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A Retrospective of VAWT Technology

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A Retrospective of VAWT Technology

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Abstract

The study of Vertical-Axis Wind Turbine (VAWT) technology at Sandia National Laboratories started in the 1970's and concluded in the 1990's. These studies concentrated on the Darrieus configurations because of their high inherent efficiency, but other configurations (e.g., the Savonius turbine) were also examined. The Sandia VAWT program culminated with the design of the 34-m 'Test Bed' Darrieus VAWT. This turbine was designed and built to test various VAWT design concepts and to provide the necessary databases to validate analytical design codes and algorithms. Using the Test Bed as their starting point, FloWind Corp. developed a commercial VAWT product line with composite blades and an extended height-to-diameter ratio. The purpose of this paper is to discuss the design process and results of the Sandia 34-m VAWT Test Bed program and the FloWind prototype development program with an eye toward future off-shore designs. This paper is our retrospective of the design, analysis, testing and commercial process. Special emphasis is given to those lessons learned that will aid in the development of an off-shore VAWT.

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INTRODUCTION

The study of Vertical-Axis Wind Turbine (VAWT) technology at Sandia National Laboratories started in the 1970’s and concluded in the 1990’s. Initial studies at SNL investigated numerous VAWT configurations, including the Savonius and the Darrieus configurations. These studies soon concentrated on the Darrieus configurations because of their high inherent efficiency. Over the years, the designs proposed and tested by Sandia grew in size and power production. The Sandia-designed 17-m (rotor diameter) research machine proved to be very successful and was commercialized by several companies. One of these companies, the FloWind Corp., built and placed over 500 turbines in commercial operation, primarily in the Altamont Pass in California.*

The Sandia VAWT program culminated with the design of the 34-m ‘Test Bed’ Darrieus VAWT. This turbine was designed and built to test various VAWT design concepts and to provide the necessary databases to validate analytical design codes and algorithms. Innovations in the design included airfoils specifically designed to optimize VAWT performance and variable-speed operation. Based on the results from the comprehensive test program for this turbine, an extensive suite of design codes was developed and validated.

Using the Test Bed as their starting point, the Sandia design team re-engineered the 34-m research VAWT into a commercial VAWT, the Point Design, a design for deployment in production wind farms. This design was presented to a number of potential developers in 1990.

FloWind Corporation, under the auspices of various government/private industry partnership programs with Sandia and NREL, used the Point Design as a starting point for their development of a VAWT product line. Their business plan started with the design of a new rotor for their existing fleet of 19-m turbines. With Sandia’s and NREL’s aid and guidance, FloWind designed an innovative, commercial 3-bladed composite rotor with an extended height-to-diameter ratio. A prototype of this turbine, the 18-EHD turbine, demonstrated that VAWTs were a viable technology for the commercial production of power for the electric grid. However, financing for a production fleet could not be obtained and FloWind closed its doors. After the closure of FloWind in the mid-1990s, VAWTs were out-of-favor with the wind turbine community. Commercial VAWTs were not pursed to any great extent, and almost all of the US DOE-sponsored VAWT-specific wind turbine research was terminated.

With the realization that off-shore winds offer a significant resource for power generation from wind turbines located near population centers, the investigation of VAWT technology is again being examined. This examination is predicated on the inherent characteristics of the VAWT designs that make them particularly favorable for the production of power in an off-shore environment. The primary VAWT characteristics

* FloWind Corp. is no longer in business.

that favor off-shore configurations are that all of the heavy equipment associated with a power generation, namely the transmission and the generator, are typically mounted below the rotor. This configuration permits off-shore designs to place these components at or below water level; thus, providing additional stability to the platform (structure) that supports the rotor and reducing its capital costs. Only the rotor and its center tower need to be above the water surface. And, as the VAWT is as efficient as the more conventional Horizontal-Axis Wind Turbine (HAWT), no penalties are paid in performance.

Another feature of VAWTs is that they operate with winds from any directions; they do not need a yaw system. The lack of a yaw system increases the reliability of the turbine while decreasing its capital costs. The elimination of the yaw system is particularly important in multi-megawatt turbines, where the yaw system is required to yaw very large turbine rotors and their associated drive-train components.. The design and operational experience for multi-megawatt HAWTs has illustrated their yaw systems are both expensive to build (large capital costs) and to operate [large Operation and Maintenance (O&M) costs].

VAWTs do have some inherent short-coming that have limited their use in commercial ground-based wind farms. Primary among these are economic concerns about the capital cost of the blade. That is, the blade of a full-Darrieus VAWT is approximately twice as long as that of a HAWT with equivalent rotor swept area. Thus, the blades for a VAWT may cost significantly more than equivalent blades for a HAWT. In the past, innovative blade designs and materials were used to reduce the cost of a VAWT blade. These designs included constant-chord blades made from extruded aluminum or pultruded fiber-glass composites. The resulting VAWT blades could be manufactured at essentially the same cost as their shorter, twisted and tapered HAWT counterparts. However, these blade technologies, with constant-chord blade profiles, did sacrifice some aerodynamic efficiency for the sake of system costs.

The question now before us is "Do the advantages of the VAWT outweigh its disadvantages when they are configured for off-shore sites?"

Before this question is addressed, one must ask "What do we currently know about the design of VAWTs and can the lessons learned from the initial design efforts of the 1970's, 80's and 90's provide insights into better VAWT designs and configurations?" A review and assessment, funded by the U.K. Department of Trade and Industry, (mainly authored by L. A. Schienbein and D. J. Malcolm) of the various worldwide VAWT research efforts was completed in 1994.[1] This report summarizes the state-of-the-art in VAWT technology at that time.

The purpose of this paper is to discuss the design process and results of the Sandia 34-m VAWT Test Bed program with an eye toward future off-shore designs. All three authors of this report, in their capacity as researchers/engineers at Sandia, were directly involved in the design, testing and commercialization of the 34-m Test Bed turbine. This paper is their retrospective of many aspects of the design, analysis, testing and commercialization process. Special emphasis is given to those lessons learned that may aid in the development of an off-shore VAWT.

After these discussions, a short discussion of several proposed cantilever VAWT designs is presented. This section is designed to place currently-proposed off-shore designs into their historical prospective with a short review of the research on the "H" VAWTs conducted in England in the 1970's and 1980's. The manuscript concludes with our prospective on how an off-shore VAWT might be configured.

This report deals in generalities. The purpose of this report is to summarize results and conclusions and not to provide technical details. All of the technical details have been published previously and most of these technical reports are available in the open literature and are reproduced on the Sandia Wind Web site[†] as downloadable files. The authors would especially draw the reader's attention to the report entitled *Selected Papers on Wind Energy Technology*.[2] This report, edited by Paul Veers, contains 16 technical papers that summarize the initial test results (1988 through 1991) from the 34-m Test Bed. Another important source of these publications is the annual proceedings of the ASME Wind Energy Symposium. This annual symposium has been, and still is, a major forum for the presentation of results in wind technology. In particular, a large body of detailed technical information on the Test Bed was released at the Eighth (1989) and Ninth (1990) Symposiums.

In addition to technical reports that are available in the open literature, the authors draw upon unpublished internal reports and program reviews. A complete list of the available, primary reports used in the preparation of this report are cited below in the section entitled "Bibliography."

[†] The current Sandia Web site is: http://windandwaterpower.sandia.gov

VAWT TECHNOLOGY

VAWTs are wind turbines that rotate about a vertically-oriented axis that is perpendicular to the wind direction (sometimes termed a "cross-flow" turbine). In typical, modern designs, the center axis is a vertical shaft (tower)[‡] that is connected to a speed-increasing transmission (gearbox). The transmission's output shaft, in turn, drives a motor/generator that converts the mechanical torque of the rotor to electrical power. Typical designs include the full-Darrieus (or eggbeater), the "H", the "V" (or "Y" or "sunflower"), the "Delta", the "Diamond" and the "Gyromill" configurations, all of which may be seen in Figure 1. Many additional configurations have been proposed, even some that turn the turbine on its side (a "squirrel cage" configuration).

The significant distinction between a HAWT and a VAWT is the orientation of their rotational axis relative to the wind direction. The HAWT's propeller type rotor is aligned with its axis of rotation essentially parallel to the direction of the wind and the VAWT's

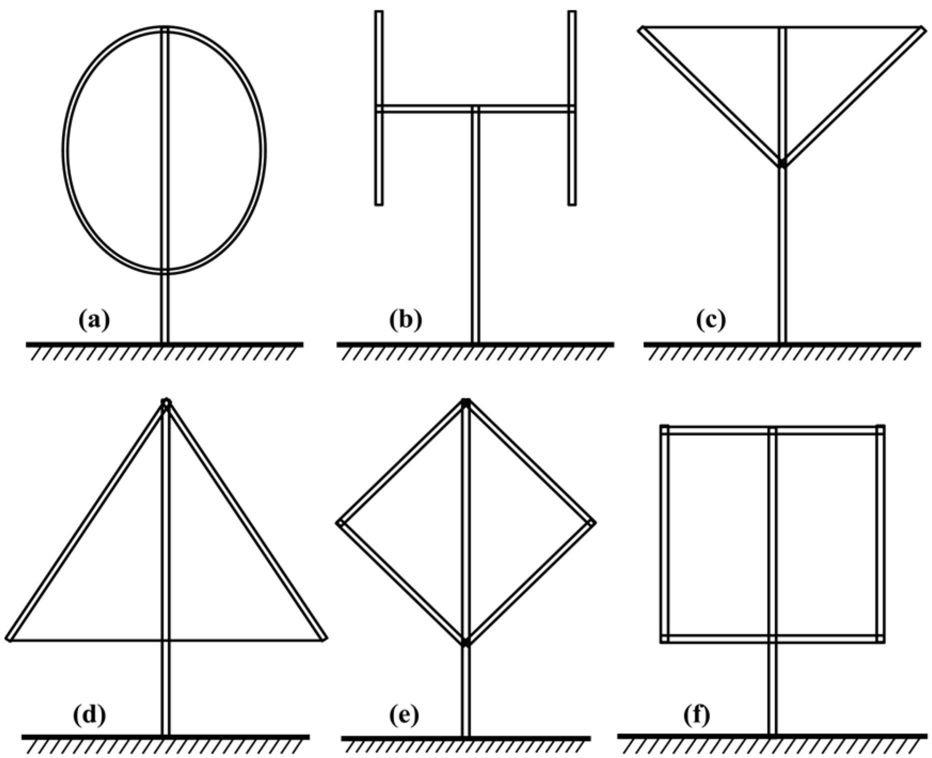

Figure 1. Basic VAWT Configurations: (a) Full Darrieus, (b) "H" , (c) "V", (d) "Δ", (e) "Diamond" and (f) "Giromill".

[‡] If the tower turns with the rotor, it is also referred to as the torque tube.

rotor is aligned with its axis of rotation essentially perpendicular to the direction of the wind.[3]

VAWTs tend to come in two main configurations: the Savonius and the Darrieus turbines. In the case of the former, power is generated using momentum transfer (a drag device) and, in the latter, using aerodynamic forces (the lift force on an airfoil). The Savonius in characterized by its high torque, low speed and low efficiency (less than half the Betz limit). The Darrieus rotor is characterized by its high speed and high efficiency (approaching the Betz limit§).[4]

Savonius turbines use rotors that typically have a "bucket" design. These rotors, studied by numerous investigators since the 1920s, have been used extensively in high-torque low-speed applications, i.e., water pumping and ventilation.[5] These rotors cannot compete with other configurations on an aerodynamics performance basis, but their ease of fabrication has yielded many applications, particularly in developing countries and do-it-yourself projects. A 2-bucket design reaches maximum efficiencies in the mid-twenties percentage range and a 3-bucket design reaches efficiencies in the high-teens percentage range. These efficiencies are obtained at rotational velocities (at the outside edge of the rotor) that are significantly less than the inflow wind speed. Practically, the efficiency is, at the very best, thirty percent and is only obtained at very low rotation rates. At higher rotational rates, the efficiency of the rotor decreases dramatically.

The Darrieus configuration was patented by the French inventor Georges Jean Marie Darrieus in France in 1925 and in the US in 1931.[6] This configuration was reinvented in the late 1960s in Canada by Rangi, South and Templin.[7]

The full-Darrieus-rotor VAWT[8] is a high efficiency design, similar to that shown in Figure 2, whose aerodynamic efficiency approaches the Betz limit, as does its HAWT counterparts. All of its heavy equipment (i.e., the gearbox and the generator) are stationary and located

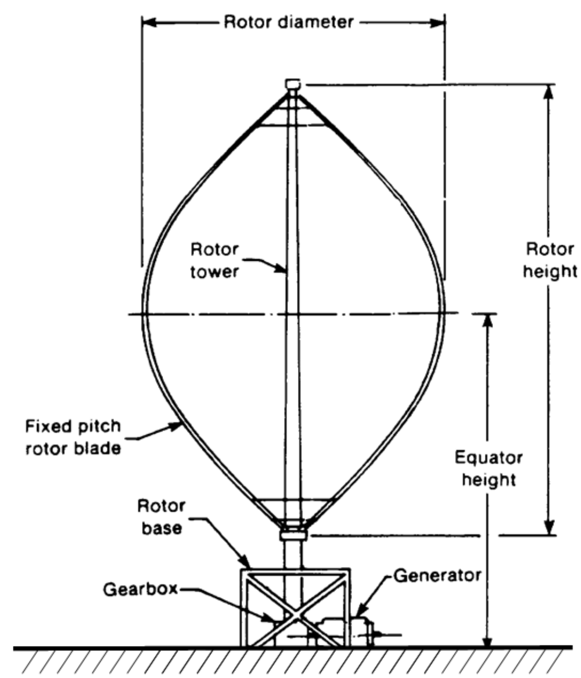

Figure 2. Typical Full-Darrieus Turbine Layout

§ Betz's law (or limit), developed in 1919 by the German physicist Albert Betz, states that no turbine can capture more than 59.3 percent of the kinetic energy in wind.

at or near ground level, where they are easy to mount and maintain. This is in contrast to HAWTs where this equipment is mounted at the top of a support tower and moves with the nacelle when the blades are aligned (yawed) with the wind.

By the very nature of the design, the blades of the VAWT move essentially perpendicular to the direction of the wind. Thus, this design works equally well with winds from any direction, enabling it to easily accommodate horizontal and directional wind shear. This is in contrast to HAWTs, which must be yawed into or out of the prevailing wind using aerodynamic forces or a mechanical system and which cannot readily accommodate strong wind shears.

As mentioned in the Introduction of this manuscript, one main perceived disadvantage of the Darrieus-rotor VAWT is that its blades are essentially twice as long as those of a HAWT counterpart of comparable swept area. However, in the traditional full-Darrieus design, the blades are connected to the center tower at both ends; thus, the VAWT blades are loaded mainly in tension and can be made lighter than their cantilevered HAWT counterparts. Also, the blades can be manufactured with constant chord and no twist with only a small effect on the aerodynamic performance of the rotor. When the blades are shaped into a troposkien,[**] their flatwise (radial) bending stresses during operation are reduced to essentially zero and the blades are loaded only in tension, a very favorable loading scenario for composite materials.

In the 1970's, 80's and 90's, these blade characteristics yielded VAWT designs that utilized aluminum blades that were shaped into the necessary airfoil cross section using an extrusion process to minimize manufacturing costs. Unfortunately, in this time frame, the fatigue loads on wind turbines (both VAWTs and HAWTs) were not well understood, and the properties for extrudable aluminum (i.e., 6063 T5) yielded blades with poor fatigue properties. The resulting premature failures of the VAWT aluminum blades, especially at joint structures where large stress concentrations were present, led to the perception that VAWTs were inherently prone to fatigue. In reality, VAWTs are no more prone to fatigue failure than are HAWTs, With the current understanding of fatigue loads, VAWT blades that reliably withstand the fatigue loads imposed upon them can and have been designed.

The lighter structural designs of the VAWT blades do lead to large flexures (both static and dynamic) of the blade. In many cases, the blades must be reinforced using struts. While struts provide the necessary stability at minimal capital costs, they may cause a significant reduction in rotor performance by introducing aerodynamic drag at the strut-to-blade joint(s). They should be placed as close to the blade-to-tower joints as possible to minimize this reduction. Aerodynamic fairings around these joints have had mixed results in reducing this joint drag and restoring rotor efficiency.

[**] The troposkien shape is the curve formed by spinning a rope with a constant angular velocity and with its ends anchored. Structurally, the rope is under only tensile forces along its length (a rope cannot support bending loads). The troposkien shape varies depending upon the orientation of the axis (vertical or horizontal), due to the action of gravity.

Although most HAWTs are self-starting, the Darrieus-rotor VAWT may or may not self-start, depending upon the wind conditions. Thus, to ensure that a VAWT is started when desired, the turbine must be equipped with a starting system. Typically, this system uses the generator as a motor to rotate the rotor until it has reached sufficient speed to start producing power. Although this is a relatively simple solution to the need for a starting system, it imposes the requirement that the gearbox be bi-directional, thus increasing the capitol costs of the turbine.

A VAWT blade produces positive torque when it crosses the wind and produces little or negative torque when it moves parallel to the wind. Thus, each VAWT blade produces two "pulses" of torque on each revolution. In even-number bladed VAWTs and, in particular, 2-bladed VAWTs, these pulses align, producing a highly variable output torque that approaches a sinusoid with a positive mean. As the gearbox and the generator do not operate well with a highly varying torque, the VAWT power train can be problematic. However, even with the earliest designs, this problem was handled effectively simply by adding compliance (in torque) to the drive train. The Test Bed design included such a coupling.

Many VAWT designs use guy-cables as a cost effective technique to stabilize the top of the rotor (obviously not the "H," "V" and "Giromill" configurations shown in Figure 1). However, this design does present some problems. When guy cables are used, the main support bearing at the bottom of the rotor must be designed to not only support the rotor weight, but also the downward force due to tension in the cables. A thrust bearing is also needed on the top of the rotor to allow for the rotation of the rotor beneath the cables. The required increase in capacity of these bearings due to the cable can contribute significantly to the capital cost of the turbine. Also, the guy cables and their anchors give the turbine a large footprint. For typical land-based installations, this footprint is usually not a problem, but for turbines in farming country, this large footprint can be detrimental. For off-shore installations, it is difficult to envision how guy cables could be effectively used.

Finally, active aerodynamic controls (i.e. variable-pitch blades and aerodynamic brakes) are relatively difficult to implement in VAWT designs. When fitted with conventional, airplane type airfoils, the output power of a fixed-speed VAWT increases monotonically with increasing wind speed. Without proper controls, this output can overdrive the system, leading to a "run-away" turbine that self-destructs. However, modern airfoil designs (for both VAWTs and HAWTs) have yielded airfoils that shed loads at high inflow wind speeds (through a controlled stall of the blade) and alleviate this problem; these have resulted in stall-controlled turbines. Variable speed operation can also be used to reduce excessive output power by lowering the rotation rate of the rotor to prevent it from reaching a "run-away" condition.

THE SANDIA 34-M TEST BED

The Sandia 34-Meter Test Bed Turbine, shown in Figure 3, was a full-Darrieus VAWT, 34-m in diameter, designed, fabricated and built by Sandia National Laboratories to provide a test-bed for research in aerodynamics, structural dynamics, fatigue life prediction and control algorithms. The Test Bed, dedicated in May 1988, was located on the U.S. Department of Agriculture (USDA) Agricultural Research Station (ARS)/Conservation and Production Station (CPS) near Bushland, Texas. At that time, the Test Bed was the largest VAWT in the U.S. The machine was a variable speed (operation range of 28 to 38 rpm) turbine with a height-to-diameter ratio of 1.25. It was rated at 500 kilowatts of electrical power at 37.5 rpm in a 12.5 m/s wind.[9]

The Test Bed was designed as a one-of-a-kind research machine. The philosophy used in its design was to provide a conservative design; optimizations for a commercially viable design were not considered. In fact, the turbine was designed to withstand the loading associated with single-blade operation (without a counter balance). This design philosophy was dictated because many of codes used in the design of this turbine had yet to be validated. Indeed, one of the major objectives of this project was to validate the suite of codes used in its design. The turbine design was also modular, in that components, including blade sections, could be changed-out as warranted by research needs.

The turbine and its site were equipped with a large array of sensors that permitted the characterization of the turbine under field conditions. A total of 72 strain signals was used to measure the mechanical response of the blades and tower.[10] In addition, twenty five (25) environmental signals, 22 turbine performance signals and 29 other electrical signals were used to fully characterize the turbine environment and performance.

The Test Bed was used for research until it was decommissioned and removed from the Bushland site in the late spring of 1998.

The myriad of data resulting from the tests conducted on this turbine[11,12] was used to validate analysis techniques and to demonstrate the improved aerodynamic and structural performance of advanced components, including natural laminar flow airfoils, step-tapered blades, and variable speed operation. The Test Bed clearly emerged as a stepping stone in the VAWT development process and

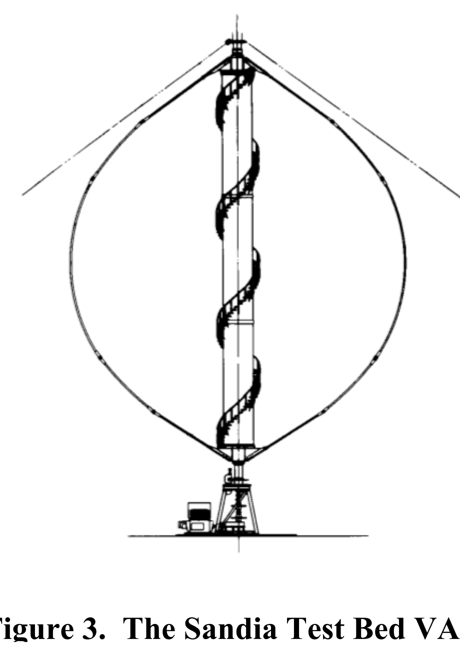

Figure 3. The Sandia Test Bed VAWT

served as an important benchmark in developing the ability to predict turbine aerodynamic and structural performance using analytical techniques.

THE ROTOR

The rotor used for the Test Bed was a conventional design for that period in time. It was designed with two blades with a height-to-diameter ratio of 1.25. The rotor was a full Darrieus rotor with fixed blade attachments to a center tower that rotated with the rotor (i.e., a torque tube). The rotor was stabilized with 3 sets (2 cables per set) of fixed guy cables. The aluminum blades were constructed using step-tapered chord airfoils that were permanently bent into a troposkien shape to minimize bending loads.

THE BLADES

The blades were constructed from 6063 aluminum using an extrusion process. The extrusion process was chosen to minimize blade costs. As this process produces straight constant-chord sections, the blades were designed in a step-tapered chord configuration. The final design used sections constructed using airfoils with chords of 1.22 m (48 in), 1.07 m (42 in) and 0.91 m (36 in). The 1.22 m section used a NACA 0021 profile, and the other two used a SAND 0018/50 profile. The solidity of the rotor was 0.13.

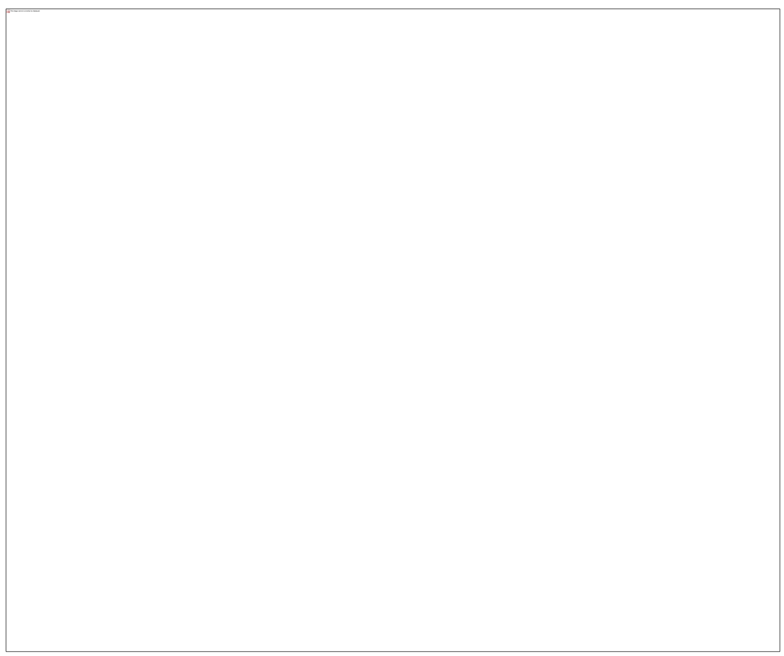

Figure 4. Blade Section Assemblies

The desired blade chords were too large for a single extrusion. Each blade section was formed using multiple extrusions, as shown in Figure 4, that were bolted together along the span. The 1.22 m section required 3 extrusions, and the 1.07 m and 0.91 m sections each required 2. These extrusions were the largest open-cell aluminum extrusions that had been produced to that time. The straight extruded blade sections were permanently bent into their final troposkien shape and then cut to length. The joint structures were mounted to them using bolted clam-shells, as illustrated in Figure 5. Once the blades were assembled and mounted to the torque tube, the span-wise joints were smoothed with body compound in an effort to minimize the performance penalty associated with the use of this joint design.

AERODYNAMIC DESIGN

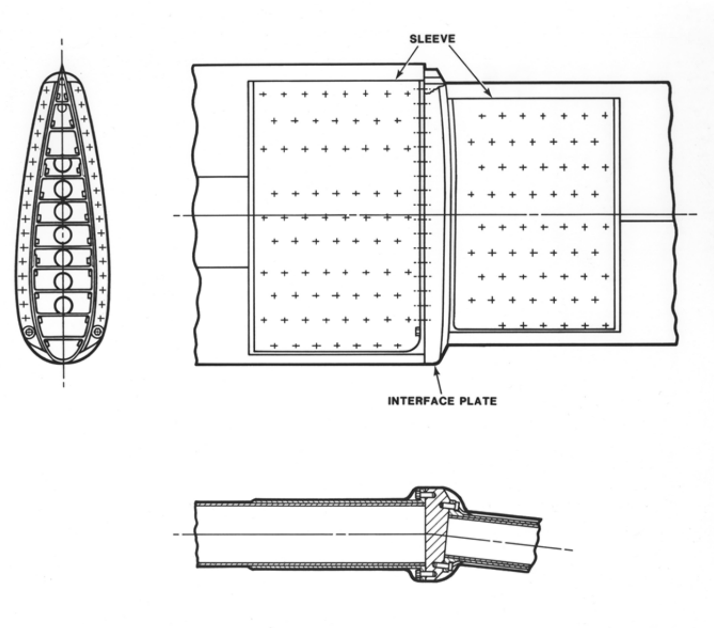

Figure 5. Blade-to-Blade Joint Detail

As mentioned above, the Test Bed blades used the SAND 0018/50 and the NACA 0021 airfoils. The SAND 0018/50 airfoil is a member of symmetric, natural laminar flow (NLF) airfoil profile sections designed specifically for VAWT applications by Klimas, Berg, and Gregorek.[13,14,15] These airfoils were the first airfoil family designed specifically for wind turbine applications. The stall characteristics of these airfoils were designed to regulate the output power of the rotor in the high-wind regime, enabling the creation of a stall regulated rotor. These airfoils were also designed to be less sensitive to leading edge roughness than the similar thickness NACA 00XX airfoil series, to reduce the variation in turbine power as the blades erode and become contaminated with bug debris over a period of time.

MECHANICAL DESIGN

The mechanical design of the rotor used step-tapered blades composed of 5 sections. The equatorial section was 19.1 m (62.7 ft) in length with a 0.91 m chord. The two transition (between the equatorial section and the root section) sections were 7.5 m (24.6 ft) in length with a 1.07 m chord, and the two root sections were 9.2 m (30.2 ft) in length with a 1.22 m chord. The rotor design did not require the use of struts.

The blades were bent into a troposkien shape to minimize bending loads during operation. The blade-to-blade joint structures (4 per blade) introduced concentrated masses along the blade length which were included in the determination of the desired troposkien shape. To facilitate the desired shape, the joints were not straight, rather they introduced a step change into the angular orientation of the blade, as shown in Figure 5. The change in angle at each joint was chosen to achieve the closest possible approximation of the desired theoretical troposkien.

Modal Response

The natural frequencies of the stationary Test Bed were measured with a modal test and compared to structural predictions. Several structural mode shapes for the turbine are shown in Figure 6.[16] Most of the predicted natural frequencies were within 2 percent of the measured values (for the first ten modes of vibration), with one (the blade edgewise

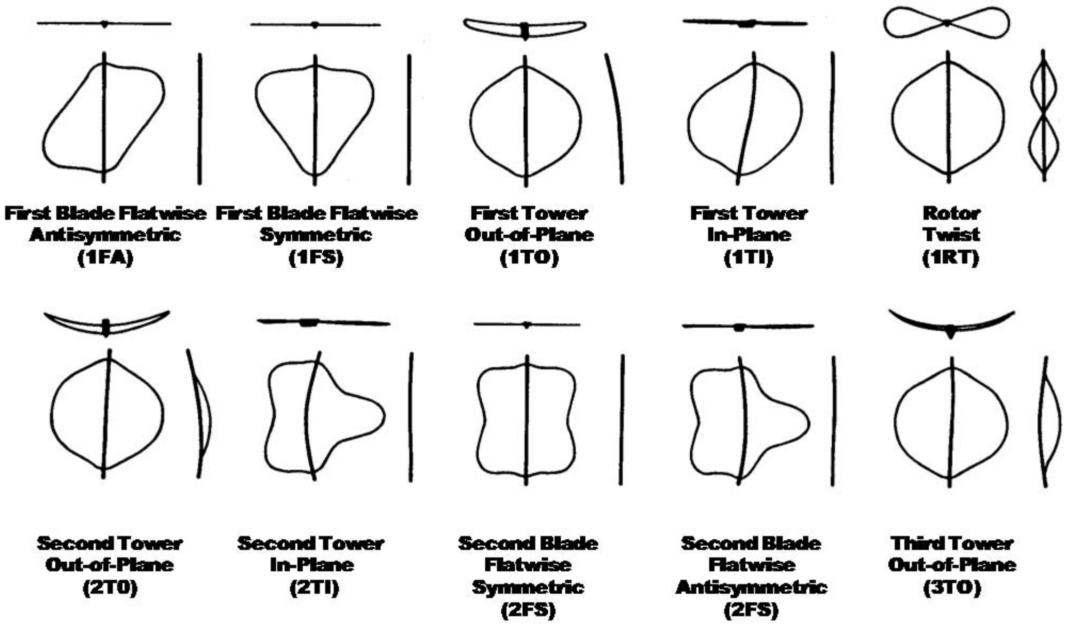

Figure 6. Mode Shapes for the Test Bed

mode) at 5 percent.[17] The predicted and measured guy-cable natural frequencies also proved to be very important in the operation of the turbine.

The Test Bed had two rotor structural resonances within or near the operation range of the turbine (28 to 40 rpm), as seen in Figure 7. The lowest resonance was at 32 rpm, the crossing of the 3P driving frequency by the first blade edgewise mode (1BE, i.e., the "butterfly" mode), illustrated in Figure 7. This resonance was in the middle of the operating range of the Test Bed and could not be shifted. This resonance was not very broad and not very powerful, so operating the turbine at this rpm caused minimal damage. The second resonance was located just above 40 rpm, the crossing of the 3P driving frequency by the "First Tower In-Plane" mode (1TI), see Figure 6 and Figure 7. This resonance was rather broad, starting to couple with the turbine rotor at 39 rpm, and very powerful, capable of rapidly destroying the turbine. This limited the practical maximum rpm of the turbine to 38.

The turbine also had two guy-cable resonances. The first occurred at 25 rpm; this was not significant during operation but had to be avoided during startup. The second guy-cable resonance was located in the operational range of the turbine, near 36 rpm. During initial operation of the Test Bed, this resonance was avoided by the controller, as explained below in the discussion of controller operation. Simply put, the controller was programmed to avoid dwelling on this resonance by passing through this rpm quickly during operations. This resonance was later moved outside of the operation range of the turbine through an adjustment of the guy cables. Thus during initial operation, the control system for the Test Bed had to avoid both of these guy-cable resonances during operation; later in the testing sequence, it only had to avoid the 25 rpm resonance.

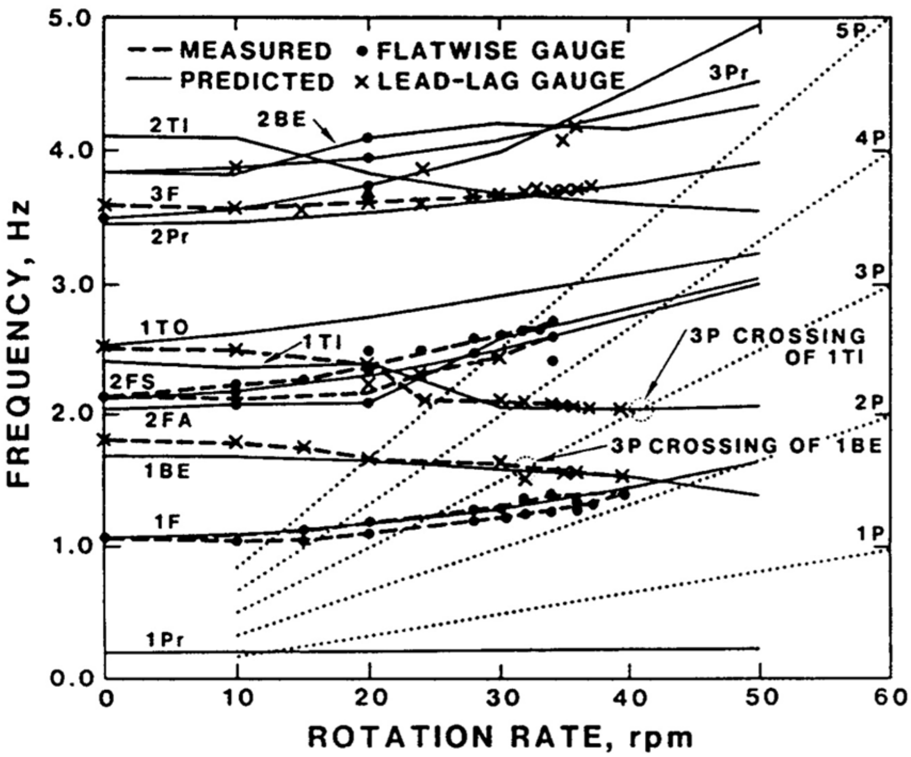

Figure 7. Rotational Campbell Diagram for the Test Bed

Structural Response

As reported by Ashwill,[11,12] comparisons between measured and predicted gravitational, centrifugal, and cyclic stress distributions compared very well with each other over the entire operational range of the turbine. A comparison of the measured and predicted flatwise gravity stress distribution along the blades is shown in Figure 8. Stresses along the blade from the top to the bottom are plotted left to right on the x-axis, and positive stress corresponds to tension on the outside of the blade. The location of the different blade sections that make up the blade are noted along the x-axis. The patterns of stress distribution for the measured and predicted are very similar, and the values agree well at the ends of the blades. Discontinuities in the stress distribution occur at the joints because of the abrupt change of blade stiffness at those locations. The analytical values show an offset along the middle blade portion,

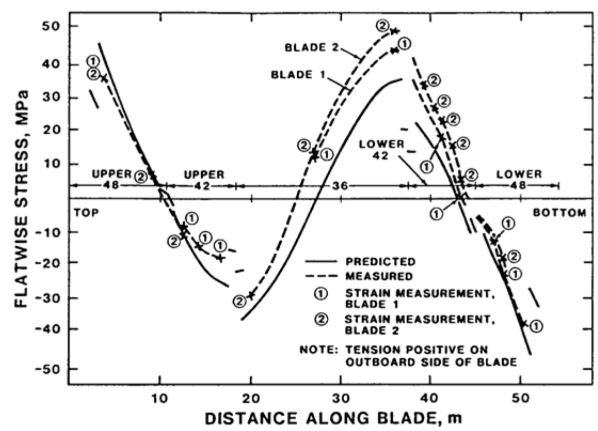

Figure 8. Flatwise Gravity Stress Distribution in the Test Bed Blades.

however, indicating an under prediction on the tension side and an over prediction on the compression side of up to 10 MPa (1440 psi). The middle blade portion is also the only location where significant differences between the blades occur. This discrepancy may be due to small errors in the blade shape geometry in the model.

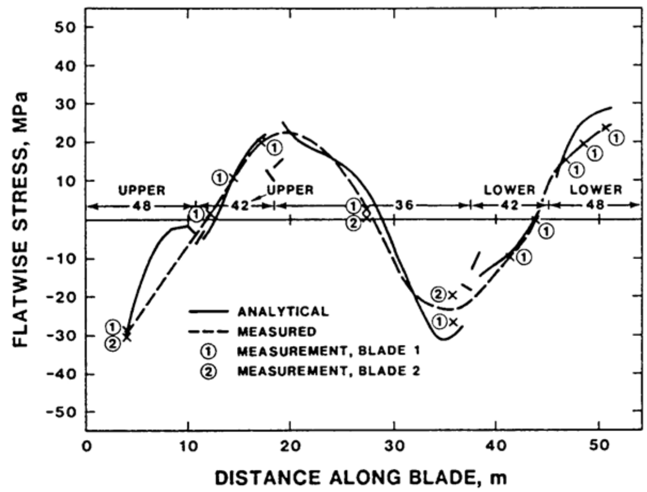

Figure 9. Flatwise Centrifugal Stress Distribution in the Test Bed Blades at 28 rpm.

A similar plot for the centrifugal flatwise stress distribution at 28 rpm is shown in Figure 9. The data presented in this plot is averaged over 40 second period to obtain a direct comparison with the predicted stress distribution. There is excellent agreement between the measured and predicted values. Comparisons were also made at 10, 15, 20, 32, and 36 rpm, and similar agreement was observed at each rpm.

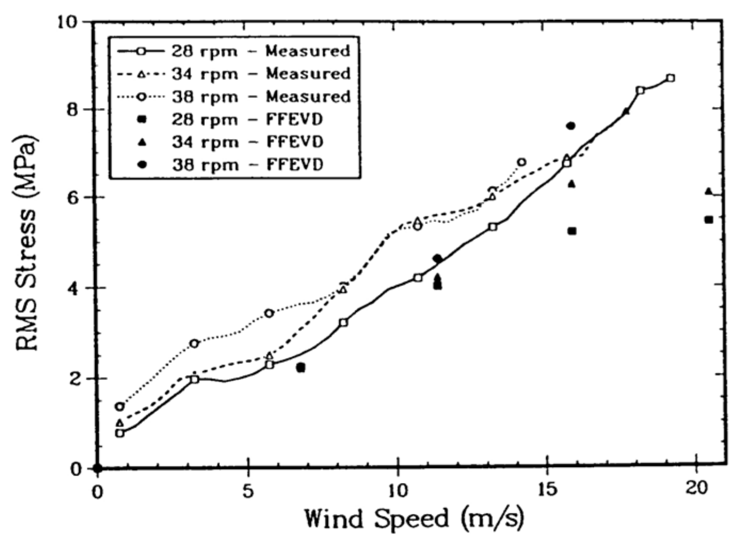

Figure 10. Lower-Root, Flatwise Stresses in the Test Bed.

The "method-of-bins"[4] was used to investigate the variation with wind speed of the blade cyclic stresses resulting from steady wind loading. These data were characterized by the standard deviation of their stress/time histories. Inspection of these data reveals that the blade roots had the highest levels of cyclic stresses, and that the stresses in the upper and lower roots were approximately equal. Figure 10 shows the variation of the lower root cyclic stresses with wind speed and rpm. The measured data in this figure is compared to predictions for steady wind inflow conditions (results from the FFEVD code discussed below). The predictions are reasonably close to measured values in low winds, but diverge from the measured data at high winds. Turbulent inflow predictions bring the higher wind speed predictions into line with the measured data.[18]

Fatigue Response

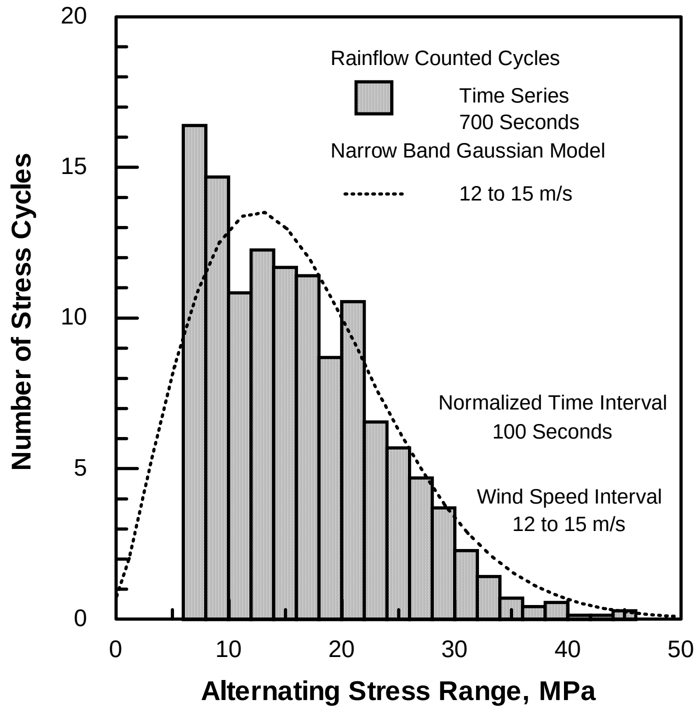

Figure 11. Semi-Log Histogram of the Rainflow-Counted Stress for the Test Bed.

Even the most successful of the early turbine designers had overlooked a significant component of the loads. The designers assumed the loads could be approximated using a spatially uniform incoming wind, with changes in the average occurring very slowly. Turbulence was originally thought to produce dynamic loads at sufficiently low frequencies to be neglected. This viewpoint overlooked the fact that the blades produce relatively large lift forces as they move through the wind, magnifying the turbulence-induced loads and shifting them to higher frequencies. The broad-band nature of turbulence also excites the natural frequencies of the turbine more than had been expected, leading to the premature failure of many of early designs. For the design 30-year life, turbine blades typically must withstand at least 10^9 cycles,[19,20] which is at least two orders of magnitude larger than the typical design life of a commercial transport airplane. The Test Bed provided one of the first detailed, measured fatigue load spectra for an operating wind turbine.[21] Figure 11 is a typical histogram for the Test Bed.

Most of the materials used in the construction of wind turbines are typical of those materials that are used in rotating machinery and towers. Thus, the turbine system is primarily composed of materials that are relatively common structural materials with extensive engineering applications and databases. However, the blades used on wind turbines are unique structural components. In most cases, even for common structural materials, a fatigue database extending to 10^9 cycles did not, and still does not, exist. For the Test Bed, the fatigue properties of the 6063 T5 aluminum used for the blades, such as those presented in Figure 12,[22] had to be determined as part of the project. The extruded aluminum proved not to be as good in fatigue as the fiberglass composite materials that were being used in equivalent composite blades.[††]

[††] An anecdote: When the Test Bed was taken down (toppled from its standing position), the blades shattered when they hit the ground. It would appear that the extrusion process aligned the grains in the aluminum, yielding an oriented, brittle material. Thus, this material was particularity susceptible to the formation and propagation of cracks near stress concentrations, as was the case in the fatigue failures that plagued the FloWind fleet.

THE TOWER

The Test Bed was designed as a research turbine, and as such, many of the components were "over-designed" to permit a wide range of test configurations. This is particularly apparent in the design of the center torque tube (or, tower). The tower was designed to be 3.0 m (9.8 ft) in diameter to facilitate testing of a single-blade configuration without a counterweight. This proposed test configuration (never tested) dictated not only the size of the tower, but also the size of its top and bottom bearings.

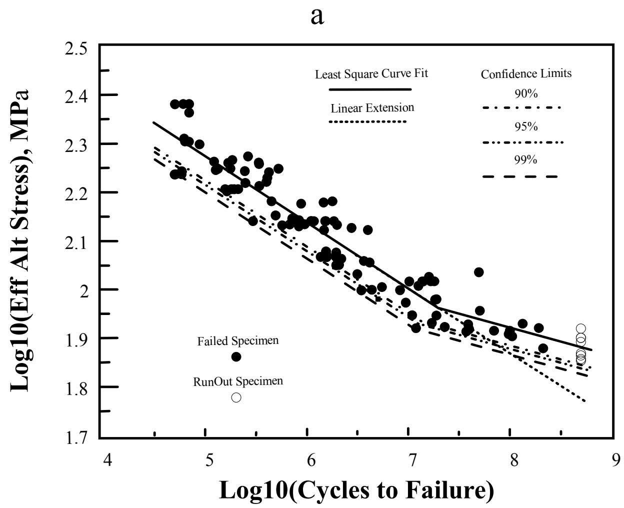

Figure 12. Normalized S-N Diagram for 6063-T5 Aluminum.

An external spiral staircase on the tower served the dual purposes of providing ready access to the top of the tower and ensuring that no coherent vortices would be shed off the large-diameter tower to cause major fatigue loading of the blades as they passed downwind of the tower.

BRAKES

The braking system for the 34-m was over-designed to allow for a variety of innovative retrofits. The system consisted of 4 calipers spaced around a large high-strength, steel disc located at the base of the tower on the low speed shaft. The calipers were spring applied and hydraulically released to be passively safe and allow for stopping if hydraulic pressure was lost in an abnormal condition. Normal stopping occurred with the application of a set of two opposing calipers after the turbine was slowed down by regenerative braking. The sets of two were alternated for normal stopping to allow for consistent brake pad wear. Each set of two calipers could stop the turbine in an emergency stop condition; however all four were applied for an emergency stop.

POWER TRAIN

MOTOR/GENERATOR

The Test Bed was equipped with a variable-speed/constant frequency motor/generator system with a programmable controller. The generator was sized to handle the turbine at its nominal power rating of 500 kW electric through its peak power rating of 625 kW electric. The generator's torque, and thus speed, was controlled by a load-commutated-inverter (LCI) variable-speed motor drive. The LCI connected the turbine to the utility system by converting the variable voltage and frequency of the generator to the constant voltage and frequency of the utility system.

TORQUE RIPPLE

The impact on the gearbox and generator of the torque ripple[‡‡] created by the 2-bladed rotor was an item of concern in the Test Bed design. To handle this problem, a pair of torque compliant couplings were inserted between the low-speed rotor shaft and the low-speed gearbox input shaft. Each coupling consisting of two facing circular disks (steel) connected by several rings of elasomeric cylinders, The compliance of each coupling could be adjusted by changing the number and/or the radial position of the elasomeric cylinders. The standard deviation of the aerodynamic torque at the base of the rotor approached 100 percent while the measured torque ripple on the generator side of the first compliant coupling was reduced to approximately 25 percent (at 28 rpm in winds of 10 m/s). The second coupling attenuated the torque ripple even further.[12] Thus, the low-cost coupling reduced the torque ripple very effectively, and the gearbox and the generator operated very effectively and reliably.

CONTROLLER

The control algorithms relied upon wind speed measurements from two anemometers that were located at 10 m and 30 m above the ground and approximately 5 rotor diameters from the turbine (to minimize shadowing effects). The control algorithms used the highest measured wind speed from the anemometers to compute a moving 200-second wind speed average that was updated every 20 seconds.

Turbine control algorithms were implemented on a programmable logic controller (PLC). The PLC allowed testing of the Test Bed using varied control algorithms which included operator-selected, single fixed-speed, dual-speed and full-variable-speed operation, full-automatic operation and regenerative braking.

The Test Bed's drive train, including its generator, was designed for a maximum sustained power production of 500 kW. As stall regulation at 38 rpm would take place in the 700 to 800 kW range, well above the design limits, the operation of the turbine in constant-speed mode at 38 rpm was limited to wind speeds of 13 m/s. When operated in variable-speed automatic-operation mode, the turbine was limited to wind speeds below 20 m/s.[23]

All of the algorithms included avoidance procedures for the three resonances within the operational range of the turbine (guy-cable resonances at 25 and 36 rpm and the 3P/1TI resonance at 40 rpm, see the discussion above). The 40 rpm resonance was handled by restricting the turbine operation to below 38 rpm. The 25 rpm resonance only had to be avoided during startups and shutdowns. It was handled by accelerating and decelerating the turbine through the resonance at a minimum rate of 0.1 rpm/sec, using starting torque or regenerative braking, respectively, to achieve that rate. During the initial testing phase of the turbine, the 36 rpm guy-cable resonance was still active and was very important during both variable-speed operation and fixed-speed operation at 38 rpm. For the latter,

‡‡ Torque Ripple is defined to be the maximum torque minus the mean torque divided by the mean torque.

the resonance only had to be avoided during startup and shutdown and was handled the same as the 25 rpm resonance, i.e., by accelerating and decelerating the turbine quickly through the resonance.

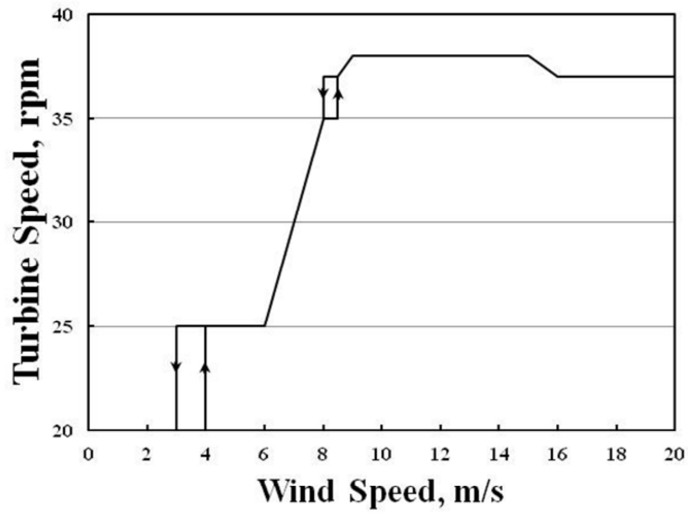

Figure 13. Variable Speed Control Algorithm

For variable-speed operation, this avoidance was significantly more difficult to handle. The turbine controller was programmed to follow the peak power curve while restricting operation within 1 rpm of the 36 rpm resonance, as illustrated in Figure 13. When the turbine was speeding up in response to increasing winds and reached a speed of 35 rpm (at 8.5 m/s wind speed), it was held there until the wind speed had increased to 9 m/s, sufficient to allow the turbine speed to be rapidly increased (a minimum rate of 0.1 rpm/sec), to 37 rpm. For decreasing winds, the process was reversed, holding the turbine at 37 rpm until it could be quickly decreased to 35 rpm. The controller required that the average wind speed remain in the proper range for a full 2 minutes before a jump was initiated. After this 36 rpm guy-cable resonance was moved outside the operational range of the turbine by adjusting the guy-cable tension, the exclusion zone was removed from this variable speed algorithm.

Various variable speed control algorithms could be used to control the output of the Test Bed.[24] The maximum power algorithm, i.e., the algorithm that follows the highest efficiency of the rotor, is the one shown in Figure 13. This algorithm started the turbine when the wind speed increased to 4 m/s and stopped it when the wind speed decreased to 3 m/s or increased above 20 m/s. For a given 200-second average wind speed, the desired turbine speed was chosen from a tabular representation of Figure 13. The turbine's rotational speed was then incremented at a rate of 0.1 rpm/s towards the new desired set point. This process was repeated every 20 seconds, even if the desired set point had not yet been reached.

POWER PRODUCTION

The performance data cited here for the Test Bed was assessed using the power production of the rotor. The rotor power was obtained using the measured low-speed shaft power (determined from the low-speed shaft torque and the rotation rate of the turbine) and the tare losses in the tower bearings. This tare losses were measured to be 3 kNm.[12] The rotor power was also adjusted to sea-level air density.[§§] A large portion of the data presented here in graphic form is available in tabular form in Ref. 12.

The electrical output of the turbine was also measured. The loss in converting the mechanical power at the low-speed shaft to electrical power was estimated to be 10 percent.

INITIAL PERFORMANCE

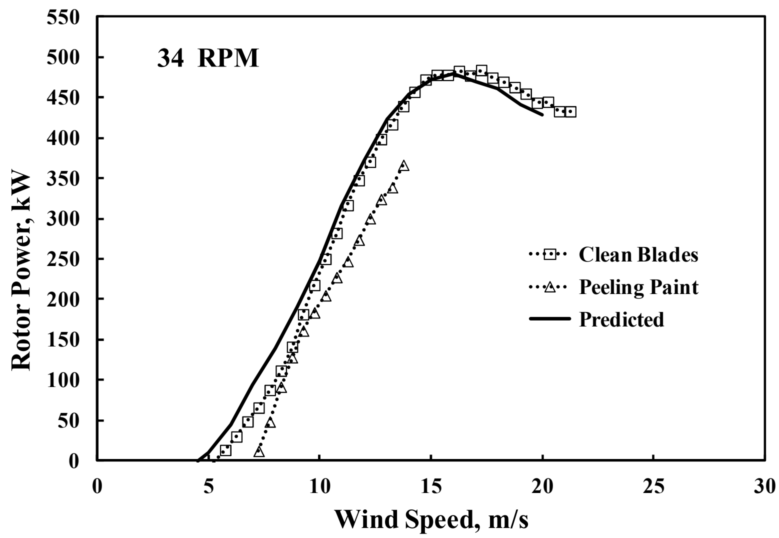

Figure 14. Power Performance Comparisons for the Test Bed, Impact of Leading Edge Paint Chips.

Initial performance data revealed that the turbine was performing well below predictions,[25] as shown in Figure 14. A thorough check of the wind turbine revealed flaking paint on the leading edge of the NLF airfoils; an example is shown in Figure 15a. The flaking had created forward facing steps near the leading edge with a height of approximately 0.25 mm (0.010 in), a very effective boundary layer trip which could be expected to destroy the laminar flow over the blade. This loss of laminar flow would cause the blade to produce higher drag and lower lift than for which it was designed, resulting in a significant degradation of turbine performance. The leading edges were smoothed by sanding away the flaking paint, as shown in Figure 15b, and the performance characteristics of the turbine improved significantly, as seen in Figure 14.

a) Paint Chips **b) Sanded Leading Edge**

Figure 15. Leading Edge Condition of the Test Bed

[§§] The Bushland site is 1183 m (3880 ft) above sea level.

PERFORMANCE

Test Bed performance data (acquired after the blade leading edges were smoothed)[25,26] are compared to predictions (made before the turbine was ever built) in Figure 16. From the figure it is clear that stall regulation is in fact occurring and that the measured data agree fairly well with the predictions. The most important discrepancy occurs at wind speeds from cut-in to about 9 m/s (20 MPH), where measured results consistently fall below analytical values. One possible contributor to these differences may be a unique feature of the Test Bed – the aerodynamically "dirty" blade-to-blade joints that are shown in Figure 5 and are discussed below.

In terms of efficiencies, the data in can be cast in terms of the turbine power coefficient,[***] as shown in Figure 17. As seen from this figure, rotor power efficiencies, C_p, reach approximately 41 percent (28 rpm at 7.75 m/s).

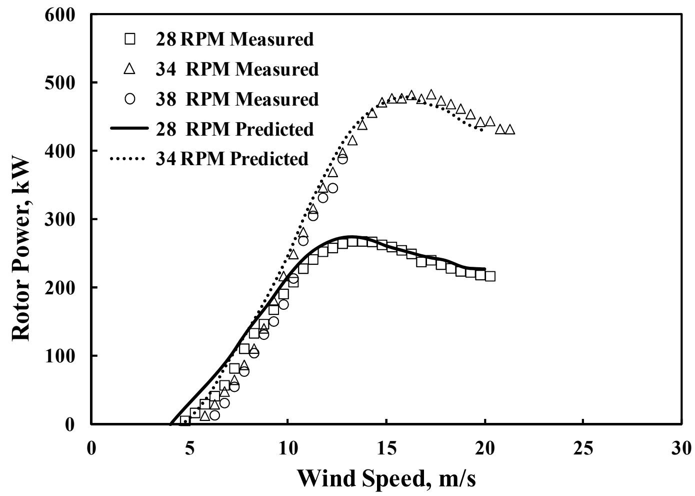

Figure 16. Rotor Power Performance Comparisons for the Test Bed, Measured vs Predicted.

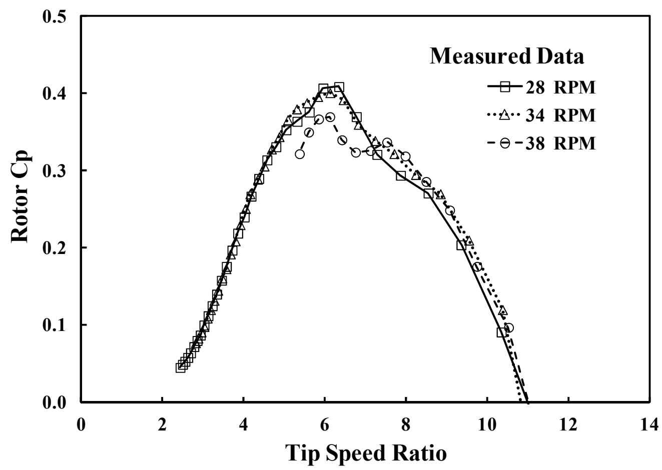

Figure 17. Measured Rotor Power Coefficient for the Test Bed, Measured vs Predicted.

[***] This coefficient is defined as the electrical power produced by the turbine divided by the power available in the wind that passes through the turbine. This term is commonly called the Power Cp. A similar term is the rotor Cp, or simply the Cp.

JOINTS

Figure 18. Fairing over Blade-to-Blade Joint.

While the pre-built predictions of power output agreed well with measurements for high wind speeds, the agreement for low wind speeds were not as good. As discussed above, one possible contributing factor to this discrepancy between measured and predicted output power was identified as the external blade-to-blade joint structures shown in Figure 5. These joints were not aerodynamically smooth: they had exposed bolt heads, blunt leading surfaces and sharp corners. These joints were aerodynamically smoothed with small, hand-formed aerodynamic fairings, similar to the one shown in Figure 18, resulting in a small increase in performance in both low and high winds, as seen in Figure 19. The efficiency of the rotor power production, C_p, increased from approximately 41 percent to 43 percent at a wind speed of 8.25 m/s (also at 28 rpm) due to the addition of the fairings.

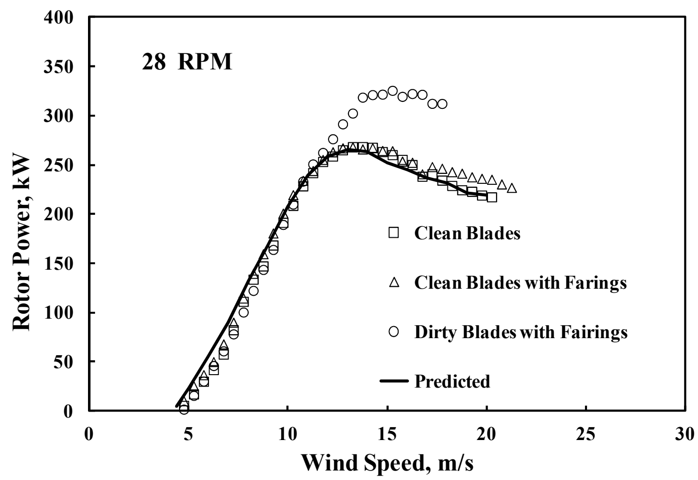

Figure 19. Power Production for Clean, Dirty and Faired Blades for the Test Bed.

For commercial turbines, an internal joint that results in smooth external transitions from one blade section to another would be used, and an increase in turbine efficiency would be expected.

DIRTY BLADES

As the performance testing of the Test Bed continued, the power production at the higher wind speeds actually increased, as illustrated by the "Dirty Blades" data in Figure 19. An inspection of the blades showed that the leading edges of the blades had become "dirty" from the remains of insects that had hit the blades during normal operation. Apparently the bug debris effectively entrained higher velocity flow into the near wall region of the boundary layer, delaying blade stall and boosting performance. Once the turbine was run during a heavy rain storm, the debris was effectively removed from the blades, and the original power curve returned.

VORTEX GENERATOR TESTING

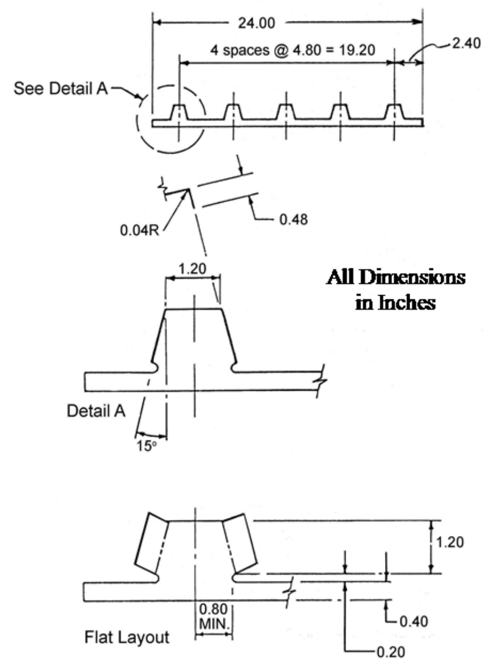

Figure 20. The Strip Vortex Generators.

Based on measured performance improvements after adding vortex generators to the blades of our earlier 17-m VAWT in the near vicinity of the tower blade/tower joints,[27] we equipped the Test Bed with vortex generators placed on the 48-in chord airfoil sections and the outboard portions of the 42-in chord sections. The vortex generator configuration used for both turbines was a counter-rotating one, based on a geometry defined by Pearcy[28] as "most likely to succeed" after his extensive state-of-the-art review of technical literature. This configuration is illustrated in Figure 20, where all dimensions pertaining to the vane geometry are based on a chord length, c, of 48 in. In non-dimensional units, vane height is 0.01c, vane length is 0.025c, vane pair trailing edge separation distance is 0.025c and vane pair separation distance is 0.1c. The leading edge of the vanes were placed at 0.1c aft of the airfoil leading edge on both the 48-in and 42-in sections.

The results were quite disappointing, as we were not able to detect any significant difference in turbine performance due to the presence of the VGs.

IMPORTANT RESULTS

Several aspects of the Test Bed program were noteworthy.

1. A full system approach to the design of the turbine was necessary to find the right trade-offs between aerodynamic performance, structural performance and manufacturability.

2. The turbine was extensively instrumented and the necessary data analysis techniques that were required to process and interpret that data were developed.

3. The resultant comprehensive and extensive database for this turbine provides data for validating future Darrieus VAWT analytical developments.

Arguably, the single most important result of Sandia's Test Bed Program was the development and validation of a set of analysis techniques (with their accompanying design codes) that permitted the determination of the aerodynamic and structural performance of a VAWT. These techniques and codes were the stepping stones for the commercial turbines discussed below. Even though the development of these techniques and codes for VAWT applications was essentially terminated in the mid-1990's, they have been extensively documented (see below), and the algorithms upon which they based are available in the open literature.

The Test Bed also yielded several hardware developments. Most notable are VAWT specific airfoils (Sandia's NLF family of airfoils), the first airfoil family developed specifically for wind-turbine applications. Their use on the Test Bed illustrated their ability for good power regulation through controlled stall characteristics.

The Test Bed was one of the first wind turbines to use true variable speed operation to maximize its aerodynamic and mechanical performance. One variable-speed algorithm programmed into the controller followed the maximum efficiency curve at lower wind speeds, avoided structural resonances, and reduced stopping loads and increased the reliability of the braking system through the use of regenerative braking (using the generator to initially slow down the rotor during shut down). With appropriately chosen parameters, the rotor could be decelerated on a schedule that permitted safe operation while minimizing loads. Mechanical brakes were still needed to fully stop the rotor and to ensure that it did not self-start.

The Test Bed program also led to the development of innovative measurement techniques. The Natural Excitation Technique (NExT) for modal testing, developed under Sandia's VAWT program, utilized the Test Bed for validating both the testing technique and the finite element analysis for predicting the modal response of the turbine.[29] The development of NExT pioneered the concept of using naturally occurring excitation (wind, seismic, vehicle traffic, etc.) as the force input for a modal test. Even though the inputs cannot be measured, as done in traditional modal testing, the modal parameters can still be extracted from the test data. This technique has since been extensively exploited for determining the modal response of many large structures, including HAWTs, bridges, walkways, buildings, and stadiums. It has become the standard in the civil engineering community and is the theme of a biannual conference, the International Operational Modal Analysis Conference (IOMAC). At the second IOMAC, Thomas Carne and George James presented the conference's keynote address, a description of the inception of operational modal analysis (NExT) in the development of testing for wind turbines.[30]

The fatigue behavior and performance of the Test Bed were analyzed extensively. This analysis led to the realization that the typical aircraft fatigue characterizations of the materials used in wind turbine blades were not adequate. Wind turbine design requires fatigue data that extend to much higher cycle counts, at least two orders of magnitude beyond that gathered for typical aircraft design fatigue databases. The development of such a fatigue database for 6063 aluminum was completed before the Test Bed program ended. The experience gained in the development of this database has been the foundation for the extensive turbine fatigue database for composite materials developed primarily for HAWT blade materials under the auspices of Sandia, i.e., the SNL/MSU/DOE Fatigue Database.[31]

The design and operation of the Test Bed also revealed various problem areas that must be addressed in future designs. In particular, roughness (bugs on the leading edge) increased the power production of the Test Bed; at low rpm this increase in the output power was of the order of 15%. While this increase is good for power production, it illustrates an uncertainty in predictive tools. This increase is in sharp contrast to the 'normal' loss in maximum power observed in conventional NACA airfoils; the FloWind 19-m turbine peak power went from 275 kW with clean blades to 220 kW with dirty blades.[32] The aerodynamic codes of that period could not predict the increase (or decrease) in power production. This change of maximum output power from the design level is detrimental to an optimized design and must be addressed in the design of the turbine. The behavior of dirty blades is an important research topic that requires additional attention.

The Test Bed also demonstrated that blade-to-blade joints that were not aerodynamically smooth had a negative influence on the turbine's performance and that smoothing them with crude fairings improved the performance. There is no doubt that the performance could have been further improved with more optimized fairings (also true of subsequent efforts by FloWind, discussed below, to fair their turbine strut-to-blade joints). The entire subject of fairings on turbine blade-to-blade and blade-to-strut joints needs detailed attention, both analytically and experimentally.

THE SANDIA "POINT DESIGN"

As a result of the technical successes of the Test Bed program, in late 1989, a government/private industry partnership was initiated to design and build a commercially viable VAWT. To facilitate this transfer of technology from the public sector to the private sector, Sandia engineers designed a "pre-commercial" version of the Test Bed, called the "Point Design."[33]

The Point Design was introduced to the private sector (15 US companies) in a Commercialization Workshop held in mid 1990 in Amarillo, Texas. This design elicited considerable interest from FloWind Corp. and became the starting point of their EHD rotor.

THE POINT DESIGN

In developing the Point Design, Sandia engineers used the best features of the Test Bed, combined with the validated design codes developed under that project. Research aspects of the Test Bed were dropped from the design and all components of the turbine were scrutinized to reduce weight and cost. Off-the-shelf components were used wherever possible. Although the 34-m diameter of the Test Bed was not optimum, the diameter and the height-to-diameter ratio were held constant to provide a direct comparison with the Test Bed and to ensure that the experience with the Test Bed was incorporated into this proposed commercial turbine. A sketch of the Point Design may be found in Figure 21.

The major changes from the Test Bed included: (1) a significant reduction in the tower diameter (2.1 m rather than 3.0 m); (2) elimination of the 1.07 m chord section of the blade; (3) a single-speed generator (36 rpm); (4) significantly lighter bearings, bearing housings, shafts and foundation; and (5) Off-the-shelf brakes.

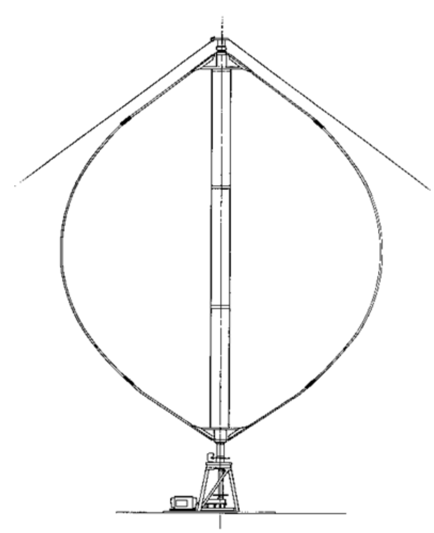

Figure 21. The Sandia Point Design, a Commercialized 34-m VAWT

IMPORTANT RESULTS

The Point Design produced a turbine that operated at 36 rpm and had a maximum output of 585 kW at 17 m/s inflow speed. Operational stresses were predicted to be very similar to those measured in the Test Bed. Initial cost estimates were provided ($800,000 in 1990 dollars) and the team estimated that production versions of the turbine could be built at half this cost (costs would decrease due to volume production and improvement in worker skills).

The estimated portioned costs of the various systems and subsystems are shown in Figure 22 and Figure 23. These estimates were obtained using the ECON90 computer code, see the discussion below in the "Codes" section of this manuscript. Elemental costs contained within the code were updated to current levels (i.e., 1990 costs) for these estimates.

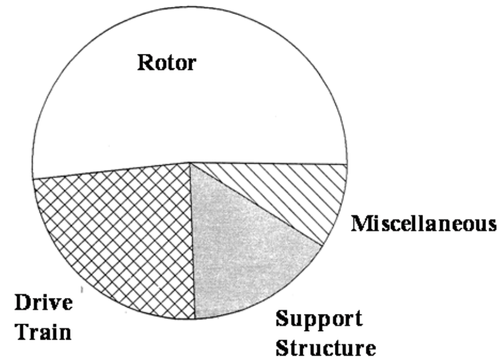

Figure 22. Point Design System Costs (one unit).

As seen in Figure 22 and as one would expect, the rotor was the turbine's highest cost system, accounting for more than half the cost of the turbine. However, despite perceived costs, the column (torque tube) was more expensive than the blades, see Figure 23.

The importance of the Point Design is that it demonstrated that a 500 kW commercial VAWT, based on the 34-m research turbine, could be designed and built and that such a design could be economically competitive with contemporary HAWT designs of comparable power rating.

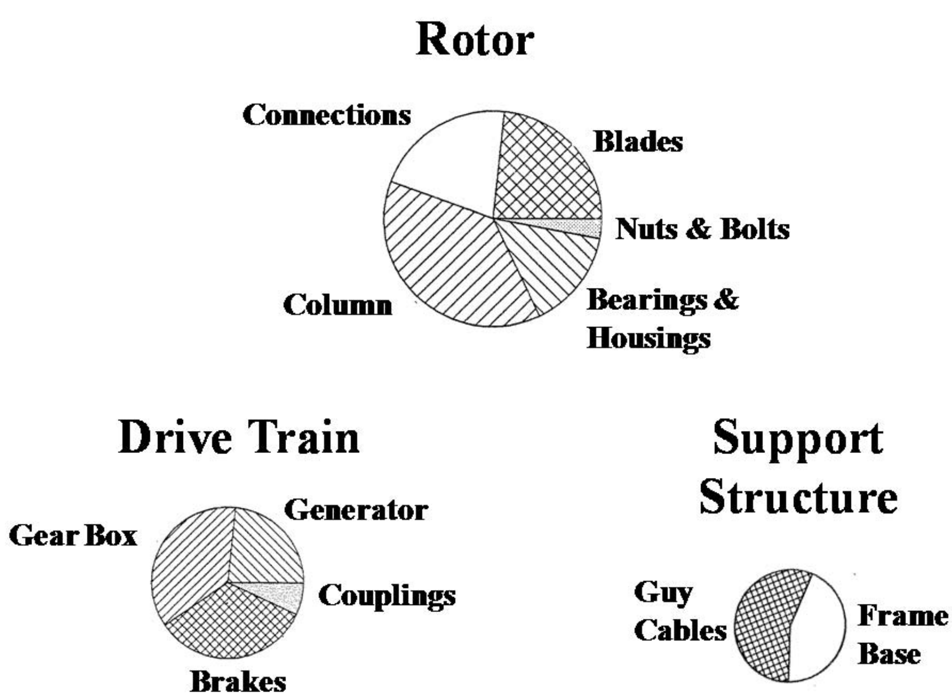

Figure 23. Point Design Subsystem Costs (one unit).

THE FLOWIND COMMERCIALIZATION PROJECTS

After the Commercialization Workshop, FloWind Corp., using various government/ industry partnership vehicles, took the Point Design and began the development of a new commercial VAWT product line.[34] After initial studies, FloWind decided that the most viable path to a new product line was through the development of a retrofit rotor for about 170 of their existing 19-m VAWT fleet. This decision was based on their desire to eliminate high maintenance cost machines (due to fatigue problems with aluminum blades) without replacing the bases, gearboxes and generators, while improving energy capture and increasing the output of the fleet.

The initial work on this effort was performed with funding awarded under a DOE "Government/Industry Wind Technology Application Project" contract, placed in July 1992 and managed by the SNL Wind Energy Program. FloWind contributed approximately 90% cost share under this contract. Sandia technical advisors worked very closely with FloWind on this effort, and FloWind acquired the capability to perform the required analyses by hiring contractors and upgrading staff capabilities. The follow-on work, starting in 1995 and leading to the final retrofit prototype turbine (the Extended Height-to-Diameter or EHD turbine), was completed with funding supplied under the NREL Value Engineered Turbine Development Project. This contract also required considerable cost sharing from FloWind. Personnel from SNL and NREL formed the Technical Review Team monitoring this contract, furnishing direction and advice to FloWind during the contract duration. FloWind performed most of the analyses required during this phase of the effort.

The initial concept for the retrofit rotor, which was started in 1992, used the existing FloWind fleet of 19-m turbines and the Sandia 34-m Test Bed as starting points. The resultant EHD rotor was a 3-bladed design with pultruded fiberglass blades that utilized the existing 19-m bases and much of the 19-m drivetrain. The blades were pultruded as single straight sections 48.2 m (158 ft) long with a constant 0.69 m (27 in) chord and the Somers S824 profile.[35] The blades were bent-in-place when mounted on the tower, yielding a rotor with an H/D of 2.7. This approach resulted in a significant increase in blade flatwise bending loads, compared to those that would have been present if a troposkien shape had been used. The long, modest chord composite blades required the incorporation of "deep" (far away from the tower-to-blade junction) struts to minimize vibration and control stresses. The use of three blades significantly reduced the size (cost and weight) of the center torque tube while increasing many aspects of the stability and enhancing the modal response of the rotor.

This final EHD design was a 2-speed (34.5 and 52 rpm), 18-m diameter stall-regulated turbine that produced rated power of 300 kW at 18 m/s (the original 19-m turbine that produced a maximum of 258 kW). This design is illustrated in Figure 24. FloWind built and tested two prototype EHD turbines, known as Adam and Eve.

Although this project concentrated on the design of a retrofit rotor, it was always intended to be the basis for a commercial line of new VAWTs.

THE ROTOR

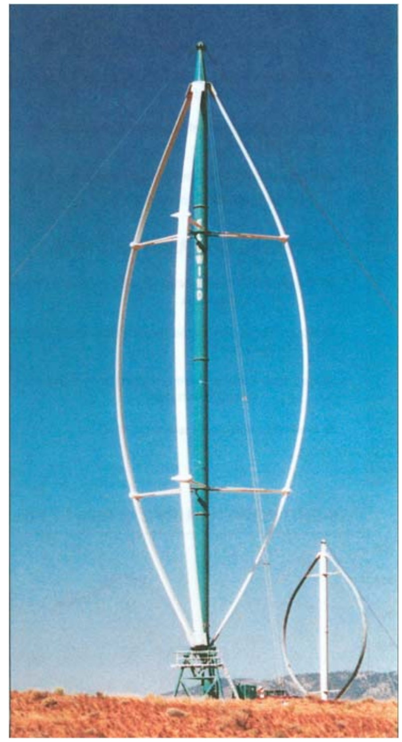

Figure 24. The FloWind EHD VAWT

The initial design envelope for the FloWind EHD turbine was opened up to include rotors that were much larger, and smaller, than the Point Design. As the optimization process was evolving, several points came to the forefront: (1) aluminum extrusions were not acceptable because they were too costly to put together and fatigue properties of the extruded aluminum were not as good as fiberglass composite materials; (2) composite materials could be formed relatively inexpensively using pultrusion manufacturing techniques; and, (3) the tower and its bearings were a major contributor to the capital cost of turbine (see the cost estimates for the Point Design in Figure 23). The final outcome utilitzed composite blades in a 3-bladed rotor. The addition of the third blade stabilized the rotor structurally by significantly attenuating the blade's in-plane and out-of-plane vibrational modes. Thus, the sizes (costs) of the tower and bearings and guy wires were significantly reduced at the expense of an additional blade. Moreover, financing would be easier to obtain because composite blade materials had become the norm in HAWTs by this point in time and investors were more comfortable with a fiberglass blade material than with the aluminum that had been used on the earlier, problem-plagued VAWTs.

Although costs and perceptions were overriding design drivers, the 3-bladed design was also a result of the pultrusion technology that was used to produce the blades. This process limited the attainable blade chord to 0.69 m (27 in). This restriction meant that three blades were required to attain the desired rotor solidity.

The higher H/D of the EHD had several desirable features. First, it enabled the composite blades to be bent into place without subjecting them to excessive flatwise stresses. And, second, it enabled the retrofit design to reach an increased peak power of 300kW, without increasing the rotor diameter or changing the basic rpm.

During assembly of the turbine, the tower was erected first, and each blade, in turn, was lifted, bent and attached to the standing tower.

AERODYNAMIC DESIGN

The first prototype turbine built by FloWind was a 300 kW machine that was based on the Test Bed technology. This turbine used an SNLA 2150 airfoil and a retrofit gearbox that increased the turbine's rotation rate to 60 rpm. Quality problems with the retrofit gearbox led to an early failure and subsequent prototypes were designed to operate with the original F-19 gearboxes at 52 rpm. At that rpm, the properties of the SNLA 2150 airfoil would not produce sufficient power, so another VAWT-specific airfoil, the Somers S824[35], was designed by Airfoils, Inc.[36] specifically to meet FloWind requirements. The initial blades built with this airfoil (the so-called "B" blades) buckled because the blade walls were too thin to withstand the bend-in-place and operating stresses. The blade was redesigned with thicker walls (the "C" blades) and blades of this design were used in the final versions of the two prototype turbines.[†††]

Struts were required to stabilize the rotor. Initial designs used a single-strut arrangement with the struts placed close to the roots (one strut near the top root and another near the bottom root of each blade for six struts total). These struts had a radial extent of approximately 69 percent of the radius of the turbine. This design proved inadequate and had to be replaced with a deep single-strut design that extented to approximately 78 percent of the radius (again, six total).

The struts were constructed in a "V" shape (the point of the "V" attached to the blade) that attached to a flange bolted to the tower. The initial set of struts was constructed from schedule 80, 89 mm (3.5 in) diameter pipe, and the final set was constructed from schedule 80, 102 mm (4 in) diameter pipe. The strut joints were not initially aerodynamically faired. In that configuration, they produced significant aerodynamic drag that shifted the rotor power curve significantly to the right in higher wind speeds. Maximum output remained the same, at about 300 kW, but the power curve was shifted to the right in higher wind speeds by 1.3 to 1.8 m/s (3 or 4 mph), or a reduction of approximately 25 percent in power production at medium wind speeds. When aerodynamically faired, the performance increased by approximately 15% relative to the unfaired configuration.

MECHANICAL DESIGN

As mentioned above, the mechanical design of the rotor used a single-chorded blade section with a Somers S824 profile. There were no joints in the blades, and struts were used for stabilization. To increase the output power of the rotor (from 258 kW for the original F-19 turbine to 300 kW), the EHD design increased the swept area of the rotor by stretching of the rotor vertically, rather than by stretching its diameter (the approach that had been typically used in the past), hence its name. The EHD design minimized the bend-in-place static stresses compared to a lower height-to-diameter rotor, but its non-troposkien shape increased the operational bending stresses relative to the earlier designs.

[†††] The bending stresses in the blade were higher than in a conventional design because the blade's bend-in-place shape did not follow its preferred (no bending stresses) troposkien shape.

Blade and strut attachment joints used a system of internal fiberglass inserts and external fiberglass clamshells to reinforce the blade section. To fabricate a joint, the blade sections were prepared by cutting them to length and removing the pultrusion residue from the mating blade surfaces. Then, several appropriately-shaped pultruded inserts (one in each internal cavity of the airfoil section) were bonded inside the blade ends and across the joint. The clamshell (two pieces) was then aligned over the ends of the blade sections and bonded in place. Finally, the reinforcements were through-bolted to safeguard against adhesive creep.

The loads on the turbine with the deep struts (78% of max radius) showed that the predicted and measured stress levels were in good agreement. As noted above, the bend-in-place operating stresses were quite high and required the use of a rather thick blade wall.

Interestingly, the highest loads on the blades occurred during installation when the blades were lifted from the ground (before being bent into shape).

THE TOWER

The use of three blades enabled a reduction in the diameter of the center tower (torque tube). The design was able to use a 1.27 m (4.2 ft) diameter center tower (decreased from 2.1 m in the Point Design) while increasing the height of the tower by over 50 percent from the Point Design.

THE BRAKES

The braking system was one of the highest maintenance items on the FloWind fleet. The rotor retrofit design examined three different replacement brake systems for primary braking. The design that was selected utilized hydraulically-applied floating calipers with an increased throat size that moved the brake pads away from the edge of the brake disk. These off-the-shelf calipers had the best performance of the three systems investigated and were the easiest to install and maintain.

The final braking system design utilized two sets of the hydraulically applied brakes. A third, fail-safe parking brake set was also incorporated in the braking system. This set of brakes was spring applied and hydraulically released. The parking brake was always applied whenever the turbine was not running.[‡‡‡] The three independent brake sets (two service sets and one fail-safe set) were applied in sequence in a normal stop to avoid subjecting the rotor to excessive stresses. During shut-down, the first set was applied while the turbine's generator was still connected to the grid. When the generator speed dropped below synchronous speed, it was disconnected from the grid and the turbine was brought to a halt. The other two sets of brakes were applied sequentially, as required to stop the rotor; i.e., if the first set was not able to stop the rotor within a specified time

[‡‡‡] A VAWT may self-start if it is not continually restrained when it is stopped.

then the second set was applied, and the third set was applied if the first two could not stop the rotor.

IMPORTANT RESULTS

THREE-BLADED ROTOR

Early Sandia studies of 2- and 3-bladed rotors (500kW or less in size) indicated that the capital and installation costs associated with 3-bladed rotors were significantly higher than those of a 2-bladed rotor. In addition, designs with fewer blades and larger chords (equivalent rotor solidity and therefore, of comparable efficiency) were structurally more effective.

The FloWind design illustrated that these considerations are less important in larger machines, where rotor dynamics dominate the design. The reduction in the size (cost) of the tower, the bearings, the guy system, foundation, etc. more than compensated for the cost of the extra blade, and in addition, the third blade provided additional rotor stability.

The 3-bladed design also shifted the torque pulses, associated with each blade crossing the wind, out-of-phase with each other, thus reducing torque ripple in the drive train. While the torque produced by the 3-bladed rotor remained variable, its variability was not sufficient to warrant a compliant coupling between the rotor and the gearbox, as was required in the Test Bed design.

From this vantage point, the use of 3 blades appears to be optimal for the FloWind design; adding more blades appears to add significant capital costs without reducing balance-of-system costs. In future designs, the use of a third blade must still be investigated on a case-by-case basis.

PULTRUSION TECHNOLOGY

The FloWind retrofit rotors were designed with composite blades that used a pultrusion process for their manufacture. This process had been used previously in small turbines for home and ranch applications, but the FloWind blades were the first to reach chord lengths over 690 mm (27 in). FloWind efforts in this area not only proved that pultrusion blades of this size could be made economically, but their experience also revealed that pultrusion machines of this period were only marginally capable of pulling these blades through the die.[§§§] Extending the pultrusion manufacturing technique to larger chords will require significantly larger pultrusion machines than those available in the mid-1990s.

[§§§] FloWind had to resort to an auxiliary pulling device to help pull the resin-soaked fiberglass rovings and fabrics through the pultrusion die.

BEND-IN-PLACE COMPOSITE BLADES

The pultrusion manufacturing technique produced fiberglass blades with straight blade sections. Unlike aluminum, the fiberglass sections could not be permanently bent into the preferred troposkien shape. FloWind chose a bend-in-place scheme to form the straight blades into the arched-blades required for the full-Darrieus rotor. This forming process led to high bending stresses in the blade that significantly reduced fatigue life. In this design, FloWind was able to overcome these large loads with additional structural strength gained with a thicker blade wall for the airfoil. Whether or not the bend-in-place concept is a viable option for other designs remains an open question.

DEEP STRUTS

The bend-in-place rotor and the pultrusion manufacturing process led to a rotor that required deep struts to stabilize it structurally. While the inclusion of struts is a common practice and is an excellent method for stabilizing the rotor structurally, the FloWind experience illustrates that their inclusion has the potential to significantly decrease the aerodynamic performance of the rotor. As noted in the Test Bed discussion above, the entire subject of fairings on turbine blades needs detailed attention if effective designs are to be forthcoming.

CODES

The extensive data set from the Test Bed was used to develop, modify and validate a suite of design codes. These codes were used in both the Point Design and the FloWind designs. The development and maintenance of these codes was terminated in the mid-1990's. A summary of those codes, lumped by general categories, is provided below.

These numerical codes may or may not be archived and they may or may not run on currently available computers. In any case, none of them are currently available through or supported by Sandia. However, the algorithms upon which several of them are based are available in the open literature. In those cases, the numerical formulations of these design algorithms will probably have to be reprogrammed to be used with today's computer operating systems.

In several cases, these codes were sufficiently generic that they also could be applied to design of HAWTs. The development of those codes, and their associated algorithms, has continued.

AERODYNAMICS

ADAM2: ADAM2 (Advanced Dynamic Airfoil Model) is a numerical analysis code for 2-dimensional airfoils in unsteady motion with boundary layer separation. The airfoil and wake surfaces are represented by a finite set of combined source and vortex panels. The source strengths are prescribed to have the same magnitude as the normal relative velocity on the surface due to the freestream and motion of the airfoil. The vortex strengths on the airfoil surface are determined by prohibiting flow through the airfoil surface and enforcing the Kutta condition. Wake shedding is governed by a dynamic free surface condition and the characteristics of the flow near any boundary layer separation points. Wake deformation is predicted by applying a geometric free surface condition. The code predicts time histories of airfoil lift, drag, and pitching moment.

DLCM: DLCM (Discrete Local Circulation Method) is a numerical analysis code that determines the performance characteristics of a vertical axis wind turbine. The output of the code includes detailed distributions of blade loads and wake convection velocities, as well as overall turbine performance. DLCM utilizes an equivalent turbine for which the number of blades goes to infinity as the blade chord goes to zero such that the turbine solidity remains constant. In this way, the highly unsteady flow through a Darrieus turbine is represented by an equivalent steady flow. The corresponding continuous distributions of shed and trailing vorticity in the wake are determined from the gradient of the local bound vorticity distribution on the turbine perimeter. The effects of Reynolds number variations, dynamic stall, and multiple airfoil profiles and chords are all included in the model. The degree of detail and calculation accuracy which can be obtained with this method are comparable to those obtained with a vortex model, and this model requires much shorter computation times.

POWERSURFACE: POWERSURFACE (Aerodynamic 3-dimensional POWER SURFACE), Version 2.2, is a post-processor for the numerical analysis code SLICEIT. POWERSURFACE uses the Computer Associates DISSPLA graphics library. Data files, obtained from at least 2 SLICEIT output data files, are displayed graphically as a 3-dimensional, perspective plot of turbine RPM versus wind speed versus shaft power.

SNLWIND: SNLWIND is a numerical synthesis code that simulates atmospheric turbulence as seen by a horizontal axis wind turbine blade rotating at a constant rate.[37] The operator may choose from a selection of turbulence models. Other input options allow the operator to control the nature of the spatial and temporal correlation structure of the turbulent wind.

SLICEIT: SLICEIT is a numerical analysis technique that analyzes the loads on a vertical axis wind turbine. The model uses a double-multiple streamtube characterization of the turbine that utilizes conservation of momentum principles and assumes independent upwind and downwind interference factors to determine the forces acting on the turbine blades and turbine performance. The interference factor for a rotor blade at any height is assumed to be constant for the upwind portion of the rotation and also constant, but different, for the downwind portion of the rotation. Wind shear effects and Reynolds number variations along the blade are included in the analysis. Multiple blade cross sections and blade chord lengths may be analyzed by this model. Input includes rotor geometry and blade section data, while output consists of blade loads and turbine performance.

STOCH_3D: STOCH_3D is a numerical analysis code that determines the aerodynamic loads on vertical axis wind turbine blade elements.[38] The code combines blade element theory (strip theory) with actuator-disc analysis to determine the load on the blade elements. In this respect, it is based upon the SLICEIT code. However, STOCH_3D removes the constant interference factor restriction that is present in SLICEIT. In addition, STOCH_3D models the 3-dimensional turbulence present in the input wind and computes the response of the turbine to this turbulent wind. The code predicts turbulent wind-induced blade loading and overall turbine performance.

VDART TURBO: VDART (Vortex method for the DARrieus Turbine) Turbo is a PC-based numerical analysis code that is similar to the VDART2 and VDART3 codes, except that it assumes that the wake convection velocity can be determined at the time of vortex generation. Both 2- and 3-dimensional versions of this code are available.

VDART2: VDART2 (Vortex method for the DARrieus Turbine in 2 dimensions) is a 2-dimensional version of VDART3.[39]

VDART3: VDART3 (Vortex method for the DARrieus Turbine in 3 dimensions) is a numerical analysis code that uses a vortex lattice method to model the interaction of the turbine with the flow around it.[39] Each turbine blade is modeled as a series of straight 2-dimensional airfoil segments. Each segment is, in turn, modeled as a single bound vortex filament (located at the blade center of pressure) and associated tip vortices. As the turbine rotates, the blade angle of attack changes, changing the lift acting on the blade and the strength of the bound and tip vortices. This results in the shedding of spanwise vortices. The continuous physical process is modeled as a series of discrete steps. The vortices that are shed are transported by the wake at the local fluid velocity, resulting in translation, rotation and stretching of the vortices. The effects of Reynolds number variation, apparent mass, circulatory lift and moment, and dynamic stall are incorporated in the code. It is capable of analyzing the performance of turbines with multiple profile/multiple chord blades. The code predicts detailed blade loading and turbine performance.

VSTOCH: VSTOCH is a numerical synthesis code that simulates the stochastic wind encountered by vertical axis wind turbines.[40] The code simulates the velocity fluctuations of turbulent wind for rotationally sampled points. A first-order convection scheme is used to describe the decrease in streamwise velocity as the flow passes through the wind-turbine rotor. The VSTOC simulation is independent of the particular analytical technique used to predict the aerodynamic and performance characteristics of the turbine.

STRUCTURAL DYNAMICS

CDD: CDD (Create Double Deck), Version 1.8, is a preprocessing procedure for the finite element code NASTRAN. CDD asks for a user-supplied "bulk data deck" and adds to it the appropriate executive and case control entries for subsequent execution of either FEVD (complex eigenvalue solution) or FFEVD (complex forced frequency solution). On completion of CDD a complete NASTRAN input deck is ready to be submitted for NASTRAN execution by the procedure SN.

DMGUTY: DMGUTY is a numerical analysis code that determines section properties (moments of inertia and cross-sectional areas) for a ribbed airfoil with a NACA-OOxx or user specified profile.

DMG6G: DMG6G is a pre-processor for the finite element analysis code NASTRAN. DMG6G develops a basic "bulk data deck" for NASTRAN analysis from a "scratch file" that was developed by DMG6S. Options include plotting the finite element grid (isometric or orthographic view), determining the troposkien shape for a given blade geometry and weight distribution, and allowing the user to make adjustments to the straight-curved-straight blade shape to more closely approximate a troposkien.

DMG6S: DMG6S is a numerical analysis code that, interactively, produces an output file that describes the geometry and structural properties of a vertical axis wind turbine. The blade shape is assumed to be a straight-curve-straight approximation to a troposkien. This output file is developed into a "bulk data" deck by DMG6G.

FEVD: FEVD is a NASTRAN preprocessor that incorporates rotational effects for a vertical axis wind turbine into a user-supplied "bulk data deck" for subsequent analysis by the finite element code NASTRAN.[41] FEVD first takes a NASTRAN bulk data deck for a VAWT and sets up a series of geometric nonlinear analyses to determine the stiffness matrix of a rotating turbine subjected to centrifugal and gravitational loading.

This stiffness matrix is then adjusted for centrifugal softening and, together with a damping matrix that includes Coriolis effects, incorporated into the deck for a complex eigenvalue solution by NASTRAN (Solution 67). The procedure CDD executes FEVD and the resulting NASTRAN input deck is submitted for execution by the procedure SN.

FFEVD: FFEVD is a NASTRAN preprocessor code that incorporates rotational effects for a vertical axis wind turbine and the loads imposed on it into a user-supplied "bulk data deck" for subsequent analysis by the finite element code NASTRAN.[42] FFEVD first takes a NASTRAN bulk data deck for a VAWT and sets up a series of geometric nonlinear analyses to determine the stiffness matrix of a rotating turbine subjected to centrifugal and gravitational loading. This stiffness matrix is then adjusted for centrifugal softening and incorporated into the deck for a complex forced frequency solution by NASTRAN (Solution 68) along with a damping matrix that includes Coriolis effects into the solution. Aerodynamic loads on the blades are computed by FFEVD using a double multiple streamtube aerodynamic model and incorporated into the bulk data deck. The procedure CDD executes FFEVD and the resulting NASTRAN input deck is submitted for execution by the procedure SN.

FLUTR: FLUTR is a numerical analysis code that incorporates aeroelastic effects into the analysis techniques used in FEVD to analyze a vertical axis wind turbine.[43] FLUTR determines the flutter speed for the turbine rotor.

FRAST: FRAST is a numerical analysis code that incorporates aeroelastic effects into the analysis techniques used in FFEVD to analyze a vertical axis wind turbine.[43] FRAST determines the aeroelastically-damped response of the turbine blade.

SN: SN (Submit NASTRAN), Version 2.4, is a pre- processor code that submits a double "bulk data deck" for the finite element analysis code NASTRAN to a Cray computer for execution. A user first executes the interactive program SN.COM on a VAX which in turn sends a subsequent non-interactive program, SN.JCI, to the Cray be executed.

SNAPP: SNAPP (Submit Nastran APPlication), Version 2.0, is a post-processor for the finite element analysis code NASTRAN. SNAPP uses the public domain Graphics Compatibility System (GCS) graphics library. SNAPP reads the data files created by the NASTRAN post-processor TRANS. SNAPP outputs data in both tabular and graphical forms.

TRANS: TRANS (TRANSlate NASTRAN data), Version 2.0, is a post-processor for the finite element analysis code NASTRAN. It extracts unformatted data from several NASTRAN OUTPUT2 arrays and rewrites the data into new unformatted data files. The basic function of TRANS is to reduce the size of the NASTRAN output data files for further post-processing. TRANS automatically distinguishes which type of NASTRAN procedure is being used and creates the appropriate files. Output files from TRANS are formatted for processing by another NASTRAN post-processor, SNAPP.

TRES4D: TRES4D is a computer code that calculates the rotor modal loads for a VAWT subjected to turbulent winds.[44] Suitable time series of turbulent wind velocities are generated and passed through a double multiple streamtube aerodynamic representation of the rotor. The aerodynamic loads are decomposed into components of the real eigenvectors of the rotor and subsequently into full power- and cross-spectral densities. These modal spectra are submitted as input to a modified NASTRAN frequency analysis to obtain the power spectra of selected responses.

VAWT-SDS: VAWT-SDS (Vertical Axis Wind Turbine Structural Dynamic Simulator) is a general-purpose structural analysis code.[45] The code simulates the stress history of a vertical axis wind turbine in either steady or turbulent winds for both constant-speed and variable-speed operation. Structural inputs to the code include the mass, damping and stiffness matrices for the turbine generated by the finite element analysis code NASTRAN. Aerodynamic data for the blade profile (i.e., coefficients of lift and drag) are also inputs to the code.

FATIGUE

LIFE2: The LIFE2 code[46,47] is a fatigue/fracture analysis code that is specifically designed for the analysis of wind turbine components. The code is written in a top-down modular format with a "user friendly" interactive interface. In this formulation, an "S-n" fatigue analysis is used to describe the initiation, growth and coalescence of micro-cracks into macro-cracks. A linear "da/dn" fracture analysis is used to describe the growth of a macro-crack.

CONTROLS

ASYM: ASYM is a numerical analysis code that evaluates the performance of a wind turbine that is subjected to a stochastic wind.[48,49] ASYM simulates the stochastic nature of wind on an hourly basis using Markov chains and utilizes the Kaimal spectrum to describe the dynamic characteristics of that wind and generates second-by-second wind speeds that drive a time-domain simulation of wind turbine fatigue and energy production. Wind turbine fatigue damage is evaluated by computing a damage rate per cycle, and an annual damage density function. Data for specific materials and Miner's linear cumulative damage rule are used to estimate cumulative fatigue damage of specific components. The code will evaluate the effects of control considerations on wind turbine fatigue life and energy production.

ASYM34M: ASYM34M is a numerical analysis code that is based upon ASYM. The code has been specialized to the analysis of the Sandia/DOE 34-m Test Bed Vertical Axis Wind Turbine at Bushland, Texas.[48,50] These modifications include the ability to model variable-speed operation, the ability to avoid (or quickly pass through) key rpm bands, and the ability to slow the turbine using regenerative braking.

PLCPC: PLCPC is an operator interface program for Allen- Bradley PLC 2/30 programmable logic controllers. The program interrogates the PLC at timed intervals. The information retrieved from the PLC is then displayed for the operator. The program also allows the operator to change process parameters stored on the PLC.

TBCONPLC: TBCONPLC is the general control code for the 34-m VAWT Test Bed. The code implements automatic and manual starts and stops, fixed and variable speed operation, and turbine protection algorithms for a turbine.

VSWTSYM: VSWTSYM is a time-domain simulation code of a generator/controller system. The code simulates wind turbines with variable-speed generators. The mean wind speed and control parameters for the variable-speed generator can be varied in the simulation. Output from the simulation includes wind speed, and rotor and generator speed and torque.

ECONOMICS

ECON90: ECON90 is an economic analysis of vertical axis wind turbines. The code models mid-1990 VAWT costs and performance factors useful for system design and optimization. This code is an update version of the earlier Sandia code ECON16 that was first released to the public in 1983.

CANTILEVER VAWT DESIGNS

For on-shore applications the stabilization and stiffening technique used for the VAWT rotor is typically a system of guy wires. While guy wires are an effective design, structurally and economically, they do pose a problem by increasing the size of the plot of land required for the installation of the turbine. This large footprint is not acceptable for off-shore applications, if for no other reasons than the high capital costs of an extended mounting platform(s) and/or anchors.

Several techniques have been used for designing VAWTs without guy wire supports. These designs typically require two concentric tubes, one fixed and one rigidly attached to the rotor. In these designs, the non-rotating spindle (cantilever) central tube is anchored at the turbine base. The spindle extends up inside the rotating torque tube that is rigidly attached to the blades. Efficient designs typically use a tapered (or, step-tapered) spindle with thrust bearing that keep the rotor from moving either up or down.

The spindle design is a design that will probably become the design option of choice for off-shore applications.

"H" CONFIGURATION

THE MUSGROVE TURBINES

Early cantilever designs for VAWTs centered on the "H" configuration that was originally proposed and developed by Peter Musgrove in the 70s, 80's and 90's.[51] The development of this configuration went through a succession of succeeding larger turbines in England. The designs were developed and tested under the auspices of Robert McAlpine Ltd. and VAWT LTD. The larger turbines in this line started with the 100 kW VAWT-260 and the 130 kW VAWT-450, and ended with the 500 kW VAWT-850.

The VAWT-260 was developed in 1987[52] and was located in Sicily. The turbine was a fixed-geometry "H" design, illustrated in Figure 1b, with a swept area of 260 m^2, a rotor diameter of 19.5 m and a maximum power of 105 kW. The blades were made from glass fiber reinforced composites with a length of 13.3 m and could not be reefed for power control. The cross-arm-to-blade joints were aerodynamically faired.

The variable-geometry VAWT-450 was developed in 1986.[51] The rotor employed a variable-geometry (arrow-head) configuration, shown in Figure 25, that reefed the blades for power control. The rotor was mounted on the top of a fixed concrete tower using a thrust bearing. The torque tube (a solid shaft) ran down the centerline of the outer tower to the gearbox and generator (located at ground level). The turbine had a 25 m diameter, 18 m long vertical blades, and a swept area of 450 m^2. Its rated power was 130 kW at 11 m/s. The blades used a NACA 0015 airfoil with a 1.25 m chord and were constructed using a carbon fiber composite over Nomex honeycomb core.

The VAWT-260 and the VAWT-450 were succeeded by the VAWT-850 in 1990.[53] The VAWT-850 had a rotor diameter of 38 m, 24.3 m long blades, and a swept area of 850 m^2. Its rated power was 500 kW. The design used fiberglass blades with a NACA 0018 airfoil. The reefing system used in the VAWT-450 was not used in this design. The gearbox and the generator for this turbine were mounted vertically in the top of the tower. A premature failure of a blade (due to a manufacturing defect) prevented the comprehensive testing of this turbine. The testing of the VAWT-850 was not continued after the blade failure because its tower and supporting structures were deemed to be too large (and expensive) for a viable commercial turbine. The size of these components was dictated by the large cyclic loads (fatigue loads) produced by the rotating blades.

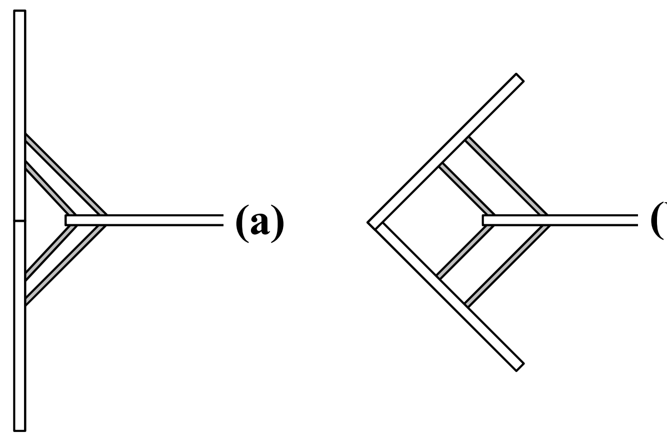

Figure 25. Schematic Diagram of the Musgrove "Arrowhead" Rotor: (a) Full Extension and (b) Reefed.

Not only were these "H" turbines relatively expensive, they had a relatively low aerodynamically efficiency. The aerodynamic losses (drag) at the blade attachment points on each blade (four for the VAWT-450 and one for the VAWT-260 and the VAWT-850) and at the free ends of each blade were so large that they precluded efficient operation. However, the VAWT-260 did reach peak rotor efficiencies in the 40 percent range during testing.[52]

FULL DARRIEUS

SPINDLE CONFIGURATION

The full Darrieus rotor with a center spindle and no guy wires has been examined in the past. The first turbine of this design was a 100 kW experimental turbine, the "Pionier I," that was designed and tested in 1982.[54] This variable-speed turbine had a 2-bladed rotor with a height of 15 m and a diameter of 14.92 m. The blades were shaped into troposkiens. The blades had a NACA 0012 cross section with a blade chord of 0.75 m. They were made of glass fiber reinforced polyester skins over a polyurethane foam core. The design used two main bearings, one at the top of the outer torque tube and the other at the center. Both bearings were placed inside the torque tube. Both the torque tube and the spindle were made of steel tubes of different length, diameter and wall thickness. Operational speed varied from 25 to 50 rpm, and the turbine produced 65 kW at 18 m/s.

To date, this design has not been extended to larger sizes.

SPACE FRAME CONFIGURATION

Another guy-less, full-Darrieus VAWT was the Adecon 19 which used a space-frame support system to eliminate the guy wires.[55] The space frame design lowered the downward thrust on the tower and it reduced the required structural strength and size of the foundation. However, the space frame was considered unsightly and it probably affected the local airflow. The Adecon 19 used hinged blade-to-tower root joints to reduce torque ripple and blade stresses. These joints proved to be difficult to maintain.

The prototype turbine was tested and evaluated at the Atlantic Wind Test Site, Canada, in 1989-90. It was lost in an over-speed event when its braking system failed after accumulating 370 hours of operation. Several commercial turbines, based on this prototype, were later placed in service in Pincher Creek, Alberta, Canada.

FUTURE DIRECTIONS

As mentioned earlier, the U.K. Department of Trade and Industry funded an assessment of the VAWTs that was completed in 1994.1. That assessment is the starting place for this discussion.

AERODYNAMICS

AIRFOILS

Until the early 1980's, Darrieus VAWTs were typically designed with NACA (0015 and 0018) symmetrical airfoils. These airfoils, while aerodynamically efficient in VAWT applications, were designed for aircraft applications, and their performance characteristics presented several problems (primarily, their inability to regulate power) when used for VAWT blades.

The natural laminar flow airfoils designed specifically for VAWT applications by both Sandia and by Somers (for FloWind) have a demonstrated capability to regulate output power of the rotor in high winds. They are also less sensitive to leading edge roughness (bugs) than their NACA counterparts, but their performance with roughness cannot be predicted at this time.

Blade designs that use a NACA airfoil near the roots and an NLF airfoil in the equatorial section appear to be the path to future optimal designs. Initial studies by Kadlec[56] showed a significant reduction in cost of energy using NLF airfoils in the equatorial section.

However, optimal designs will only be achieved with additional understanding and modifications of the airfoils. The impact of roughness on performance is of significant concern. Whether performance is increased (NLF airfoils) or decreased (NACA airfoils) by roughness, any change in maximum output power will result in decreased revenues and/or increased capital costs. With conventional HAWTs of modern design, the changing output due to roughness can usually be mitigated with changes in blade pitch and rotational speed. The output of most VAWT configurations can only be actively controlled with changes in the rotational speed. Thus, we believe that variable speed operation will become the norm for future VAWTs to enable optimal performance with minimal capital cost.

As illustrated by FloWind's EHD analysis and testing, the bend-in-place blade design yielded many challenges to the aerodynamic and structural designer. The Sandia series of NLF airfoils did not yield the desired performance under the design restraints imposed upon this turbine.[****] An alternate NLF airfoil, designed by Somers, was used by

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[****] As discussed above, the rotational rate of the rotor was limited by the gearbox, and the pultrusion process limited the blade to a single, continuous profile along its length. The latter is in contrast to the Test Bed, where rotor efficiency was increased by using a NACA profile for the root sections of the blade.

FloWind to alleviate the performance deficiency, but the initial design had a wall thickness that was too thin to support the blade loads without premature failures. The final design handled the loads, basically, by increasing the wall thickness of the airfoil without changing its exterior dimensions.

In this instance, an increase in the structural stiffness of the blade was achieved by increasing the wall thickness of the airfoil. While this approach can be an effective solution, it can lower the natural frequencies of the turbine (the stiffness increase is offset by the increase in weight). In designs where lowering the natural frequencies creates resonance problems, another approach may be warranted. One approach, that has been used successfully in multi-megawatt HAWTs, is to increase the structural performance (bending stiffness) of the airfoil by increasing its thickness ratio, thus creating the so-called thick-airfoil families. The NLF and/or the Somers airfoils should be extended to thick-airfoil series that have superior structural performance without decreasing their aerodynamic performance.

STRUTS AND JOINTS

An over-riding theme from the turbines discussed above, and also noted in the literature, is the significant reduction in aerodynamic performance resulting from external joints and strut attachment points. This reduction can easily reduce the power output of a turbine by 20 percent. That said, the use of struts to stabilize a VAWT blade is an important design option, and, in many designs, struts are required.

The obvious implication is that struts should be eliminated from designs if at all possible, or at best, moved as close to the rotor-to-tower joints as possible. Fairing these obstructions and moving them away from the equatorial section of the blade can reduce their effects, but additional work is definitely required to optimize their aerodynamic and structural performance. Innovations in future designs, i.e., the thick airfoils discussed above and the use of molded composites discussed below, offer the potential to reduce the use of struts altogether, or at least minimize their effects on aerodynamic performance.

BLADE MATERIALS

As discussed above, the Point Design and the FloWind EHD rotor illustrated that a commercial VAWT in the 500 kW range could be built that was economically competitive with a comparable size HAWT, in the mid 1990s time frame. The economic viability of these turbines was achieved, in part, by using extrusion or pultrusion fabrication processes to manufacture their blades. These manufacturing techniques yielded blades that were cost competitive with their HAWT counterparts, even though they were twice as long. Unfortunately, these manufacturing processes essentially reached their maximum capabilities with the 1.22 m chord for the aluminum bladed Test Bed and the 690 mm chord for the fiberglass bladed FloWind EHD turbine. Blades for rotor systems beyond this size will require larger chords than available using these manufacturing techniques.

The use of molded composites, the blade manufacturing technique of choice for HAWTs, will allow the designer to build blades beyond these manufacturing limits and will provide the designer with the versatility to address aerodynamic and structural inefficiencies associated with constant-chord blade sections. The initial capital costs for blade molds and the high manufacturing costs for molded composite blades are the largest impediment to their implementation in future designs.

Attachments and joints in extruded or pultruded blades have always been a very challenging design problem. With their constant cross-sections, blade sections had to be stiffened to withstand the high stress concentrations associated with these structural elements. Typically, bolted and/or glued “clam shells” and/or internal inserts have been used to strengthen these sections of the blade. However, these reinforcements typically terminate rather abruptly, creating stress concentrations that can significantly reduce fatigue life if not handled properly. Molded composites permit the designer to build taper joint structures within the airfoil structure that will reduce stress concentrations and increase sectional properties.

Joint and attachment structures in VAWT blades have also been associated with significant reductions in the aerodynamic efficiency of the rotor. Molded composites will permit the designer to shroud the joints and attachment points with integrated structures that have aerodynamically-designed contours that increase the efficiency of the blade both aerodynamically and structurally.

Finally, molded composites are currently the material of choice in HAWTs. As the certification organizations now have extensive experience with these materials, designs using molded composites will be easier to certify.

There is still an important manufacturing question that must be addressed in the use of molded composite blades on VAWTs: Should the molded composite blade designs incorporate blade sections that are molded straight to minimize manufacturing costs? If used, the resulting blade would have to be bent for its installation on the rotor, thus inducing significant static and dynamic stresses into the blade, and, probably, increasing the height-to-diameter ratio of the rotor. At this point in time, the jury is still out on this question. Detailed designs and testing will be required to determine if the cost reduction is sufficiently large to warrant an increase in the blade stresses and loss of fatigue life.

Thus, the inherent design options available in molded composite structures will permit the designer to address many of the efficiency problems associated with the design constraints associated with the pultrusion and extrusion manufacturing techniques, namely, only constant chord blade sections with constant structural properties can be used in the design. Also, the current ability of the wind industry to design and construct reliable attachment joints (i.e., hub joints in HAWTs) in composite blades makes molded composite blades an attractive design option. We believe that the obvious direction for future turbines with larger rotors will be with molded composite blades, but the path to an optimized rotor is not obvious because of the high capital cost associated with this design.

DRIVE TRAIN AND POWER COMPONENTS

The power transmission and generation components for a VAWT typically are at, or, in the case of off-shore installations, below grade level. This position allows a design flexibility not available in HAWTs. This inherent advantage has yet to be fully exploited, but is a design concept that has a significant and what appears to be very favorable financial and structural potential in the design of off-shore wind turbines.

GEARBOX AND GENERATOR

The significant difference between VAWT and HAWT gearboxes is the orientation of the low-speed shaft. As most generators are designed to operate in a horizontal position, VAWT configurations dictated that the gearbox not only act as a speed increaser, but that it also must transfer the power through a right angle, typically on the low-speed shaft side of the gearbox. As the wind turbine certification agencies currently only examine parallel-shaft gearboxes, the certification of a non-parallel gearbox will be problematic.

If the generator(s) could be mounted vertically, then a parallel-shaft gearbox with a vertical orientation could be designed. Unfortunately, large, multi-megawatt generators are not designed for vertical installation, but some smaller generators have been.

Even better, with a direct-drive design, the gearbox could be entirely eliminated. Two direct-drive designs seem to have the highest potential, but need extensive study. The first utilizes a direct-drive generator(s) like those that have been developed and used in some HAWTs. In this design, the torque tube is attached directly to the generator, as in several HAWT turbines and the EOLE turbine[57] (a large multi-pole generator). The second design concept utilizes a bull gear design with the torque tube attached directly to the bull gear.

Vertically mounted generator(s) are an important option for VAWTs, both on-shore and off-shore, that need extensive study.

In these designs, the major components of the drive train could be located inside the supporting platform for environment protection and system stability.

TORQUE RIPPLE

The compliant coupling used on the Test Bed was an inexpensive and efficient solution to torque ripple. The FloWind EHD turbine with its 3-bladed rotor did not require a compliant coupling(s) in its drive train.

BRAKES

The braking system for most VAWTs relied on dual or triple independent blade sets to provide primary and fail-safe shut downs of the rotor. In these systems, the primary brake set(s) are used during normal operation of the turbine, but when the primary brake set(s) fail or when the rotor is in a runaway condition, the final (emergency) set is activated. This final set is typically sized large enough to stop the turbine without any additional braking forces (be they from the other brake sets or the generator).[††††]

Many brake systems can provide time-varying torques between full off and full on. Stopping loads can be minimized by gradually applying the brakes. However, the braking schedule must be chosen very carefully to ensure that the brake pads do not overheat and lose their ability to stop the turbine. High blade loads can be experienced as the blades "ring" (at their natural frequency) back-and-forth about the tower when the tower first stops turning in a shutdown sequence. These loads can be minimized by releasing the brakes for a moment when the torque tube first stops, allowing the tower to turn with the blades, and then reapplying the brakes. Thus, programmable service brakes have a significant potential for reducing loads, but, again, the high capital costs and high maintenance of the brake system may preclude the inclusion of variable-torque braking system in future designs.

With a variable speed generator, the use of regenerative braking is an alternate (or, synergistic) path to a low-load braking system.

The brake system can be, and has been, placed on either the high-speed or low-speed shafts. The high-speed shaft location (between the gearbox and the generator) is chosen to minimize the cost of the brake system. In this position, the torque required to stop the rotor is less (lower capital costs for the brake system), but the gearbox is subjected to large loads with each shutdown of the turbine. These loads are sufficiently large that they can reduce the useful lifetime of the gearbox, even when the brake torque is applied gradually. If the gearbox fails, the brake system cannot stop the turbine. The low-speed shaft location is chosen to reduce loads on the gearbox and minimize the chance of a runaway turbine. In this position, the loads on the gearbox are significantly reduced and the rotor can still be stopped with a broken gearbox or coupling. However, the high torque requirement for brakes on the low-speed shaft make them high maintenance items that are subject to failure. Many designs of both types use hydraulically or pneumatically applied brakes. Thus, pump failure can lead to failed brakes and the turbine self-starting when not desired. Finally, cold weather activation is always problematic.[‡‡‡‡] All of the various types of the braking systems need to be continually exercised to ensure they do not "freeze up" and are no longer useable when needed.

For certification, redundant braking systems are a must, and most certification agencies want both a mechanical and an aerodynamic brake to stop a turbine in abnormal conditions before it can self destruct. For pitch controlled HAWTs, both blade pitch and the brakes can be used to stop the turbine. For most VAWTs (and stall-controlled HAWTs), blade pitch is not an available option. Some initial studies and designs used

[††††] The emergency brake set is usually overdesigned to ensure that it can stop the rotor in an over-speed event. The high torque loads on the rotor that are produced by such a full application of the emergency brakes can damage the rotor and power train, if they are not controlled properly.

[‡‡‡‡] The 34-m Test Bed used spring-applied and pneumatically released brakes to alleviate this problem.

flaps for aerodynamic braking. Typically, these flaps (spoilers) were flat plates, located in the equatorial section of the blade for maximum stopping torque. They could be hinged at the trailing edge or somewhere forward. To ensure a uniform stopping torque, flaps had to deploy on both sides of the blade. Designs used both automatic and manual resets. These systems proved to be very problematic, at best. They were difficult to maintain, subject to nuisance deployments, and difficult to reset in icing conditions.

As the reliability of mechanical braking systems for wind turbine systems increased, most aerodynamic breaking systems were abandoned. The path forward for aerodynamic brakes on VAWTs is not obvious at this point. Thus, another challenge for the design of a commercial competitive VAWT is the design and implementation of a reliable aerodynamic brake.

CONFIGURATIONS

THREE-BLADED DESIGNS

During the beginning stages of the development of commercial turbines for electrical generation, many different configurations were examined. VAWT designs soon converged on the 2-bladed configuration because, as with 2-bladed HAWTs, this configuration reduced the capital cost of the turbine with only a relatively small reduction in production relative to a 3-bladed turbine. However, the 2-bladed configuration creates some very challenging design problems, as a direct result of the symmetry of the rotor. For 2-bladed VAWTs, the resultant symmetric loadings are important drivers for the in-plane and out-of-plane vibrational modes of the rotor, the two lowest natural frequencies of the rotor. With a 3-bladed VAWT, there are no in-plane and out-of-plane rotor vibrational modes.

A typical technique used to minimize these turbine loads (for both 2- and 3- bladed VAWTs) involves an increase in the guy-cable stiffness, which is usually achieved by increasing the tension in the cables. The increased tension cascades through the system, requiring a larger torque tube and larger bearings (i.e., increased capital costs). Another technique used to modify the natural frequencies of the rotor is the use of struts, which can be very detrimental to the aerodynamic efficiency of the rotor. In any case, the turbine cannot be allowed to operate near the rotor's natural frequencies. Thus, as with the Test Bed and the FloWind EHD, when a natural frequency lies within or near the operating range of the turbine, either the turbine design must be modified, or the control system must systematically avoid these regions and/or ensure that the turbine passes through them quickly.

The symmetrical loading of a 2-bladed (VAWT) rotor can be eliminated entirely by the inclusion of a third blade. In addition to an increase in the dynamic structural stability of the rotor, the third blade reduces torque ripple in the rotor's torque output, thus reducing the compliance that must be built into the drive train of a 2-bladed VAWT.

Two-bladed designs have the advantage of ground level assembly of the rotor, while 3-bladed designs require installation after the tower is erected.

The issue of 2- or 3-bladed rotors is complicated by the fact that if the turbine design requires a given solidity, then the blades of the 3-bladed rotor must have a smaller chord than that of an equivalent 2-bladed rotor. The resultant lessening of the structural stiffness of the 3-bladed rotor yields designs with deeper struts than those required by an equivalent 2-bladed rotor.

Thus, the question becomes: Is the increased capital cost of the third blade warranted by the increase in structural stability of the rotor (reduced stresses and longer life) and by the reduction in capital costs of other components in the turbine system. For the FloWind EHD rotor, the answer was yes. Perhaps the future optimized VAWT designs will be 3-bladed configurations.

CANTILEVER DESIGNS

"H" Configuration

For off-shore designs, this configuration offers the reliability of a fixed external tower that is shorter than other, equivalent VAWT and HAWT configurations and that will provide better environmental isolation of the drive train components (located inside the tower and/or the support platform) from the corrosive nature of the off-shore environment. The design does require that the main thrust bearing for the rotor be mounted to the top of the fixed tower. Mounting the gearbox and generator in the top of the tower is probably not a good design option for off-shore applications.

A fixed "H" design that uses molded composite blades and with no reefing capability would reduce the aerodynamic losses at the blade-to-cross-support joint. With molded composites, the joint could be aerodynamically and structurally integrated into its support arm to minimize the joint losses. The 40 percent rotor efficiency measured on smaller turbines must be maintained or even increased if this design is to be used in a commercial turbine. A relatively small increase in the overall blade length could also be used to make up some of these losses The tip losses must be addressed as well.

The center support arms for the blades offer a unique platform for the inclusion of aerodynamic brakes in the system. These arms are in the main flow field of the blades and their horizontal orientation offers a perfect orientation for the deployment of flap-type aerodynamic brakes.

We feel that this configuration offers promise and that it warrants consideration in future designs. The primary design challenge for this configuration is the blade-to-strut joints. These joints will be highly stressed and will probably produce large aerodynamic losses.

"Y" Configuration

The "Y" configuration, also called the "V" configuration, is an interesting design concept for off-shore platforms. As opposed to the "H" design, the blade-to-tower joints are moved inboard to the base of the tower, thus reducing the aerodynamic losses from these joints. However, many of the designs require struts (cables) to stabilize the ends of the cantilevered blades. With molded composite blades, the struts can probably be eliminated entirely, or at least moved down the blade and/or aerodynamically faired to minimize aerodynamic losses.

An important option offered by this configuration is the possible of locating a flatwise hinge in the root joint, the so-called "sunflower" configuration. Such a hinge would permit the blades to be folded (reefed), either towards or away from the tower, thus providing the system with a fail-safe aerodynamic brake system and with a "storm" configuration that could minimize damage from strong off-shore storms, i.e., hurricane force winds. However, hinged blades have proven to be problematic, even on land. They are hard to maintain and susceptible to environmental assaults that can freeze the joints. A hinged configuration in an off-shore environment will have to be designed very carefully to ensure ease of maintenance and protection from environmental factors.

<u>Comment</u>

Whether or not either of the full Darrieus rotor design is superior to the "H" or the "Y" rotors remains to be determined, but the latter two configurations do have a significant economic advantage in that, for a given level of power production, they have shorter blades.

VAWT DESIGNS FOR OFFSHORE DEPLOYMENT

In 2011, Sandia National Laboratories obtained a research grant from the U.S. Department of Energy to investigate the feasibility of VAWTs for offshore deployment. This research effort is focused on the rotor sub-system, and will examine different rotor configurations, novel load control and braking concepts, and new materials and manufacturing techniques for large VAWT blades. The cost-competitiveness of a multi-MW, floating VAWT will also be investigated through a series of design studies. The following potential advantages for VAWTs in the offshore environment will be exploited: the potential to scale to very large machine sizes, the low foundation costs associated with a low center of gravity, and the lower O&M costs associated with a more accessible drive train. The design studies will be enabled by development of new VAWT aero-hydro-elastic design codes as well as new cost models specific to offshore deployment. The design philosophies, design codes, lessons-learned, and knowledge gaps identified in the present report will enable a meaningful and efficient research path to be charted by this, and other, new VAWT research efforts.

SUMMARY

The "best" configuration for off-shore wind turbines remains an open question. HAWT designs have a distinct advantage in that their designs have continued to advance since development of VAWT technology effectively halted in mid-1990s. Large multi-megawatt land-based turbines have been built and tested, and these turbines are economically viable in today's market.

VAWTs, while starting from an inferior position due to the lack of development over the past 15 years, do have significant advantages over HAWTs in off-shore applications. Primary to these is the location of the heavy drive-train components at or below grade (sea level) in the support platform. Their primary disadvantage remains the longer blade length required by the full-Darrieus VAWT configurations and the lack of a proven, reliable aerodynamic braking system. Both the "H" and the "Y" configurations offer potential configurations for variable, cost-effective offshore designs.

We believe that a viable economic VAWT for off-shore applications will have the following characteristics:

1. Molded composite blades that incorporate aerodynamic fairings around all joint structures;

2. VAWT specific NLF airfoils with "thick" cross sections;

3. Variable speed with regenerative braking;

4. Direct-drive power train with vertically-mounted, multiple generators.

A 3-bladed rotor is still a debatable option that will require detailed structural, aerodynamic and economic analyses to determine its true potential and its use in future designs.

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