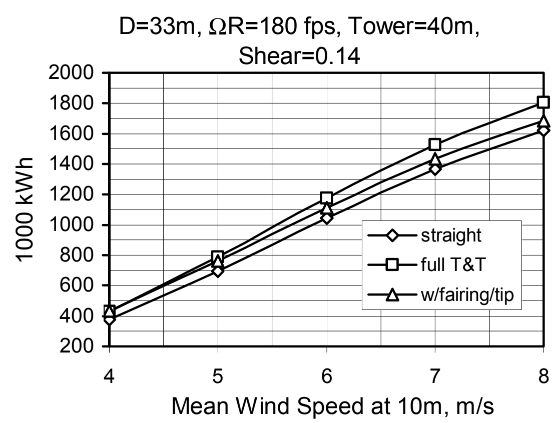

Figure 12. Annual energy production.

WEIGHTS AND COSTS

Weight and cost estimates were made for the rotor configurations described in the previous section. The pultrusion costs were based on actual vendor quotes. Initial tooling costs would be minimal when amortized over high production volume, so for this study they were not factored into the part costs. The cost of the root buildup was based on hand-lay-up fabrication, and resin transfer molding (RTM) was assumed for manufacturing of the fairings and tips. An additional 30% was added to the fairing weight to account for attachment materials. The weights and costs are summarized in Table 2, which shows that the pultrusion weight for the hybrid design is less than that of the basic pultruded blade. This is attributable to reduced skin thickness, because the fairing improves torsional stiffness out to 40% radius, thereby extending the stall-flutter boundary.

Table 2. Blade Weights and Costs

Blade config.	Base blade, b=5			Hybrid blade, b=5		
R=16.5m	Wt,lb	$/lb	cost, $	Wt,lb	$/lb	cost, $
Pultrusion	908	2.490	2,261	549	2.490	1,367
Doublers	190	6.000	1,140	0	0	0
Fairing	0	0	0	280	6.000	1,680
Tip	0	0	0	31	6.000	186
Ballast	106	0.500	53	95	0.500	12
Shipping	-	-	75	-	-	75
Blade	1204	2.869	3,454	955	3.476	3,320
All blades	6020	2.869	17,270	4775	3.476	16,598
MWh (6 m/s)	1,042			1,113		
$/kWh	0.0166			0.0149		

Estimated weights for a high-stiffness blade with full twist and taper are given in Table 3. Results are shown for a 5-blade rotor and a 3-blade rotor, both with the same solidity of 0.07 as the pultruded rotors in Table 2.

Table 3. Weights/Costs of High Stiffness Blade

# of blades, c	b=5, c(.75R) = 28.6 in			b=3, c(.75R) = 47.6 in		
R=16.5m	Wt,lb	$/lb	cost, $	Wt,lb	$/lb	cost, $
Base blade	1624	6.000	9,744	4507	6.000	27,042
Shipping			100			100
Blade	1624	6.062	9,844	4507	6.022	27,142
All blades	8120	6.062	49,220	13521	6.022	81,426
MWh (6 m/s)	1,175			1,152		
$/kWh	0.0419			0.0707		

It is clear there is a significant weight penalty with increased stiffness. Table 3 also shows that increasing the number of blades for the same rotor solidity reduces total blade weight by 40%. It should be noted that the weight estimates for the tapered blade were based on a single-skin airfoil with a skin thickness of 1.1% of the chord. This was the average value necessary to match the weight data obtained from the literature of several manufacturers.

The most important comparisons from Tables 2 and 3 are those showing blade costs. The pultruded blades, while suffering a small performance penalty, have a cost per kWh only 35% that of the 5-blade twisted and tapered rotor. This translates into a 6% reduction in COE, assuming the original blade cost represented 20% of the turbine cost, and the turbine was 80% of the initial capital cost.

SYSTEM DYNAMICS

An assessment was made of the system dynamics of these pultruded rotors, with particular attention given to yaw behavior, tower strikes, cyclic loads, and fatigue damage.[3] To determine if undesirable yaw behavior might be a potential operating problem, yaw data from the scaled-rotor field tests were examined. Four data sets covering winds up to 22 m/s were studied. The typical turbulence intensity for these cases was 24%. Yaw motion was judged to be stable in all cases, with yaw errors of approximately $+10^0$ in low wind and -30^0 in high wind. As expected, there was less yaw activity observed in the lower winds. Flap motion, which also appeared to be stable, was dominated by one-per-rev (1p) oscillations driven primarily by yaw rate. The blade-root in-plane bending moment was also dominated by 1p, although there were some events at approximately 3p, which is the first in-plane bending-mode natural frequency.

Two computer simulations were developed to evaluate the phenomenon of tower strikes. The first used the YawDyn code[9] and the second used the ADAMS code.[10] The validity of the YawDyn model was established by replicating tower strikes known to occur on the Windtech 75, which employed a highly flexible 2-blade rotor. To demonstrate the efficacy of the current design approach, a dynamically-scaled version of the 400kW design, which employs stiffer blades, was subjected to the same operating conditions. For this case, a large blade-tower clearance was predicted.

This rotor was also modeled in ADAMS and subjected to a variety of wind conditions and rotor speeds in an attempt to simulate a tower strike. These conditions included extreme direction changes at 12 and 24 m/s; imposed yaw errors from -180^o to $+180^o$ in winds from 8 to 24 m/s; and a variety of wind gusts. The only conditions that produced a tower strike were at wind speeds above 20 m/s with a yaw error of 140^o or greater. However, it is not expected that a yaw error could reach this magnitude under normal operation.

Blade-root fatigue damage was also investigated. The analysis considered contributions from the blade-root flapping moments only. Partial safety factors were 2.13 on stress and 1.4 on materials. Wind inputs for an International Electrotechnical Commission (IEC) Class 2 design were used, and simulations were run at turbulent wind speeds of 8, 12, 16, 20 and 24 m/s. Fatigue strength of the pultruded blade material was obtained from Mandel and Samborsky.[7] Rainflow cycle counting followed by damage-accumulation estimates showed that the predicted fatigue life was well in excess of 30 years. This study was preliminary in nature, and a more detailed analysis, including edgewise loads, at other blade locations would normally be performed to obtain better estimates of fatigue life. However, it appears that an acceptable fatigue life is achievable for this blade design.

FIELD TESTS

The NREL subcontract included the fabrication and testing of a 5-blade flexible rotor utilizing pultruded blades. The 400 kW design with a diameter of 33 m was scaled to a rotor diameter of 15.5-m (51-ft) and a blade chord of 0.34 m (13.5 in). All nondimensional blade frequencies of the 400 kW design were maintained for the scaled version, so that the blade dynamic characteristics could be simulated. The rotor, including a new spoiler-flap for overspeed control, was mounted on an available chassis and tower and installed at a site in San Gorgonio Pass near Palm Springs, California. The turbine's power train was rated at 80 kW and the rotor speed was constant at 69 rpm. The turbine operated downwind, was free in yaw, and supported by a 16.8-meter freestanding tubular tower. The purposes of the test were to demonstrate dynamic stability of the rotor, measure rotor performance and blade loads, compare results with theory, and provide validation of the design codes.

The test rotor was instrumented to measure root strains on two blades, spoiler-flap position and loads, and low-speed shaft torque. A sonic anemometer, installed upstream of the rotor, measured all three components of inflow velocity. The prevailing wind was from the west, but there was a considerable north-south unsteadiness present.

Although the field tests were limited to a 5–week test period because of a gearbox failure, they demonstrated some important characteristics of rotors using flexible pultruded blades. These were (a) absence of stall flutter up to wind speeds of 24 m/s, (b) stable yaw behavior, (c) acceptable operating loads, and (d) predictable power performance.

The rotor power, derived from 10-second averages of low-speed shaft torque, is shown in Figure 13. The power curve predicted using PROP93[8] is also included. This curve was based on 16%-thick airfoil data measured in the Ohio State University wind tunnel.[11]

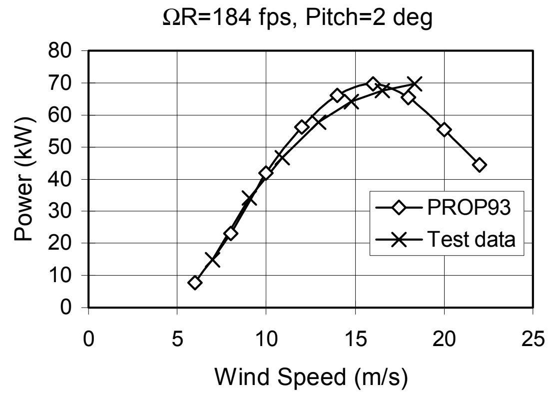

Figure 13. Scaled-rotor power.

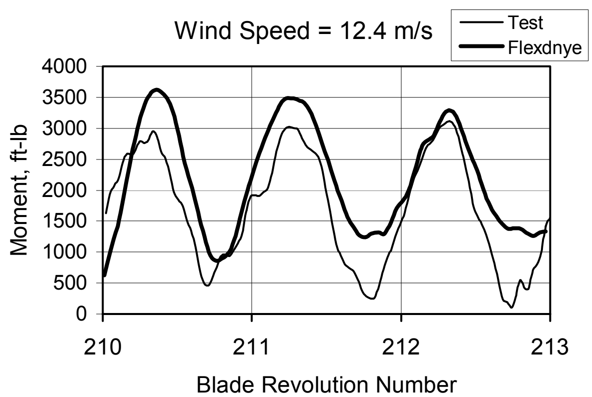

Figure 14. Scaled-rotor root flatwise moment.

Blade-root bending moments were derived from the measured strains, and sample results are presented in Figures 14 and 15. Predicted results from the FLEXDYNE aeroelastic code[12] are also included. The program can also perform rainflow cycle counting to assist in determining fatigue damage. A sample of this feature is presented in Figure 16, where the predicted peak-to-peak (PTP) flatwise moment cycles are compared to scaled-rotor test results. The measured wind, including the high frequency components provided by the sonic anemometer, are used as input to the program. FLEXDYNE is a modal analysis in which differential equations of motion for coupled bending-torsional deformations of blades subjected to yawing motions about a vertical tower are solved by numerical

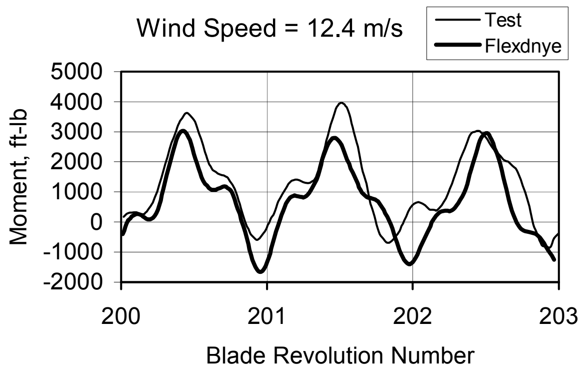

Figure 15. Scaled-rotor root edgewise moment.

integration. It simulates the first five fully-coupled modes of the flatwise, edgewise, and torsion degrees of freedom. The turbine yawing degree-of-freedom is included, and turbulent gusts can also be simulated. Additional features include tower shadow and tip-brake aerodynamics. The program was written for a personal computer using a spreadsheet format.

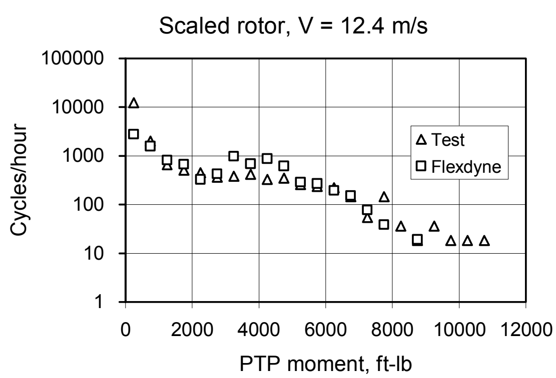

Figure 16. Comparison of root moment cycles/hour.

CONCLUSIONS

An investigation was conducted to evaluate various issues pertaining to the use of pultruded blades for wind turbine rotors. The technical areas that were studied included materials, blade design, stall-flutter, yaw stability, tower strikes, power performance, airfoils, blade aerodynamics, hub design, rotor weight, modifications to improve performance, and rotor costs. In support of these analytical efforts, field tests were conducted of a 15.5-m rotor, dynamically scaled from a 33-m utility-grade design.

No technical issue was revealed in any of the studies or tests that would be considered a ‘fatal flaw’ in the concept. Furthermore, cost studies comparing rotors using pultruded blades with rotors having conventional tapered and twisted blades showed exceptional reductions in rotor cost per kWh. Specifically, the following conclusions may be drawn from the study:

- The pultrusion process applied to wind turbine blades is capable of producing excellent ultimate- and fatigue-strength properties in comparison to materials manufactured by more traditional blade fabrication processes.[7]
- Acceptable fatigue life is indicated for the pultruded blades manufactured in this study.
- Estimated blade weight for a hypothetical 5-blade rotor is approximately 40% less than for a 3-blade rotor.
- Field tests of a 5-blade 15.5-m rotor demonstrated stable yaw behavior as indicated by smooth motion and good yaw tracking.
- Structural dynamic simulations show no cause for concern over tower strikes for the tested rotor.
- Energy capture degrades by approximately 12% for straight pultrusions compared to blades with twist and taper.
- Energy capture degrades by approximately 5% for hybrid blades with root fairings and modular tips.
- The use of pultruded blades, either alone or in combination with root fairings and modular tips, has the potential to significantly reduce the cost of wind turbine rotors.

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