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A 34-METER VAWT POINT DESIGN*

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ABSTRACT

The Wind Energy Division at Sandia National Laboratories recently completed a point design based on the 34-m Vertical Axis Wind Turbine (VAWT) Test Bed. The 34-m Test Bed research machine incorporates several innovations that improve Darrieus technology, including increased energy production, over previous machines. The point design differs minimally from the Test Bed; but by removing research-related items, its estimated cost is substantially reduced. The point design is a first step towards a Test-Bed-based commercial machine that would be competitive with conventional sources of power in the mid-1990s.

INTRODUCTION

In support of the Congressionally-mandated increased DOE role in technology transfer, Sandia's Wind Energy Division recently initiated an effort to create a point design based on its research turbine, the 34-m VAWT Test Bed. The Test Bed is a unique Darrieus wind turbine whose design and operation incorporates the latest innovations in VAWT technology. These innovations, such as tailored blades and VAWT-specific airfoils, have advanced VAWT technology by increasing energy capture and reducing stresses compared to previous VAWTs. The purpose of the point design is to take a first step towards a Test-Bed-based commercial machine by developing an economical VAWT that incorporates these innovations in a cost-effective manner. This paper discusses the unique features of the Test Bed and the details of the point design, including predicted power levels and structural behavior. Point design cost estimates are examined and compared to those of the 34-m Test Bed.

34-M TEST BED

Description

A picture of the Test Bed is shown in Fig. 1. This variable speed machine is rated at 500 kW in a 12.5 m/s (28 mph) wind at 37.5 rpm. The design operating range spans from 28 to 38 rpm, although it can be operated at any speed from 6 to 40 rpm.

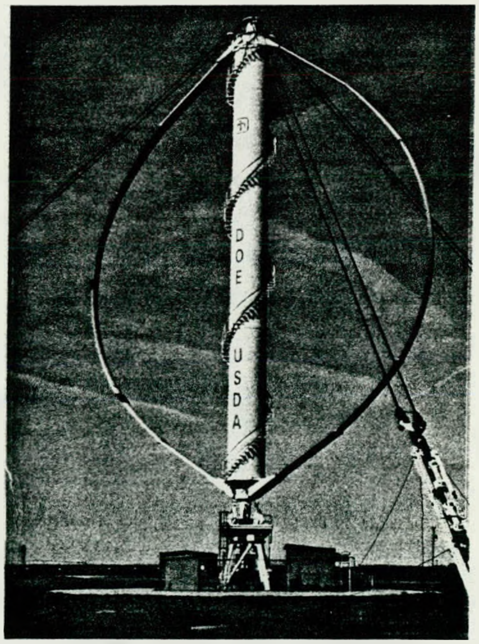

Fig. 1. 34-m Test Bed

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The turbine has several features that are different from previous turbines:

Size. The Test Bed is the largest VAWT is the U. S. and the largest research VAWT in the world.

Blades. The blades are unique in several ways. They were tailored in the design process both structurally and aerodynamically and consist of five different spanwise sections constructed of extruded 6063-T6 aluminum. The two root sections have a 1.22-m (48-inch) chord with a NACA 0021 profile; the equatorial section has a 0.91-m (36-inch) chord with a SAND 0018/50 profile; and the two transition sections have a 1.07-m (42-inch) chord with a SAND 0018/50 profile. The SAND 0018/50 profiles are the first wind-turbine-specific, natural laminar flow (NLF) airfoil sections ever produced. These airfoils are designed to stall at moderate to high winds, thereby regulating power. The blade roots remain NACA in profile to minimize drag in this high angle-of-attack region. The structural design uses a step-tapered nature to the blades to minimize weight and maximize strength. Slope discontinuities of 6 to 7 degrees are incorporated at the blade-to-blade joints to reduce mean stresses by more closely approximating a 37.5 rpm troposkien. (A troposkien is the blade shape that minimizes bending stresses due to gravity and centrifugal loading.)

Tower. The central tower or column is 3.0 m (10 ft) in diameter. The tower is designed to allow for blade changes without tower modifications. This includes the possibility of single blade operation. The structurally-stiff rotor, which consists of the large diameter tower coupled with six 63.5 mm (2-1/2-in.) support cables, maintains the tower in-plane mode crossing of the three per-rev (3P) harmonic above the operating speed.

Generator. The variable-speed-constant-frequency (VSCF) generator and programmable controller permit a range of operating strategies for studying the performance of the turbine and its impact on the utility grid. These strategies include regenerative braking capabilities and the ability to maximize energy capture by changing the rotation rate as a function of wind speed.

Instrumentation. In order to accurately establish machine performance, extensive instrumentation and a comprehensive data acquisition and analysis (DAAS) system are incorporated into the VAWT Test Bed. These systems have advanced the state of art in wind turbine performance analysis. Over 100 data channels provide detailed turbine response that includes blade and tower strains, turbine performance, electrical performance, and environmental conditions.

Structural and Aerodynamic Performance Results

Testing and operation of the 34-m Test Bed generated a substantial quantity of performance data that have been compared to analytical predictions developed in the design process (1,2,3). Measured and predicted aerodynamic performance (shaft power versus wind speed) of the Test Bed at 34 rpm is shown in Fig. 2. Test data and predictions compare very closely except at the very low winds. The decline in power at high wind speeds verifies that the blades, as designed, are providing power regulation. Power regulation is advantageous economically because the drive train, including transmission and generator, can be downsized due to a lower maximum power. These close comparisons hold true at 28 and 38 rpm, as well (3), and provide confidence in our aerodynamic code, CARDAA (4), as a design tool. This code not only predicts power curves for any VAWT of choice, but also determines the input loads for our steady-wind, NASTRAN-based structural forced response code, FFEVD (5).

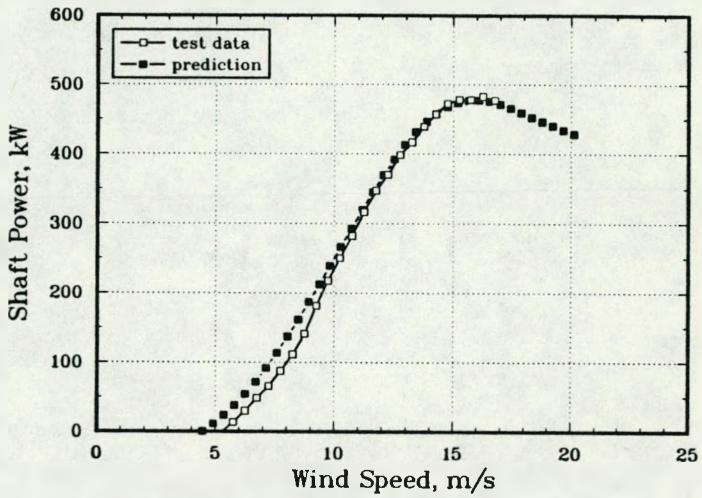

Fig. 2. Predicted and measured shaft power at 34 rpm for the 34-m Test Bed

Dodd (6) compared predicted annual energy capture of the Test Bed to a commercial machine using a Rayleigh wind speed distribution for a site with a 6.3 m/s (14 mph) annual average wind speed. This comparison, Fig. 3, shows an increase of 40 percent more energy (kWh per square meter) with the Test Bed and its laminar flow blades over a 19-m commercial VAWT with NACA blades that is operated in California wind farms. Approximately half of this increase in energy capture is due to the variable speed mode of operation of the Test Bed, and half is due to the blade design with the laminar flow airfoils. The Test Bed demonstrates that significant improvement in energy production is possible with this improved technology.

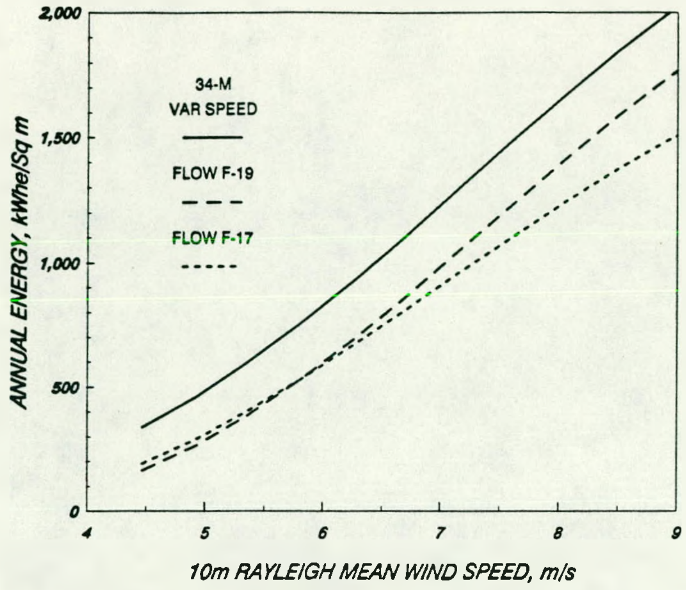

Fig. 3. VAWT energy capture comparisons - zero shear

Comparisons between predicted and measured structural data have provided confidence that the structural model is accurate (1). In a vibrating structure, it is important to know the frequency of structural vibrations and at what rotation rate resonances occur. Figure 4 shows the first eight mode shapes of the stationary Test Bed predicted with our structural code, FEVD (7). These modes were also measured with a modal analysis technique developed at Sandia (8). For this technique, accelerometers are attached to the blades and tower at several places. The structure is excited by the wind or with a cable snap release. Structural response data are processed to produce mode shapes, their frequency of vibration, and damping values. The measurements showed all predicted natural frequencies were within 5 percent of measurements - most were within 2 percent, as shown in Table I.

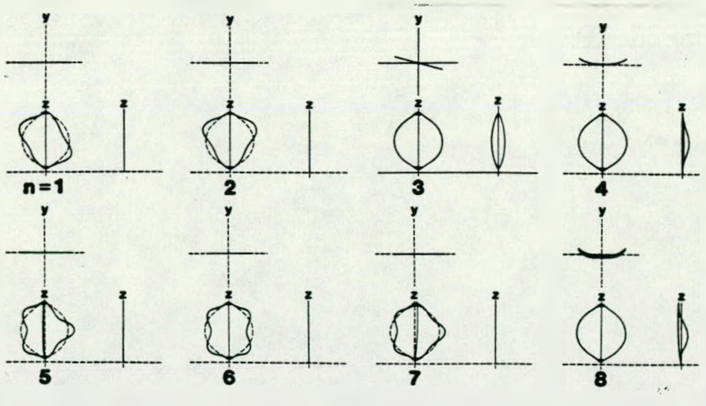

Fig. 4. Predicted mode shapes - 34-m Test Bed

TABLE I. STATIONARY MODAL FREQUENCIES (Hz): MEASURED VS. ANALYTICAL

MODE NUMBER	MODE SHAPE*	MODAL ANALYSIS (WIND EXCITATION)	ANALYTICAL	ANALYTICAL/ MEASURED DEVIATION
1, 2	1FA/1FS	1.06	1.05	1.0%
3	1Pr	1.52	1.56	2.6%
4	1BE	1.81	1.72	5.2%
5	2FA	2.06	2.07	0.5%
6	2FS	2.16	2.14	1.0%
7	1TI	2.50	2.46	1.6%
8	1TO	2.61	2.58	1.2%

*Shape Key:
- 1FA = First flatwise - antisymmetric
- 1FS = First flatwise - symmetric
- 1Pr = First propeller
- 1BE = First blade edgewise
- 2FA = Second flatwise - antisymmetric
- 2FS = Second flatwise - symmetric
- 1TI = First tower in-plane
- 1TO = First tower out-of-plane

Measured mean stresses, both gravity and centrifugal, were compared to those calculated by FFEVD. The comparisons show excellent agreement and verify, as predicted, a 50 percent mean stress reduction due to the Test Bed blade shape, which more closely approximates a troposkien (1). An example of the close comparison between measured and predicted centrifugal stresses at 40 rpm is shown in Fig. 5.

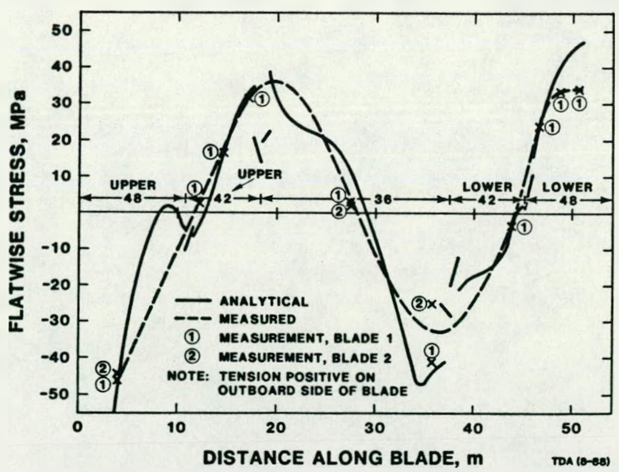

Fig. 5. Centrifugal stress distribution at 40 rpm: analytical vs. measured

Measured vibratory stresses were also compared to steady wind and turbulent wind predictions (2). The steady wind comparisons are close in the low winds, but differ significantly in the high winds. The turbulent wind predictions calculated by TRES4 (9), a NASTRAN-based forced response structural code, show good agreement to measured values at most wind speeds and rotation rates, although more work is required to determine aeroelastic damping levels (2). Fatigue life calculations computed with Sandia's LIFE2 code (10) show that life estimates using analytical data are conservative when compared to those computed using measured data (11).

In summary, operation of the Test Bed has shown the success of the many design innovations. Aerodynamic performance with the new airfoils has shown significant improvements over previous airfoils. Stress-reducing features are successful in lowering mean stresses. The validation of our design tools from Test Bed data provides confidence of their usefulness for specific commercial design activities.

POINT DESIGN

In developing a point design it was appropriate to use the best features of the Test Bed while eliminating the research items, reducing component weights and lowering costs. Although the size of the rotor was not optimized, a 34 meter diameter turbine was chosen based on our experience with the Test Bed, its known costs, and the quality of comparisons between measured and analytical data. The design philosophies of a point design are much different than that of a research machine.

For the case of a research machine, the Test Bed design philosophy can be summarized as follows:

1. Provide a conservative design. This design was the first time our full suite of design tools was used in a design process, and their accuracy was not completely known. Also, fatigue-critical stress concentrations in the blade joints were not well known.
2. Incorporate a modular design. This feature allows change-outs of components, including individual blade sections.
3. Realize that the turbine is a one-of-a-kind construction. Optimization of manufacturing methods for mass production (as would be done for a commercial machine) was not addressed.
4. Allow for single blade operation without counterbalance. This required larger bearings and bearing housings to withstand the inherent eccentric loading.

On the other hand, our point design philosophy is as follows:

1. Minimize the changes from the existing, tested configuration, and maintain the operating stress levels. Recognize that this is just one point design, which is useful in economic studies, and probably not the optimum design.
2. Maintain a maximum power output level of between 500 and 600 kW.
3. Reduce oversized items.
4. Minimize the number of blade joints.
5. Reduce costs by using off-the-shelf items wherever possible.

Two additional constraints placed on the point design because of experience with the 34-m Test Bed are:

Maintain power regulation by retaining the laminar flow blade sections.

Keep the tower and cables stiff enough to place the tower in-plane 3P resonance above the operating speed.

Figure 6 shows a drawing of the 34-m point design and the features that distinguish it from the Test Bed. The major changes from the Test Bed are the following:

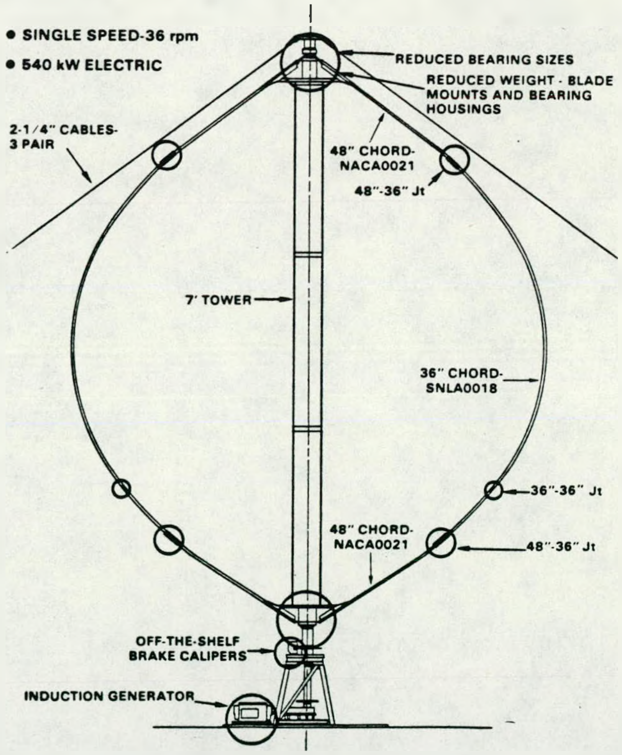

Fig. 6. 34-m point design

1. Tower. The tower is smaller (2.1-m diameter instead of 3.0-m), which restricts the turbine to two-bladed operation.

2. Blades. A laminar flow blade with a 0.91-m (36-inch) chord and SAND 0018/50 profile for the equatorial section maintains the power regulation. The root sections remain a 1.22-m (48-inch) chord, NACA 0021 profile. This configuration eliminates the 1.07-m (42-inch) transition sections on the Test Bed and two blade-to-blade joints on each blade. One additional blade-to-blade joint in the 0.91-m (36-inch) chord equatorial section on each blade is necessary due to limits on the lengths of extrusion (24m). Possible extensions to this extrusion length or slight reductions in the overall turbine size would eliminate this joint.

3. Generator. Operation occurs at 36 rpm in a single speed mode. Two-speed operation is possible, but the variable speed option is eliminated because of its current high cost.

4. Others. Due to the elimination of the requirement for single blade operation, the bearings, bearing housings, and bearing shafts are significantly lightened. The use of off-the-shelf brakes will lower costs substantially as compared to the Test Bed brakes, which were designed and built in-house.

Aerodynamic Characteristics

Figure 7 shows the predicted performance for both the point design and the Test Bed at 36 rpm. The point design shows a power curve very similar to that of the Test Bed. It exhibits power regulation and actually has higher power levels at high winds due to the longer 1.22-m (48-inch) NACA section. Peak shaft power is 585 kW at 17 m/s.

Structural Characteristics

Structurally, the goal was to keep the point design operational stresses at levels no higher than those of the Test Bed and maintain the tower in-plane mode crossing of the 3P harmonic above the operating speed. The tower in-plane mode is so-named because at zero rpm its shape is mostly an in-plane tower motion. By 30 to 40 rpm, however, coupling has occurred with other modes resulting in tower and blade motion (both flatwise and lead-lag). A resonance at the turbine speed where this mode crosses the 3P harmonic can cause large stresses at the root of the blade in the lead-lag and flatwise directions. For example, Fig. 8 shows the measured lead-lag stress levels in the blade

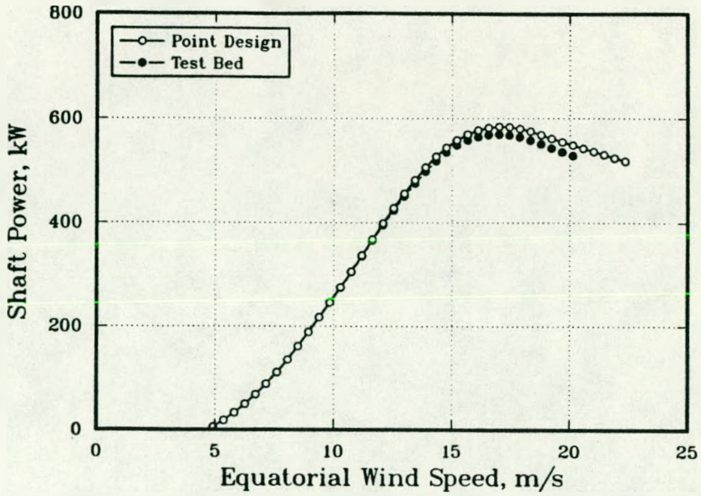

Fig. 7. Predicted shaft power at 36 rpm: 34-m point design vs. 34-m Test Bed

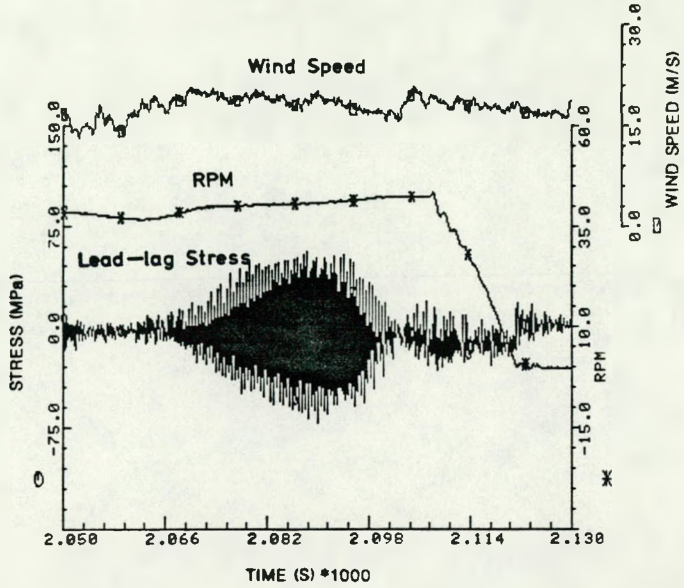

Fig. 8. Lead-lag response during tower in-plane resonance condition (40-42 rpm) for the 34-m Test Bed

when the Test Bed is ramped through this crossing in high winds. Peak-to-peak stresses of some 139 MPa (20,000 psi) were measured - levels of a magnitude that cause rapid fatigue damage. For the point design we decided to reduce the tower and cable sizes as much as possible, but to keep this mode above the operating speed. Several iterations between the aerodynamic code, CARDAA, and the structural codes, FFEVD and FEVD, indicate that we can place the mode slightly above 40 rpm and still retain a rotor that achieves a maximum power of 585 kW at 36 rpm.

Figure 9 shows frequencies of vibration of the tower in-plane mode as a function of rpm for several different point design

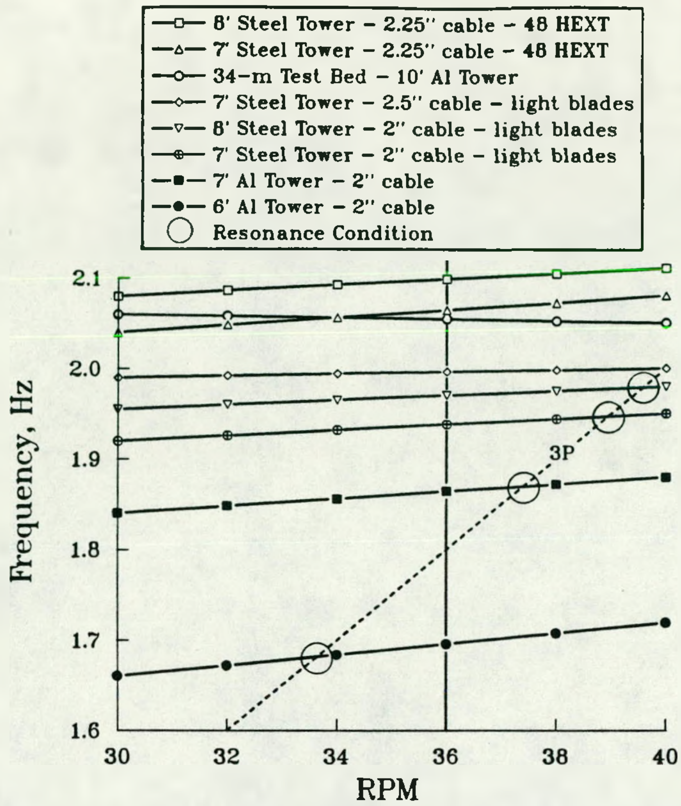

Fig.9. Tower in-plane frequency vs. rpm for different 34-m point design configurations

configurations. Again, the goal was to keep the 3P crossing of the tower in-plane mode at or above 40 rpm to provide some leeway above the operating speed of 36 rpm. Reducing the tower diameter from the Test Bed size of 3.0 m (10 ft) down to 1.83 m (6 ft) drops the mode crossing from above 40 rpm down to around 34 rpm, which is too low. A configuration with a 2.13-m (7-ft) diameter, aluminum tower and 50.8-mm (2-in.) diameter cables raises this crossing up to approximately 37 rpm - still not high enough. A 2.13-m (7-ft) or 2.44-m (8-ft) steel tower with a lighter blade configuration (termed the "light blades" case in Fig. 8) brings this mode up higher; however, stresses in the roots increase substantially. The final configuration of a 2.13-m (7-ft) steel tower with 57.1-mm (2 1/4-in.) diameter guy cables and the blade configuration that has NACA 0021, 1.22-m chord roots and SAND 0018/50, 0.91-m chord equatorial sections (labeled HEXT in Fig. 9) drives the crossing up above 40 rpm - which is acceptable. The final fanplot for the point design is shown in Fig. 10. Operation at 36 rpm is clearly between all predicted resonant conditions.

Predicted operating stresses at 36 rpm in a 11.2 m/s wind for both the point design and and the Test Bed were examined, and, as expected, the stress distribution along the blades is very similar for both cases. Examples of operating stresses are shown in

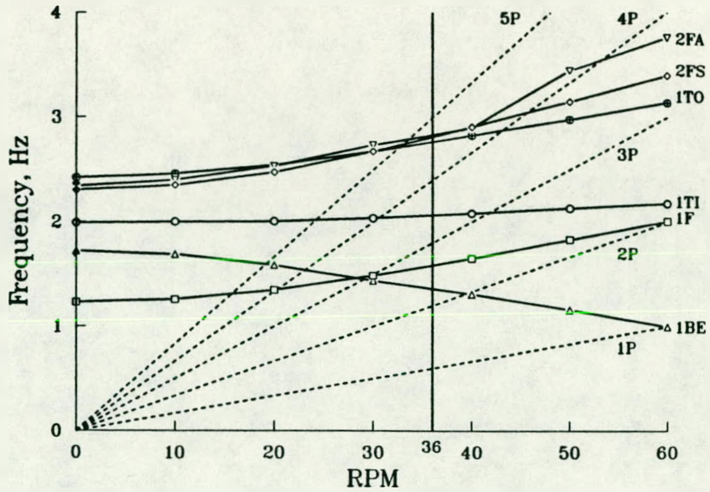

Fig. 10. 34-m point design fanplot

Table II. The upper root in the lead-lag direction is the only location where the point design has a blade operating stress significantly higher than the Test Bed.

TABLE II

PREDICTED OPERATING STRESSES (MPa) AT 36 RPM AND 11.2 m/s

	34-M TEST BED	POINT DESIGN
Upper Root Flatwise (Inner)	6.32 (911 psi)	5.93 (854 psi)
Upper Root Lead-Lag (Trailing)	2.83 (408 psi)	3.11 (448 psi)
Upper Blade-to-Blade Joint (Leading Edge)	3.03 (436 psi) Upper 48-42 Jt.	4.55 (655 psi) Upper 48-36 Jt.

COSTS

Cost estimates for the point design began by stripping out the costs of "research" items from the known costs of the Test Bed. Table III breaks down the Test Bed costs as shown.

TABLE III

TEST BED EXPENDITURES	
PROGRAM	$ 7.4 MILLION
HARDWARE ACQUISITIONS AND SERVICES	$ 3.5 MILLION
HARDWARE (MINUS SERVICES, INSTRUMENTATION)	$ 2.1 MILLION
HARDWARE (MINUS NON-RECURRING COSTS)	$ 1.75 MILLION

The entire five-year Test Bed program totals $7.4 million. Removing development costs (including design, analysis, procurement, and construction management) from that figure leaves an amount of $3.5 million for hardware acquisitions and services. Hardware only (with costs of instrumentation and services removed) comes to $2.1 million. Hardware, excluding non-recurring costs (mostly extrusion dies and blade bending frame), totals $1.75 million. This figure is the hardware cost of a Test Bed turbine (in 1987 dollars) without the instrumentation. This is the starting number, then, for the point design to improve upon.

The costs for the point design were developed from supplier quotes and Test Bed costs. Significant cost reductions were achieved largely in five main areas as summarized in Table IV. By going to a single speed generator, rather than the variable speed generator used on the Test Bed, savings of $135 K were obtained. Reductions of the tower diameter, bearing sizes and associated housings and shafts achieved a large savings of $212 K. Blade costs could be reduced by $139 K with bulk production. The blade connections could save $148 K with simpler designs and their reduced number. And, finally, simpler guy cable attachments and tensioning devices could save $80 K. Other minor cost reductions are not enumerated here.

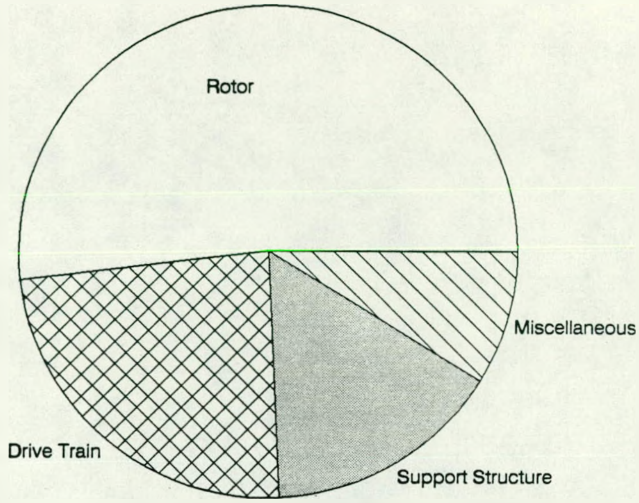

Fig. 11. Point design system costs (one unit)

TABLE IV

POINT DESIGN SAVINGS

ITEM	COST REDUCTION (K $)
GENERATOR (SINGLE SPEED)	135
COLUMN (SMALLER DIAMETER)	212
BLADE CONNECTIONS (LESS)	148
BLADES (BULK EXTRUSIONS)	139
GUY-CABLES (SIMPLER)	80

The final estimated cost of the point design hardware is $800 K in 1990 dollars. (Six percent inflation is assumed each year from 1987.) Figure 11 shows the breakdown of this total cost by subsystem - rotor, drive train, support structure, and miscellaneous. Figure 12 shows a further breakdown of components for each subsystem. Estimated costs are made only for a single commercial turbine and would drop substantially with a large production quantity. Also, the point design still retains some of the conservatism of the Test Bed. A more detailed design of the components would allow for a lighter structure and more cost savings. It is estimated that the cost would drop by at least 50 percent to under $400 K (1990 dollars) with a more detailed design and quantity production.

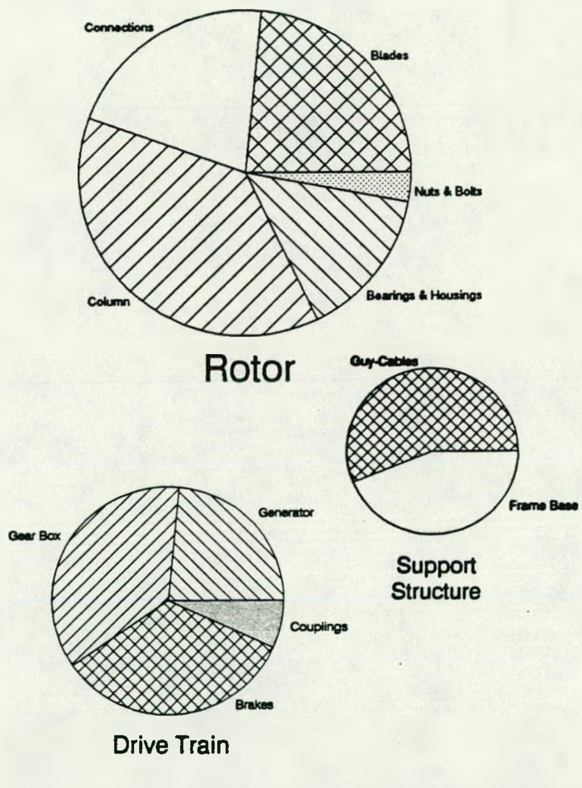

Fig. 12. Point design subsystem costs (one unit)

CONCLUSIONS

The 34-m Test Bed research machine incorporates several innovations that were improvements to Darrieus technology. These improvements increase energy capture over previous machines. Data collected on the Test Bed validate our suite of aerodynamic and structural design tools. The point design differs minimally from the Test Bed with its proven performance and low risk. Initial hardware cost for the point design is $800,000, but with a more detailed design and quantity production this cost should be more than halved. The point design is a first step towards a Test-Bed-based commercial machine that would be competitive to conventional sources of power in the mid-1990s.

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