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INNOVATIVE WIND TURBINES

THE CIRCULATION CONTROLLED VERTICAL AXIS WIND TURBINE

Progress report for March 1, 1976 - December 31, 1976

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FOREWORD

The research described in this report was performed for the United States Energy Research and Development Administration (ERDA), with funding by Contract E(40-1)-5135. This report details programs for approximately the first half of the project, from March 1, 1976 to December 31, 1976. Dr. Robert Thresher, ERDA Division of Solar Energy, is the Technical Monitor for the project; Mr. R. A. Fahey, Oak Ridge, Tennessee, is the ERDA Contracting Officer.

The research is concerned with investigation of concepts for two innovative wind machines which can convert wind energy to more useful forms: a vortex concentrator device, and a vertical axis wind turbine operating with circulation controlled airfoils. The purpose of the work is to investigate the technical feasibility of these concepts, both theoretically and experimentally, and to evaluate the structural and economic aspects of the designs. This report is concerned with the vertical axis wind turbine development. Results for the vortex concentrator device are given in the following report:

Loth, J. L., Ness, N., and Huq, R., "Innovative Wind Turbines; The Wind Vortex Generator", Progress Report for March 1, 1976 - December 31, 1976, ORO/5135-77/2, Department of Aerospace Engineering TR-53, West Virginia University, January, 1977.

Work was accomplished by several members of the staff and graduate students of the Department of Aerospace Engineering of West Virginia University. Portions of this work were performed by personnel at the

Allegany Ballistics Laboratory (ABL) of Hercules, Incorporated. ■

Dr. P. Ward Hill is in charge of the ABL effort. The efforts of individuals in the preparation of sections of this report is acknowledged in the CONTENTS section; the Project Manager, Richard E. Walters, was the report editor. Mrs. Cheryl Fries and Mrs. Carolyn Swecker typed the manuscript.

SUMMARY

This report describes theoretical and experimental research accomplished in evaluating an innovative concept for vertical axis wind turbines (VAWT). The concept is that of using straight blades composed of circulation controlled airfoil sections. The theoretical analysis has been developed to determine the unsteady lift and moment characteristics of multiple-blade cross-flow wind turbines. To determine the drag data needed as input to the theoretical analysis, an outdoor test model VAWT has been constructed; design details, instrumentation, and calibration results are reported herein. Initial testing is with fixed pitch blades having cross-sections of conventional symmetrical airfoils. Future tests will include blade pitch and the use of circulation controlled airfoil sections. Costs of building the test model are included, as well as estimates for blades constructed with composite materials. These costs are compared with those of other types of wind turbines.

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1. INTRODUCTION

In the continuing effort to evaluate various ways to obtain more energy, wind turbines are under extensive study. The many types of turbines and other machines under consideration are in different stages of development. Some, such as the horizontal-axis "propeller-type" turbines, are relatively well developed from a design standpoint. Other types are much newer in their concepts and development procedure. The wind turbine discussed in this report, the circulation controlled vertical axis wind turbine (CC VAWT), is one of the newer types being investigated.

This concept is that of a vertical axis wind turbine which uses straight blades composed of airfoil shapes having high efficiency, i.e., high lift to drag ratios. This would be attained by using circulation controlled airfoils for the blades; these airfoils contain slots near the rounded trailing edges through which a small amount of compressed air is blown to obtain high lift forces. Straight blades allow cyclic pitch control, and also locate each blade element at a large radius from the shaft so that the maximum rotor torque is produced. The vertical axis design allows extraction of power from winds of varying directions with only minor adjustments of the blade pitching and air blowing cyclic schedules. Fig. 1.1 shows a cross sectional view of a circulation controlled airfoil.

As evaluated by using strip theory, the qualitative effect of

using CC blades is shown in Fig. 1.2. These initial calculations showed that in addition to increasing the aerodynamic efficiency of the blades which in turn increases the power obtained, the turbine with CC blades reaches peak efficiency at a lower rotational speed than one with conventional blades. The decrease in rotational speed is important, since it greatly reduces the critical blade centrifugal forces. A more advanced and realistic aerodynamic model has now been developed for calculating the power output of the turbine, and is described in this report.

In the final evaluation of a wind powered energy conversion system (WECS), it is well recognized that the "bottom line" is simply the total costs of installation and operation per unit of power obtained over a certain period of time. Unfortunately, to obtain this information requires several intermediate steps involving knowledge of the WECS's initial cost, operating cost, efficiency, operating lifetime, and the wind intensity-duration characteristics at the installation site. The machine size, aerodynamic and structural design, and construction materials used, are all important factors. In attempting to evaluate these factors for the CC VAWT, it was found that several basic "tools" had to be developed.

Specifically, the aerodynamic theory utilizing a non-steady lifting surface theory was required to calculate the real effects of finite blade size, true blade shed-vortex/blade interactions, and blade pitch change. This theory development has been approached in steps starting with (a) one conventional blade (symmetrical airfoil)

having no pitch change and subsequently adding (b) multi-blades, (c) variable pitch, and finally, (d) CC blade calculations. Steps (a) and (b) are reported herein, together with some parametric calculations for step (b) which are in progress at the present time.

For use with the theoretical aerodynamic calculations, it was necessary to provide input data for the blade and rotor drag coefficients. This requirement led to the development of a test model VAWT at WVU (Fig. 1.3). The primary purpose of this test model is to provide accurate aerodynamic data for computational use. However, the construction and testing of this turbine allows realistic cost analysis, hardware and detail design experience, and operating experience to be obtained which would not otherwise be possible. It is believed to be an important part of the real-world process necessary to develop a practical CC VAWT. Initial testing is with conventional blades, to be followed with higher solidity and CC blades. Mechanisms to change the blade pitch and to supply blowing air to the blades are also under investigation. Some initial aerodynamic tests with low solidity conventional blades (using NACA 0015 airfoils) are presently underway.

The remainder of this report consists of the following parts: section 2 details the non-steady lifting surface theory development, while section 3 describes the outdoor VAWT test model and initial operational results. Sections 4 and 5 concern the economics of wind energy, with summaries of the costs of the WVU design and other WECS designs, respectively.

Fig. 1.1. Cross-section of circulation controlled airfoil.

Fig. 1.2. Calculated effect of circulation controlled blades on vertical axis wind turbine performance.

Fig. 1.3. WVU test model vertical axis wind turbine.

2. NUMERICAL SOLUTION FOR THE UNSTEADY LIFTING CHARACTERISTICS OF VARIABLE PITCH CROSS-FLOW WIND TURBINES

Introduction

A numerical procedure has been developed for determining the unsteady lift and moment characteristics of multiple-blade cross-flow wind turbines. The method employs a force-free wake model, accounts for wake/blade interaction and reflects transient aerodynamics. Introduction of experimental drag data will permit calculation of the energy extraction capacity for such devices.

The basic theoretical approach was reported^{2.1} in September, 1975. The mathematical model was conceptually simple, but development of the computer codes was predicted to be demanding and tedious. The first stage of analysis was therefore confined to the single bladed rotor with emphasis on investigating the effect of computational variables and scrutiny of preliminary results.

This single-bladed analysis was completed in February, 1976, and produced some fascinating results. Of particular interest are the lift coefficient (C_L) transients, the effect of wake-blade interaction on shed vortex trajectories, and observation of an expanded zone of influence upstream of the rotating turbine. Reference 2.2 summarizes the preliminary results and presents relevant details of the methodology. Much of this material was presented^{2.3} at the 11th

Intersociety Energy Conversion Engineering Conference, Lake Tahoe, Nevada, on September 14, 1976.

This report outlines the progress of the extension of the analysis to turbines with multiple blades. During this time computer codes have been modified to handle multi-bladed machines with a variety of blade pitch schedule options.

Review of the Mathematical Model

The problem is approached in the spirit of classical potential theory; that is, fluid viscosity is neglected and the flow is assumed to be irrotational. The blades are assumed to be of infinite span, and their speeds are assumed small enough to treat the flow as incompressible. Under these circumstances lifting characteristics are well predicted theoretically by replacing the airfoil with its mean camber line. The problem is then resolved into one of finding a flow pattern having the instantaneous streamline coincident with the mean camber line (the flow tangency boundary condition at the body) and insuring that the flow leaves the trailing edge smoothly (the Kutta condition).

Using the appropriate mathematical relationships for the flow tangency condition, an expression is derived for the normal velocity (resulting from the free stream wind and the blade motion) of a fluid particle on the blade surface. In terms of the moving non-inertial (x,y) coordinate system, the linearized expression for a symmetrical airfoil is

$$U_n = U_y = x \dot{\alpha} + V_R \alpha \quad (2-1)$$

U_n and U_y are the fluid velocity components normal to the chord line and normal to the relative inflow velocity respectively; α is the blade angle of attack and $\dot{\alpha}$ is its time rate of change. V_R and x are defined in Fig. 2.1.

The loading on the lifting surface is represented by a distribution of bound vorticity, and, for numerical convenience, it is assumed that this distribution can be represented by discrete bound vortices attached to the cambered surface. The physical principle of primary importance in this unsteady problem is the conservation of circulation. This implies that the time rate of change of the bound and wake circulation is zero, and that the net circulation around the airfoil and wake must remain constant. Thus, if the bound circulation changes due to airfoil motion or flow fluctuations, there must be an equal but opposite change in the wake. This is possible when the bound circulation (Γ_b) changes by shedding into the wake vortices of appropriate strength and sense (Γ_w) from the blade trailing edge. The shed vortices convect at their own local velocities and their trajectories describe the wake.

The bound and shed vortices induce velocities everywhere in the flow field. At a "control point" on the blade chord the sum of the normal velocities induced by the bound vortices (u_{nb}) and the

shed vortices in the wake (u_{n_w}) must be equal in magnitude but opposite in sign to U_n in order that the flow tangency condition be satisfied. For the symmetrical airfoil,

$$u_{n_b} + u_{n_w} + U_n = 0. \quad (2-2)$$

Eq. (2-2) is a generalized form of the governing flow equation. The Biot-Savart law provides an expression for the induced velocities in terms of the unknown discrete bound vortices. The velocity induced at $\bar{\zeta}_c = x_c \bar{x} + y_c \bar{y}$ by a bound vortex of strength γ located at $\bar{\zeta}_b = x_b \bar{x} + y_b \bar{y}$ is given by

$$u_x = -\frac{\gamma}{2\pi} \frac{(y_c - y_b)}{(x_c - x_b)^2 + (y_c - y_b)^2}, \quad (2-3a)$$

and
$$u_y = \frac{\gamma}{2\pi} \frac{(x_c - x_b)}{(x_c - x_b)^2 + (y_c - y_b)^2}. \quad (2-3b)$$

Resolved into components in the (ξ, η) and (s, n) coordinate systems, the equations are

$$u_\xi = -(u_x \sin \beta + u_y \cos \beta), \quad (2-3c)$$

$$u_\eta = -(u_x \cos \beta + u_y \sin \beta), \quad (2-3d)$$

$$u_s = (u_x \cos \alpha + u_y \sin \alpha), \quad (2-3e)$$

and
$$u_n = -(u_x \sin \alpha - u_y \cos \alpha). \quad (2-3f)$$

In complex mathematical notation, Eq. (2-3f) is

$u_n = \gamma \text{Im} \{ C_{ij} \}$ where $C_{ij} = C(\bar{\zeta}_{c_i}, \bar{\zeta}_{b_j}, \alpha)$ is a complex constant. If there are ns bound vortices on the chord line (representing the lifting surface) and m shed vortices representing the wake, Eq. (2-2) applied to the i th control point is

$$\sum_{j=1}^{ns} C_{ij} \gamma_j + \sum_{l=1}^m C_{il} \Gamma_w + \{ x \dot{\alpha} + v_R \alpha \} = 0 \quad (2-4)$$

The equation is written for ns control points and a system of ns equations for the ns unknown γ 's is developed.

The generalized aerodynamic forces on the blade are a local lift normal to the inflow velocity, a local drag parallel to the inflow velocity, and a moment about the mid-chord. The lift and moment are derived from Bernoulli's equation for unsteady, incompressible, inviscid flow written in the (ξ, η) frame. Potential theory provides the mechanism for evaluating these equations in terms of the vortex strengths found by application of Eq. (2-4)

Numerical Solution

For single bladed turbines application of Eq. (2-4) to the various control points gives a system of linear equations for the unknown γ 's. With multiple blades, however, the influence of the bound vortices of the additional blades must be considered. If there are p blades, each represented by ns bound vortices, the number of resulting equations is ns times p . The additional

complexity of this situation is not readily apparent, but in developing the computer codes for the multi-bladed case it was quickly ascertained that the computer time required for solution would be significantly increased, if the mathematical model was adhered to rigidly. A numerical technique was therefore developed which considers each blade individually and assumes that blade motion occurs in discrete time steps and is independent of the motion of other blades. In the limit of infinitesimally small time steps this model corresponds exactly to the actual situation of simultaneous blade motion. To clarify the procedure a three bladed turbine having four segments per blade (four bound vortices and four control points) is considered. Figs. 2.2 and 2.3 illustrate the situation.

Blades 1, 2, and 3 are considered sequentially in a clockwise fashion. At some initial time, $t = 0$, the lift on each blade is zero. It is assumed that blade number 1 is impulsively started while blades number 2 and 3 remain stationary. The strength of the bound vortices are found from the following system of equations:

$$C_{11} \gamma_1 + C_{12} \gamma_2 + C_{13} \gamma_3 + C_{14} \gamma_4 = - \left\{ x_1 \dot{\alpha} + v_R \alpha \right\}, \quad (2-5a)$$

$$C_{21} \gamma_1 + C_{22} \gamma_2 + C_{23} \gamma_3 + C_{24} \gamma_4 = - \left\{ x_2 \dot{\alpha} + v_R \alpha \right\}, \quad (2-5b)$$

$$C_{31} \gamma_1 + C_{32} \gamma_2 + C_{33} \gamma_3 + C_{34} \gamma_4 = - \left\{ x_3 \dot{\alpha} + v_R \alpha \right\}, \quad (2-5c)$$

$$\text{and } C_{41} \gamma_1 + C_{42} \gamma_2 + C_{43} \gamma_3 + C_{44} \gamma_4 = - \left\{ x_4 \dot{\alpha} + v_R \alpha \right\}. \quad (2-5d)$$

Since the strength of the bound vortices on the other blades is still set at zero, and no vortices have been shed into the wake, the above terms are the only ones that need be considered. Now the bound circulation is given by $\Gamma_{b_1}^1 = \sum_{j=1}^4 \gamma_j^1$. Following the conservation of circulation principle, a vortex is formed at the trailing edge (Fig. 2.3a); its strength is $\Gamma_{w_1}^1 = - \Gamma_{b_1}^1$. In these equations the second subscript on Γ indicates the blade under consideration. The superscript indicates the mth time step, in this case $m=1$. Since all the required information is available, the position that this shed vortex will have at the next time step is calculated and stored for future use. Ref. 2.3 presents more of the details of the procedure.

The second blade is considered next. It is impulsively started while blades 1 and 3 remain stationary. The strength of the bound vortices are found from expressions similar to Eqs. (2-5) except that the influence of the bound and shed vortices of blade number 1 must be considered. Herein lies the advantage of the simplified numerical technique previously mentioned.

The position* and strength of these vortices are known from the previous calculation. It is not necessary to determine them anew by simultaneous consideration in the equations developed for blade number 2. The strength of the vortex formed at the trailing edge of blade number 2 is $\Gamma_{w_2}^1 = -\Gamma_{b_2}^1 = -\sum_{j=1}^4 \gamma_j$. Its position at the next time step is calculated and stored as was done for blade number 1.

Blade number 3 is considered next. In determining the strength of its bound vortices the influence of the bound and shed vortices of blade numbers 1 and 2 must be considered, since their formation has preceeded the calculations for blade number 3. This completes the procedure for the first time step. Since the strength and position of all vortices are known each time a blade is isolated for consideration, the aerodynamic lift and moment coefficients can be determined. From these the torque and power coefficients of the turbine can be derived.

*It should be noted that in considering a particular blade the position of all vortices in the field must be known in the spatial (x,y) coordinates of that particular blade. This requires that after each calculation the (x,y) coordinates of each shed and bound vortex be transformed to the (ξ , η) system. After proceeding to the next blade the coordinate transformation is reversed so that vortex locations are known in the new (x,y) system.

In succeeding time steps the blades are moved individually through an arc corresponding to an angular increment, $\Delta\theta$. Determination of the vortex strengths proceeds as previously described, taking into consideration all bound and shed vortices of known strength and position. In the general case for the m th time step the shed vortex strength is obtained from the relation,

$$\Gamma_{w_p}^m = \Gamma_{b_p}^{m-1} - \Gamma_{b_p}^m. \quad (2-6)$$

Computer Codes

The numerical procedure described above has been programmed on the West Virginia University IBM 360 computer. The complex arithmetic facility of FORTRAN IV has been used extensively to facilitate the analysis. The acronym CROFTAN I (CROss Flow Wind Turbine Analysis) is applied to the computer code which determines the blade lift and moment coefficients. It consists of approximately 500 cards and requires 130 K of central processor storage. Usually five blade revolutions at $\Delta\theta$ steps of 8° are required to reach a periodic solution to the unsteady problem. This requires substantial computer time. In order to avoid lengthy turn around times, CROFTAN I output is stored on a disk file (also printed and punched on cards) for later processing by the data analysis program, CROFTAN 2. Coincidentally, a library of aerodynamic data is created

which may be reprocessed later pursuant to the investigator's particular interests.

Fig. 2.4 shows the results of some calculations for a fixed pitch turbine with two blades, as obtained from the computer program; the power coefficient is shown plotted vs. the tip speed ratio. Various values of the blade profile drag coefficient, C_{D_o} , are shown on the figure.

Future Work

An extensive parametric analysis of a straight bladed vertical axis wind turbine performance will begin in January 1977. Computational variables* will be checked first to see what combination gives satisfactory results in the least computer time. Then, a matrix of independent variables will be investigated to develop the complete performance data. These will include the following:

1. p , the number of turbine blades.
2. c/R , the non-dimensional blade chord.
3. $\alpha(t)$ or $\alpha(\Theta)$, the blade pitch schedule. (Various options will be considered including fixed pitch, optimum pitch and a number of possibilities in between.)
4. $\omega R/V_\infty$, the ratio of blade rotational speed to the free stream wind speed (tip speed ratio).

*The number of chord segments, n_s , the step size, $\Delta\Theta$, and the number of complete blade revolutions necessary for convergence to a periodic solution, n_r , are called the computational variables.

Data will be presented in carpet plots. This manner of graphical representation is very convenient for observing the effect of several variables simultaneously, for interpolating between curves, and for superposing other constraints.

In addition to the parametric analysis, CROFTAN results will be compared to the WVU experimental data and to other analytical and experimental results available in the literature. In particular, an attempt will be made to deduce turbine drag characteristics by this approach.

Conclusions

Substantial progress has been made in formulating this complex problem in the detail believed necessary to obtain useful results. The computer programming of the multi-blade cross-flow wind turbine analysis is essentially completed and variations for the wind turbine design parameters will follow soon. Results will be compared to the available experimental data.

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Fig. 2.1a. Coordinate systems.

Fig. 2.1b. Four-vortex model of the lifting surface and the shed vortex model of the wake.

Fig. 2.2 Three bladed turbine. (For numerical convenience, it is assumed that each blade advances through $\Delta\theta$ independently of the other blades.)

Fig. 2.3a. Determination of bound and shed vortex strengths for blade number 1 in the first time step.

Fig. 2.3b. Determination of bound and shed vortex strengths for blade number 2 in the first time step. (Blade 2 is influenced by the vortices "previously formed" at blade 1.)

Fig. 2.3c. Determination of bound and shed vortex strengths for blade number 3 in the first time step. (Blades 1 and 2 influence the vortices formed at blade 3.)

Fig. 2.4. Calculated power coefficient (C_p) vs. tip speed ratio (TSR) for a two bladed VAWT. (Key: N_B - number of blades, C - blade chord, R - turbine radius, σ - solidity of rotor.)

3. WVU VAWT OUTDOOR TEST MODEL

Introduction

The basic design features of the WVU outdoor test model vertical axis wind turbine (VAWT) have been described in reference 3.1. In this section, a more detailed description of the VAWT will be given, with particular emphasis on the rotor design.

The purpose of this section is to describe the design, instrumentation, and calibration of the vertical axis wind turbine test model built by the Department of Aerospace Engineering at West Virginia University (WVU). The turbine was designed for research purposes, in order to confirm theoretical performance calculations developed by the Department. Areas of special interest are both aerodynamic performance and structural analysis.

The vertical axis turbine has a central axis perpendicular to the wind direction, around which the blades rotate (Fig. 3.1). Vertical axis machines with fixed blade pitch have the advantage of operating independently of wind direction, and since they are mechanically somewhat more simple than the more popular horizontal axis machine, construction and maintenance costs are potentially lower. The WVU turbine is similar in operation to a Darrieus rotor, in the respect that both generate their torques from the lifts of the blades, not blade drag, as does the Savonius rotor. The difference between the Darrieus rotor and the WVU Vertical Axis Wind Turbine

(VAWT) is that the Darrieus rotor has curved fixed position blades, while the WVU VAWT has straight blades that can be articulated about their quarter chord to produce higher torques. In general, vertical axis turbines that operate by lift have higher efficiencies but have the disadvantage of performing best at high rotational velocities. This is a drawback because the large angular velocities produce large centrifugal forces on the blades. The large centrifugal forces create the possibility of blade failure due to the high stresses. For this reason care had to be taken when designing the WVU VAWT rotor to make sure the blades were adequately supported for the calculated limits of operable angular velocities and wind speeds. Calculations were performed to determine stresses on the blades and support wires. To verify these calculations, the wind machine was extensively instrumented with strain gages on one of the blades and load cells on various guy wires. Also, the instrumentation could be used to monitor the VAWT while operating to keep the blades and other parts from approaching critical stresses when operating at high angular velocities.

The turbine is a two bladed model with a rotor diameter of 3.05 m (10 ft) and blades 3.25 m (10 ft 8 in) in length (Fig. 3.1). The rotor was built with provisions to allow for easy future addition of blades and change of parameters such as blade length and machine radius. The total height of the machine, measured from the top of the shaft to the bottom of the base pads, is 7.62 m (25 ft). Three equally spaced cables connected to a combination

radial/thrust bearing support the top of the shaft. At the base, a cylindrical sleeve was placed over the shaft, which houses two self-aligning radial/thrust bearings. These bearings support the bottom of the main shaft. On the outside of the sleeve, eight legs complete the base of the machine. The four cables that are each attached from one end of the blade to a collar on the shaft are the pre-load tension cables. They add support to the overall structure and extra rigidity to the blades. Lead-lag force cables are also present on the VAWT; these cables help the blade struts resist torsion loads about the shaft during starting, stopping, and wind gusts. They are the cables that run from one end of the strut, through a small pole extended from the shaft and over the opposite strut end. There are four such cables on the machine. The center of each of the blades is supported by two cables which are attached to the blade cuff which clamps over the blade. The other ends of the cables are attached to a center sleeve on a bearing at the middle of the shaft. Each blade has a separate sleeve, so the blades can change pitch independently of each other. These cables will be used to actuate the blades to different blade pitch angles during operation of the VAWT. For the initial testing described here, the blade angles were fixed at 0° (measured perpendicular to the struts). The cables also provide structural support to the center of the blades.

Under the VAWT and inside the instrumentation building, a stand (Fig. 3.2) supports various instrumentation and control devices. The shaft of the VAWT extends through the roof and into the stand as

shown. On the stand are a generator, a motor used to drive the VAWT for starting and testing purposes, a pulley drive system, and instrumentation. The instrumentation on the stand consists of a torque cell for power measurements, a tachometer generator to measure RPM, and 10 high quality low-noise slip rings which allow the various strain gages and other equipment on the rotor to be monitored while it is turning.

Rotor Design

Structural Forces on the Blades of the VAWT

The blades which were selected for initial operation of the wind turbine were originally used on a helicopter. These blades were used to lower construction time and costs, since they were already available. Thus, operational experience could be gained and data collected for later comparison to data obtained with circulation controlled blades.^{3.1}

The blades used were manufactured with a 15.5% thick symmetrical airfoil and a chord length of 17.27 cm (6.8 in). A cross-sectional view of the airfoil is shown in Fig. 3.3. They were constructed of an aluminum alloy with a mass (weight) of 2.091 kg/m (1.4 lb/ft) of span; after adding structural reinforcement and balancing weights, the mass (weight) of the blades was 2.98 kg/m (2.0 lb/ft) of span. The blades have lengths of 3.25 m (10 ft 8 in).

Before the blades could be installed on the machine it was first necessary to determine what operational forces would be. When the wind machine is in operation the blades are loaded approximately uniformly with a combination of aerodynamic and centrifugal forces.

The uniform load on the blade is given by the following equations derived from Fig. 3.4:*

In the X direction

$$F_x = -L \sin \alpha + D \cos \alpha - m \omega^2 R \cos \alpha_0, \quad (3-1)$$

where $-L \sin \alpha + D \cos \alpha$ is due to the aerodynamic loads and $-m \omega^2 R \cos \alpha_0$ is due to centrifugal loads.

In the Y direction

$$F_y = L \cos \alpha + D \sin \alpha - m \omega^2 R \sin \alpha_0, \quad (3-2)$$

where $L \cos \alpha + D \sin \alpha$ is due to the aerodynamic loads and $-m \omega^2 R \sin \alpha_0$ is due to centrifugal loads.

The aerodynamic loads of both F_x and F_y are then put into nondimensional coefficient form giving

$$F_x = \frac{1}{2} \rho V_{REL}^2 c [-C_L \sin \alpha + C_D \cos \alpha] - m \omega^2 R \cos \alpha_0, \quad (3-3)$$

and

$$F_y = \frac{1}{2} \rho V_{REL}^2 c [C_L \cos \alpha + C_D \sin \alpha] - m \omega^2 R \sin \alpha_0, \quad (3-4)$$

Eqs. 3-3 and 3-4 give the uniform loads on the blade. The equations were then nondimensionalized by dividing by $\frac{1}{2} \rho V_{\infty}^2 c$,

*Symbols are defined in Appendix A.

giving the forces in coefficient form, giving

$$C_{F_x} = \frac{F_x}{\frac{1}{2} \rho V_\infty^2 c} = \frac{V_{REL}^2}{V_\infty^2} \left[-C_L \sin \alpha + C_D \cos \alpha \right] - \frac{2m}{\rho} \left(\frac{\omega R}{V_\infty} \right)^2 \frac{1}{Rc} \cos \alpha_o, \quad (3-5)$$

and

$$C_{F_y} = \frac{F_y}{\frac{1}{2} \rho V_\infty^2 c} = \frac{V_{REL}^2}{V_\infty^2} \left[C_L \cos \alpha + C_D \sin \alpha \right] - \frac{2m}{\rho} \left(\frac{\omega R}{V_\infty} \right)^2 \frac{1}{Rc} \sin \alpha_o. \quad (3-6)$$

This was done to make the equations simpler for computer programming. Eqs. 3-5 and 3-6 are now independent of V_∞ (wind speed) and are dependent instead on turbine tip speed ratio, $\omega R/V_\infty$. The value V_{REL}^2/V_∞^2 can be found by trigonometry from Fig. 3.4 to be the following:

$$\frac{V_{REL}^2}{V_\infty^2} = 1 - 2 \frac{\omega R}{V_\infty} \sin \theta - \left(\frac{\omega R}{V_\infty} \right)^2, \quad (3-7)$$

where

$$\alpha = \alpha_i + \alpha_o, \quad (3-8)$$

$$\alpha_i = \tan^{-1} \left[\frac{\sin \theta - \omega R/V_\infty}{\cos \theta} \right], \quad (3-9)$$

and α_o is the pitch angle of the blade. These equations are all derived from Fig. 3.4.

For numerical calculations, the values of $C_L = 0.1\alpha$ and $C_D = 0.02 + 0.026 C_L^2$ may be used, as was done with strip-theory calculations.^{3.2} A computer program (Appendix B) was written to calculate these uniform blade forces.

Structural Analysis of the Blades of the VAWT

After the aerodynamic loads and centrifugal forces were determined for the blades at different tip speed ratios and wind velocities, a structural analysis of the blades was performed. The analysis was needed to determine what method should be used to attach the blades to the VAWT. The blade supports had to provide adequate strength for the blades to maintain stresses and deflections within reasonable limits.

Simply Supported Case

The first design configuration studied was the blade supported at each end by struts from the shaft. Self-aligning bearings at each end of the blade provided supports which are equivalent to a simply supported beam. A uniform load on the blade was assumed, consisting of the aerodynamic loads and centrifugal forces. The loads were determined from the computer program of Appendix B. The loads fluctuate throughout the 360° blade revolution; therefore, only the maximum values were used for the stress, moment, and deflection calculations. The formulas for maximum moment and deflection can easily be derived or found in most structural handbooks.^{3.3} Fig. 3.5 illustrates the blade as a simply supported beam with a uniform load. The maximum moment can be calculated from the following equation: $M = \frac{Wl^2}{2}$, where $l = 1.63$ m (64 in), half the blade length, and the uniform load due to the centrifugal forces

and aerodynamic loads is W.

To find the maximum stress on the blade the equation for bending stress was used:

$$\sigma_b = \frac{M e}{I_x} \quad , \quad (3-10)$$

where e equals the distance between the extreme fiber and the neutral axis of the blade. I_x is the moment of inertia about the chord of the blade and is equal to 5.62 cm^4 (0.135 in^4).

The equation for stress at the midpoint of the blade can be rewritten as

$$\sigma_b = \frac{W l^2 e}{2 I_x} \quad . \quad (3-11)$$

The equation for the deflection at the blade midpoint is

$$\gamma = \frac{5}{24} \frac{W l^4}{E I_x} \quad . \quad (3-12)$$

For the configuration of Fig. 3-5, the only variable in the equations for the maximum stress and deflection is W, the uniform load (lb/in).

Calculations from the previous sections were used to determine a maximum value for W. $\omega R/V_\infty = 6$ and $V_\infty = 6.29 \text{ m/sec}$ (20 ft/sec) are good estimates of actual running conditions. Using these values the maximum W for the blade would be 2.04 kg/cm (11.4 lb/in).

The maximum stress and deflection calculations gave $\sigma_{max} = 6119 \text{ kg/cm}^2$, (86,850 psi) and $\gamma_{max} = 75.2 \text{ cm}$ (29.6 in). These figures were above reasonable limits; therefore, a different method

for attaching the blades had to be devised.

Simply Supported Case with Longitudinal Tension Loads

The second configuration considered used blade pre-tension cables from the ends of the blades to the shaft as shown in Fig. 3.6. These cables reduce the blade stresses and deflection. The maximum stress and deflection occur at the center of the blade as in the simply supported case.

The equation for deflection at any point on the blade can be written as^{3,4}

$$y = \frac{W}{SK^2} \left[\frac{\cosh(Kl - Kx)}{\cosh Kl} - 1 \right] + \frac{W}{2S} (2l - x)x, \quad (3-13)$$

where $K^2 = S/EI_x$ and $S = P \sin 28.8^\circ$. The second derivative of the deflection equation gives the moment equation:

$$\frac{d^2 y}{dx^2} = \frac{W}{S} \left[\frac{\cosh(Kl - Kx)}{\cosh Kl} - 1 \right], \quad (3-14)$$

$$M = EI_x \frac{d^2 y}{dx^2}, \quad (3-15)$$

and

$$M = \frac{W}{K^2} \left[\frac{\cosh(Kl - Kx)}{\cosh Kl} - 1 \right]. \quad (3-16)$$

From the moment equation the stress can be written as

$$\sigma = \frac{Me}{I_x} + \frac{S}{A}. \quad (3-17)$$

If large enough pre-tension loads (P) are applied in the cables, the blade deflections and stresses can be greatly reduced. However, because of limited length of the shaft the maximum cable angles could only be 28.8° ; therefore, for the tension load applied in the cable, only about one half of it acts in the axial direction, S. Tensions of about 2727 kg (6000 lbs) were needed to reduce the stresses to reasonable limits of below 1761 kg/cm^2 (25,000 psi). It was felt that this much tension would be difficult to obtain and maintain because of the possibility of cable failure. For these reasons it was decided that a center wire support should be added between the blades and shaft for extra support, and the loads in the pre-tension cables could be reduced.

Statically Indeterminate Beam With Combined Loading of Uniform Lateral Load and Longitudinal Load

The final blade support configuration considered is shown in Fig. 3.7. With the addition of the center support to the blade, the beam has three simple supports as shown in Fig. 3.8a. The center support cable will also be used to articulate the blade. The equations for moment, stress, and deflection can be solved by using the method of superposition. The blade structural analysis, as shown in Fig. 3.8a, can be separated into two independent problems, as shown in Fig. 3.8b and Fig. 3.8c. The solutions obtained from each can be added together to give the total solution to the original problem.^{3.4}

Using Fig. 3.8b, the solutions for moment, stress, and deflection

were derived. First, summation of moments were taken about the left and right halves; for the left half

$$M = -S\gamma + \frac{R\tau}{2} \quad , \quad (3-18)$$

and from beam deflection theory,

$$M = EI \frac{d^2\gamma}{d\tau^2} \quad . \quad (3-19)$$

Substituting this into the above equation gives

$$\ddot{\gamma} + K^2\gamma = \frac{R\tau}{2EI} \quad , \quad (3-20)$$

where $K^2 = S / EI$.

The homogeneous solution for Eq. 3-20 can be written in the form

$$\gamma_h = C_1 \sinh K\tau + C_2 \cosh K\tau \quad . \quad (3-21)$$

For a particular solution

$$\gamma_p = \alpha\tau^2 + \beta\tau + \gamma \quad , \quad (3-22)$$

$$\dot{\gamma}_p = 2\alpha\tau + \beta \quad , \quad (3-23)$$

and

$$\ddot{\gamma}_p = 2\alpha \quad . \quad (3-24)$$

Substituting Eqs. 3-22 and 3-24 into 3-20 gives

$$2\alpha + K^2(\alpha\tau^2 + \beta\tau + \gamma) = \frac{R\tau}{2EI} \quad .$$

Therefore, $\alpha = 0$, $\gamma = 0$ and $\beta = \frac{R}{2S}$. For the left half, the general solution $y = y_h + y_p$ can be written as

$$y = C_1 \sinh Kx + C_2 \cosh Kx + \frac{R}{2S} x \quad , \quad (3-25)$$

and using the same procedure the solution for the right side can be derived giving

$$y = C_3 \sinh Kx + C_4 \cosh Kx + \frac{R}{2S} (2l - x) \quad . \quad (3-26)$$

The constants C_1 , C_2 , C_3 , and C_4 can now be found using the boundary condition at the end supports, plus the requirement of continuity of deflection and slope at the center of the blade.

The boundary conditions (B.C.) are

1. $x = 0$, $y = 0$,
- and 2. $x = 2l$, $y = 0$.

From B.C. 1 for the left half of the beam,

$$C_2 = 0 \quad , \quad (3-27)$$

and from B.C. 2 for the right half,

$$C_4 = - C_3 \frac{\sinh K 2l}{\cosh K 2l} \quad . \quad (3-28)$$

From the continuity at $x = l$ the deflection equations give

$$C_1 \sinh Kl = C_3 \left(\sinh Kl - \frac{\sinh K 2l}{\cosh K 2l} \cosh Kl \right) \quad . \quad (3-29)$$

The slope equations for left and right halves give

$$C_1 K \cosh K l = C_3 K \left[\cosh K l - \frac{\sinh K 2 l}{\cosh K 2 l} \sinh K l \right] \quad (3-30)$$

Solving 3-29 and 3-30 as simultaneous equations for C_1 and C_3 gives

$$C_1 = \frac{R \sinh K l}{S K \sinh K 2 l} \quad (3-31)$$

and

$$C_3 = \frac{-R \sinh K l \cosh K 2 l}{S K \sinh K 2 l} \quad (3-32)$$

Only the solution for the left half is needed since the blade loadings are symmetrical about the center support. Because of this, the deflections, moments, and stresses are the same on the left half and the right half of the blade. The solution for the left half of the blade can now be written as follows:

$$y = \frac{R \sinh K l}{S K \sinh K 2 l} \sinh K x + \frac{R x}{2 S} \quad (3-33)$$

To obtain the full solution, the beam problem shown in Fig. 3.8c must be solved. The problem is the same as the one discussed before, i.e., a beam simply supported at the ends with longitudinal tension loads. This solution is

$$y = \frac{W}{S K^2} \left[\frac{\cosh (K l - K x)}{\cosh K l} - 1 \right] + \frac{W}{2 S} (2 l - x) x \quad (3-34)$$

The total solution is the sum of both of the deflection equations; however, first the center reaction (R) must be found. To find the center reaction, the sum of the two deflection equations is set

equal to 0 deflection at the point of reaction R, which gives the following:

$$-\frac{R \sinh Kl}{SK \sinh K2l} \sinh Kl + \frac{Rl}{2S} + \frac{W}{SK^2} \left[\frac{1}{\cosh K2l} - 1 \right] + \frac{Wl^2}{2S} \quad (3-35)$$

This can be then reduced to

$$R = \frac{2W}{K2 \sinh^2 Kl - lK \sinh K2l} \left[\frac{\sinh K2l}{\cosh Kl} - \sinh K2l \right] + \frac{Wl^2 K \sinh K2l}{2 \sinh^2 Kl - lK \sinh K2l} \quad (3-36)$$

The total deflection for any point on the left half of the blade from superposition can be written as

$$y = -\frac{R \sinh Kl}{SK \sinh K2l} \sinh Kx + \frac{Rx}{2S} + \frac{W}{SK^2} \left[\frac{\cosh (Kl - Kx)}{\cosh Kl} - 1 \right] + \frac{W}{2S} x(2l - x) \quad (3-37)$$

To obtain the moment equation the second derivative of the deflection equation must be taken. The first derivative gives the slope equation:

$$\frac{dy}{dx} = -\frac{R \sinh Kl}{S \sinh K2l} \cosh Kx - \frac{WK}{SK^2} \left[\frac{\sinh (Kl - Kx)}{\cosh Kl} \right] + \frac{W}{S} (l - x) + \frac{R}{2S} \quad (3-38)$$

Taking the second derivative gives

$$\frac{d^2 y}{dx^2} = -RK \sinh Kl \sinh Kx + \frac{W}{S} \left[\frac{\cosh(Kl - Kx)}{\cosh Kl} \right] - \frac{W}{S} \quad (3-39)$$

The moment equation for beam deflection problems is

$$M = EI \frac{d^2 y}{dx^2} \quad (3-40)$$

To obtain the moment for this particular case Eq. 3-39 is substituted into Eq. 3-40:

$$M = -\frac{R}{K} \frac{\sinh Kl}{\sinh Kl} \sinh Kx + \frac{W}{K^2} \left[\frac{\cosh(Kl - Kx)}{\cosh Kl} \right] \quad (3-41)$$

The stress at any point on the blade can then be determined from the bending stress plus the stress due to the longitudinal tension loads:

$$\sigma = \frac{Me}{I_x} + \frac{S}{A} \quad (3-42)$$

The quantity e is the distance between the extreme fiber and the neutral axis of the blade and can be taken on the top side or bottom side of the blade depending on whether tension or compression is present in the blade.

Fig. 3.9 shows the moment and deflection curves for pre-tension load, S , equal to zero. From the curves it can be seen that the

maximum moments occur at the center of the blade and $\frac{3}{8} \ell$ from the ends of each blade, having values of $\frac{W\ell^2}{8}$ and $\frac{9}{128} W\ell^2$ respectively. The maximum deflection occurs at 0.42ℓ .

A computer program was written to calculate the moments, stresses and deflection for different input values of uniform load and longitudinal tension loads. The analysis showed that the pre-tension loads have little effect on the stresses and deflections. Also, the program results showed the maximum allowable uniform load to be about 2.06 kg/cm (11.5 lb/in) if stress is kept under 1761 kg/cm^2 (25,000 psi). This uniform load corresponds to wind speed of 6.1 m/sec (20 ft/sec) at $\omega R/V_\infty = 6$. The program was also used to compare the results from the experimental static blade load tests, which will be discussed later. The program is located in Appendix C.

Instrumentation

General

The VAWT was instrumented with strain gages for the purpose of measuring the loads on various support wires, blades, and the shaft, when the machine was in operation. Also, strain gages were used to construct a torque cell to measure operating torque, so that power output of the VAWT could be obtained. All of the strain gages, except three load cells that were designed for the three guy wires running from the bearing at the top of the shaft, were located on the

rotor part of the VAWT. This meant that slip rings were needed to connect the wiring from the strain gages on the rotor to the readout equipment. Because slip rings are an expensive item, the slip ring used was limited to 10 connectors. For this reason only one strain gage bridge could be read at a time (other than the torque cell output, which was measured continuously) when the VAWT was in operation. A stepper switch was employed to make it possible to switch from one bridge to another. A 22 position stepper switch was available for this purpose. For reasons that will be discussed later, full bridges had to be employed at all test points. The location of all the strain gages on the VAWT are shown in Fig. 3.10.

Design of Load Cells

The load cells were designed to measure the forces on the structural support wires on the rotor itself and also on the guy wires that support the shaft top bearing. Another use of the load cells was to set desired pre-tension loads in certain wires. Seven of the load cells were designed for the structural support guy wires on the rotor itself. Four of those were designed for the pre-tension cables, two for the center blade support wires, and one was placed on one of the lead-lag wires. The other three load cells were designed to be attached to the three guy wires that run from the top bearing on the main shaft to three support poles located about 10.06 m (33 ft) from the building. The wiring from these three gages did not have to pass

through the slip rings since they were located on the stationary cables. These guy wires were used as a safety feature which would be valuable in case of vibration, and also to add more support to the shaft.

Since drag is a major factor in the performance of the VAWT, the load cell shape and placement were important. For this reason, the seven load cells on the rotor were made cylindrical in shape and were kept short in length. Also, they were located as close to the shaft as possible to lower their angular velocities when the VAWT was in operation. The cells were fabricated of 2024 T-4 aluminum rods of various diameters.

It was necessary to determine what diameters and lengths would be needed to give the load cells satisfactory strength and yet give sensitive strain readings. From the design criteria of the VAWT the expected loads on the load cells could be determined and used to calculate the diameters of the load cells. It was determined from the program for the structural analysis of the blades that pre-tension loads of 1818 kg(4000 lb) would be used on the 4 tension cables on the VAWT to add structural strength to the blades. Also from the program, calculations showed that at high tip speed ratios and high wind velocities the forces on the center wire supports would be around 454.5 kg (1000 lb). This gave an average of 227.3 kg (500 lb) on each blade center support wire. The load on the lead-lag wires was estimated to be around 90.9 kg (200 lb) during operation and slightly higher during stopping, starting, and wind gusts. Using

the load figures given above, and choosing different diameters of aluminum rods that were available, strain calculations were performed. These involved using a form of the deflection equation for axial loads, changing $\gamma = FL/AE$ to $\gamma/L = \epsilon = F/AE$, where $A = \frac{1}{4} \pi D^2$, in order to find what diameters would give sensitive enough strain readings. These calculations also took into account whether the load cells would be big enough in diameter and length to be able to drill 2.54 cm (1 in) deep holes with diameters of 0.635 cm (1/4 in), 0.794 cm (5/16 in) and 0.953 cm (3/8 in). These were to be used to attach the load cells in conjunction with the support wires and tension adjusting turnbuckles. The sizes of the holes that were to be drilled in the ends of the load cells were determined by the sizes of the eye bolts that were used in the turnbuckles. This allowed using a straight piece of bolt threaded on both ends to connect one end of the turnbuckle with one end of the load cells, and in the other end of the load cell the old eye bolt from the turnbuckle could be used.

In designing the three load cells for the guide wire supports running from the top bearing on the shaft to the three poles 10.06 m (33 ft) from the building, load cell size and shape was not a factor. It was decided to use flat plates for the design of the load cells in order to facilitate construction. The maximum tension to be applied in the cables would be about 227.3 kg (500 lb). This would be enough to minimize cable sag and would keep the top of the shaft from moving off center while holding the down force (thrust) on the bearings to a minimum. Again, 2024 T-4 aluminum was the material chosen to be

used for the load cells because of its availability. Doing the calculation for strain from the equation $\epsilon = F/A E$, the cross-section of the load cell was chosen to be 0.3175 cm (1/8 in) by 2.54 cm (1 in).

Construction of the Load Cells

After the load cells were machined the strain gages were applied. Since full bridges were chosen to be used at all data points, the gages had to be placed on the load cells (Fig. 3.11) for the measurement of axial loads. In this bridge configuration, gages 1 and 4 are mounted in the axial direction and gages 2 and 3, referred to as Poisson ratio gages, are mounted laterally. When a tension load is applied gages 1 and 4 will react to the tensile strain, while gages 2 and 3 will sense a compressive strain resulting from the lateral contraction of the load cell. The bridge configuration will provide temperature compensation and cancel any effect from torsion or bending. Also, it provides a bridge constant of $2(1 + \nu)$ where ν is Poisson's ratio and would be 0.03.

The gages used in constructing the load cells were foil-type strain gages and were made by Micro-Measurements. The gages had a resistance of $120.0 \pm 15\%$ OHMS, gage factor of $2.11 \pm 0.5\%$, and a temperature range of -73.3°C (-100°F) to 176.6°C (350°F). These gages were chosen because of their size, relatively low cost, and because of the large pads they had which made it easier to solder

wire leads to them. M-Bond 200 epoxy was used to apply the gages to the surface of the load cells. After the wires were soldered on, a protective acrylic coating was applied to the surface of the gage and to a portion of the wire leads to provide partial protection from weather and damage. Then a silicone coating was applied over the acrylic coating for further protection against weather and possible damage.

For purposes of identification all of the load cells were stamped with numbers from one to ten. The four load cells on the four pre-tension load cables were numbered L-1 through L-4. The two load cells on each of the blade center support wires were numbered L-5 and L-7. The lead-lag wire load cell was stamped L-6 and the load cells on the guy support wires were numbered L-8 through L-10.

When mounted, the wires from the seven load cells on the rotor (L-1 through L-7) were all fastened to the shaft and run down to the bottom of the shaft. There they were wired to the stepper switch box which was fastened to the shaft just above the bottom collar on the shaft. The wires running from the three load cells on the guy wires (L-8 through L-10) were run down one of the guy wires and directly into the instrumentation room bypassing the stepper switch and slip rings, since these load cells do not rotate with the shaft.

Design and Construction of Torque Load Cell

A torque cell was designed for the purpose of measuring the torque output of the VAWT when in operation. From the torque output

and the angular velocity of the machine, the power output of the VAWT can be found.

The torque cell design had to be both sensitive enough to torque changes and also strong enough to be an integral part of the shaft. From the equation for shear stress for a hollow pipe,

$$\tau_{MAX} = \frac{16 M_t}{\pi d^3 \left(1 - \frac{d_i^4}{d_o^4}\right)},$$

and substituting approximate values, an inside and outside diameter was chosen for the torque cell.^{3.3} The torque cell was designed to have an inside diameter of 4.7625 cm (1.875 in) and an outside diameter of 5.08 cm (2.0 in) and was made of type 2024 T3 aluminum. The torque cell is shown in Fig. 3.2, just above the large pulley. The cell strain gage installation is given in Ref. 3.1.

In use, the torque load cell was located above the instrumentation package and above the pulley belt drive system, where the machine was loaded by the generator and/or the motor. The torque cell rotated with the shaft and the wire leads from the cell passed through slip rings which were located inside of the power slip ring assembly. The torque cell had its own set of slip rings, unlike the strain gages on the blades and wires, since the torque was desired to be constantly monitored.

The strain gage bridge on the torque cell was excited by a Brush model 13-4312-00 D.C. bridge amplifier, which also amplified the signal from the bridge. The amplifier output was connected to a voltmeter or strip chart recorder.

Shaft Gages

A full bridge was placed on the shaft to measure bending forces on the shaft during operation. The orientation of the gages is shown in Fig. 3.12. The gages were placed 45.72 cm (18 in) below the center of the shaft. The gages were similar to those on the load cells, and were bonded to the shaft with M-bond 200 epoxy. This strain gage configuration is relatively insensitive to axial and torsion loadings, and provides temperature compensation.

Application of Blade Gages

Strain gages were applied at certain strategic locations on one of the blades to monitor blade stresses during operation of the VAWT. Monitoring the blade stress was important because they were the weakest part in the design, and thus the governing factor in the maximum allowable running speed.

Using the calculations for the deflections, moments, and stresses on the blades developed with the theory previously described, the locations for the strain gages were selected. Sixteen gages were mounted in all; eight on one side of the blade and eight more at the same positions on the other side of the blade. All of the strain gages were placed on only one half of one of the blades, since the stresses, moments, and deflections were the same for each half of the blade. The strain gage positions are shown on Fig. 3.13; gages B-1 through B-8 were mounted on the side of the blade facing the shaft.

Gage B-3 was mounted at the quarter chord 60.96 cm (24 in) from the end of the blade to measure the maximum compressive stress on

that side. Gages B-6 and B-2 were mounted on either side of B-3 at the quarter chord to see how the stresses decreased on both sides of the maximum compressive stress point and gages B-4 and B-5 were mounted behind B-3 to see what effect bending had near the trailing edge. Gage B-8 was located near the middle of the blade where the maximum tensile stress was located. Gages B-7 and B-1 were mounted in the position shown to fill out the rest of the stress curve. Gages B-9 through B-16 were mounted in the same positions on the other side of the blade in the positions shown in Fig. 3.13. They were placed as before to show the maximum stress points, but the tension and compression positions will be reversed. The reason for placing gages on both the top and bottom of the blade was that the stresses on the tension side and compressive side of a point on the blade were not the same. This was because the pre-loaded tension cables added an axial tension load on the blade.

The type of gages used were made by Micro-Measurements, and had a large grid 1.27 cm (1/2 in) by 0.635 cm (1/4 in). The large grid was needed to reduce the possibility of measuring concentrated load points on the skin due to the thinness of the skin on the blades. The gages were 120-ohm gages with a gage factor of $2.125 \pm 0.05\%$.

The gages were bonded to the surface of the blade and then were wired. A flat strip of 16 conductor color-coded 24 gage wire was glued to both sides of the blade near the trailing edge and was used to wire the gages. The strip ran the length of half of the blade and along the strut and to the stepper switch box. The gages were weather

coated with an acrylic coating. A coating of silicone was then applied on the gages and spread over the wires and to the trailing edge for further weather protection and to smooth out the surface of the blade aerodynamically.

Since full bridges had to be used at all test points, three dummy gages had to be employed for each of the 16 gages on the blade. Eight pieces of aluminum bars were used for this purpose. Three dummy gages were placed on each side of the bar and were placed in the stepper switch box. There they were wired to the stepper switches and to the corresponding active gages on the blade. The signals from the blade gages, as for the load cells, pass through the stepper switch box, then through the slip rings, and finally to the bridge amplifier.

Stepper Switch Box

A 22 position stepper switch was used to switch the bridge amplifier signal from one strain gage on the VAWT to another. The stepper switch was needed because only 10 slip rings were available for the instrumentation packet. Four of those slip rings were needed for the torque cell and would be used only for the torque cell, since torque was desired to be constantly monitored. Two more slip rings were used for the two potentiometers which measured pitch angles of the blade. The four remaining slip rings, enough for one full bridge, are used for the 16 blade gages, 10 load cells, and 1 shaft bridge.

This came to a total of 27, which was more than the 20 positions available on the stepper switch after it was wired. The stepper switch was wired to be able to use the 16 blade bridges and 4 other full bridges. It was decided to monitor the 16 blade gages, one of the load cells on the pre-tension wires, one on the lead lag wire, another on the flip wire and, lastly, the bridge on the shaft. This arrangement could be switched around to read other load cells if desired.

Along with the stepper switch mechanism, 48 dummy gages were located in the box. These were installed to be wired to the blade gages to make each of the 16 blade gages a full bridge, providing 3 gages for each gage on the blade. The full bridge arrangement provides complete temperature compensation, which was needed since the stepper switch mechanism and slip rings were being used. The reason for this was that when metallic gages and bridge switching are simultaneously employed, variations in switch resistance can be a problem. There exists the possibility that the change in switch resistance from one stepper switch to the next might be of the same order of magnitude as the strain gage resistance that is of interest, thus resulting in large errors in readings. It was therefore necessary that all the switches be placed completely outside the bridge circuit and complete bridges are switched into and out of the measuring system, which is known as interbridge switching. The bridge balance is not altered using interbridge switching even though the same variations in switch resistance occur as before. The only effect is that bridge sensitivity is altered slightly.

The problems encountered with slip rings are similar to the ones for switching. Factors such as ring and brush wear and changing contact temperature, also cause the need for the use of full bridges at all test points.^{3.5}

The stepper switch box was fastened to the shaft near the bottom, just above the bottom collar to which the pre-tension cables were attached. The leads from the gages were connected to it, connecting with the 4 power and signal leads that run inside the building through the slip rings at the base of the shaft, then to the control box and finally to the strain gage amplifier.

Calibrations

General

Calibrations were performed for the 10 load cells and the torque cell, and static load tests were performed on the assembled rotor. The static load tests were performed in the lab where the VAWT rotor was assembled and supported with the shaft in a horizontal position. The aerodynamic and centrifugal loads on the machine were simulated with weight placed on one of the blades.

Calibration of Load Cells

The load cells L-1 thru L-10 had to be calibrated before they could be installed on the rotor and guy supports of the wind turbine. Load cells L-1 thru L-4 were designed to be placed with the turnbuckles

and the cables of the four pre-tension cables on the rotor and would have tensions of up to 1818 kg (4000 lb). These four load cells were calibrated on a Baldwin Universal testing machine for loads ranging from 0 kg - 1818 kg (0 - 4000 lb) against the corresponding strain in μ in/in using a Budd strain gage indicator.

The load cells were tested with the eye bolts and turnbuckles connected to the load cells to simulate actual operating conditions and to observe whether the turnbuckles and eye bolts could undergo the high tension loads. The calibration curves were all linear. A sample calibration curve for load cell L-1 is shown in Fig. 3.14.

Load cells L-5 thru L-10 had to be calibrated only for maximum loads of 227.3 kg (500 lb). The load cells were calibrated by the method of dead weight testing. The calibration was done for load in pounds verses strain in μ in/in. A Budd Strain Gage Indicator was used to measure the strain. The results of the calibrations were linear curves of load vs. strain.

Torque Cell

The torque load cell was calibrated on a torsional testing machine. Calibration was done for the range of zero to 346.4 (cm-kg) (25 ft-lb) for both increasing and decreasing torque. Both strain and millivolt output were recorded. Both curves were very linear. Only the curve for millivolts vs. in-lb is shown on Fig. 3.15.

Testing

Rotor Static Lab Load Test Procedure

After the load cells were calibrated the VAWT rotor was then assembled in the aerospace lab facility in a horizontal position. The rotor was supported by wooden supports at each end of the shaft. One support was located under the bearing support at the top of the shaft and two other supports were located under the bearing casing near the bottom of the shaft. In this position the static lab load tests were conducted.

The purpose of static tests was to conduct a controlled study of the VAWT under static conditions, and most importantly, to study the effect of different loadings on the blades while varying the pre-tension loads in the tension cables. It was also desired to examine what effect the blade loadings would have on the pre-tension cables, the shaft, the lead-lag wires, and the two center support wires. Another piece of information desired was to determine the forces needed to actuate blade pitch change at different loading conditions. This information was necessary to design a future blade pitch actuating device. Finally, the information from the tests could be used for comparison with the theory from earlier blade analysis.

Sand bags and lead bars were used to apply an incremental uniform load on the lower blades. This was done to simulate the effect of aerodynamic and centrifugal loads that would normally act on the blades

when the machine was in operation. Because just one of the blades was loaded, the shaft was supported at the center using a hydraulic crane. The reason for doing this was that the shaft would otherwise deflect at the center due to the absence of an equal and opposite load on the other blade, as would occur in normal operation.

All of the 24 bridges in the rotor were monitored during these tests. This included the 16 blade gages, 1 shaft bridge, 4 load cells on pre-tension cables, 1 lead-lag wire load cell and 2 load cells on the center cables. The readings from the gages were recorded using strain gage amplifiers. A Vishey-Ellis digital strain gage amplifier with 10 channels was used to record 10 of the blade gage readings, and two Budd strain gage amplifiers, each with 10 channels, were used to record the rest of the data. Strain readings were taken for increasing and decreasing uniform loadings. The tests were performed for three different tensions in the pre-tension wires: 0 kg (0 lb), 227.3 kg (500 lb), and 454.5 kg (1000 lb).

Results of Static Load Tests

Typical data illustrating some of the more important results is given here. Data from each strain gage was compared for the three different pre-tension loads applied during the testing. This was done to study the effectiveness of the pre-tension loads and to study how the strain varied with the addition of blade loading. Also included in the results are comparisons between the static load tests and values obtained from the computerization of the theory for equivalent loading

conditions.

The first two graphs in the results (Fig. 3.16 and 3.17) are data obtained from load cells L-2, and L-3. The location of these gages are shown in Fig. 3.10. Load cells L-2 and L-3 are located on each of the two pre-tension wires that run between the shaft and the ends of the blade on which the uniform load was applied for the tests. It can be observed that the strain readings of the load cells increase with additional uniform load. This trend is especially noticeable for pre-tension loads of 0 kg (0 lb) and 227.3 kg (500 lb). But strain reading for both L-2 and L-3 at 454.5 kg (1000 lb) preload appears to decrease instead.

This tends to indicate that at higher loadings the turn buckles and wires were slacking, thus relieving the load on the wires. This problem was corrected by tightening the connections between the wires. Also from the two graphs it can be seen that increase in the blade loadings does not have any major effect on the tension wires which could cause failure of the cables.

Fig. 3.18 shows test results of load cell L-5, which is located on one of the center blade support wires as shown in Fig. 3.10. The graph shows that strain readings decrease for equivalent distributed loads with increased load in the pretension cables. This trend is in agreement with the theory developed in the program of Appendix C.

A graph for the required force to actuate the blade pitch change is shown in Fig. 3.19. The data is from the tests performed on the two wires with a spring scale and weights described in the test

procedure. Only a few data points were obtained, since the blade load was difficult to distribute evenly fore and aft of the blade quarter chord. Also, once the blade moved slightly, the load center of gravity moved from directly above the quarter chord, often causing a dumping of the load. Although the data has considerable scatter, it is useful to indicate the approximate magnitudes of the forces needed to change blade pitch under load. The differences in leading and trailing edge forces were probably caused by improper placement of the sandbags slightly to the rear of the quarter chord; the two forces would be equal if the load acted at the quarter chord. In practice, the proper design value to be used would probably be an average of that shown by the two curves.

The remaining graphs from the static load tests are based on the blade strain gages results and also blade stress theory. Figs. 3.20 to 3.25 are graphs of test results from the load tests corresponding to blade gages B-3, B-2, B-6, B-4, B-5, and B-8, respectively. All of these are located on the top side of the blade which faces the shaft (Fig. 3.13). The graphs were plotted for strain versus total distributed load for each of the three pre-tension loads tested: 0 kg (0 lb), 227.2 kg (500 lb) and 454.5 kg (1000 lb). Fig. 3.20 is a graph of data from strain gage B-3 located 60.96 cm (24 in) from the end of the blade. This is the location of the maximum compressive stress on that side of the blade. The results of the tests contain some scatter in data, especially at 454.5 kg (1000 lb) preload of tension in the cables, but in general the results are fairly linear. Figs. 3.21 and 3.22 are graphs of strain gage locations B-2 and B-6. These gages are located at 2.62 cm (3 in) on either side of strain gage

B-3 as shown in Fig. 3.13. The values of strain for these two gages should be slightly less than Fig. 3.20 for the same corresponding distributed loads and pretension loads, according to the theory. As it can be seen from the two graphs when compared to Fig. 3.20 this was not always true. This error can probably be contributed to the nonuniformities in the distributed loads represented by the sandbags and lead bars. In general the results are representative of the theory.

Figs. 3.23 and 3.24 are results of test data gathered from strain gages B-4 and B-5 respectively (Fig. 3.13). These gages are located at 60.96 cm (24 in), the same as B-3, but not at the quarter chord, as are the other gages. They were placed behind the quarter chord as shown to study the loading effect at these locations. The strain readings, as can be seen from the two graphs, are mostly less than Figs. 3.20, 3.21 and 3.22, which demonstrates that the major part of the load on the blade acts at the quarter chord. Fig. 3.25 is a graph of readings from blade gage D-0. The gage is located 5.08 cm (2 in) from the middle of the blade which is location of the maximum tensile and overall stress. The graph is fairly linear for tension loads of 0 kg (0 lb) and 227.2 kg (500 lb) but not so for the 454.5 kg (1000 lb) tension load; this could be caused by unevenness in the distributed load. The strain curves decrease with increasing tension loads, as should all of the figures. Deviations away from this trend are probably due to nonuniformities in the distributed load.

The strain gage readings of the bottom side of the blade are not included here, since the same trends occur but of opposite sign. However, the tension load on the bottom of the blade will be slightly larger in magnitude than the corresponding compression load on the top of the blade, due to the added axial pre-tension load on the blade.

The five graphs, Figs. 3.26 to 3.30 are plots of selected static load test results compared with corresponding data of the theory. Fig. 3.26 shows a graph of blades stresses at 60.96 cm (24 in) from the end of the blade on the top surface. The graph is plotted for stress versus total distributed load at 0 kg (0 lb) pre-tension loads. It can be seen that the test matches fairly well with the theory, especially at the lower distributed load values. This graph is for the blade location of maximum compressive stress on the top side of the blade. Fig. 3.27 is a similar curve illustrating the blade stresses at 157.48 cm (62 in) from the end of the blade. This is the location of the gage with the largest tension and overall stress values on the blade. The load test data and theory match extremely well.

The last 3 graphs of this section are Figs. 3.28, 3.29 and 3.30. These graphs are plotted for blade stress versus one half the blade length for a set distributed load and pre-tension. This was chosen to investigate how the overall blade stresses from the tests compare with the theory. Fig. 3.28 is plotted for a total distributed load of 176.4 kg (388 lb) and a pretension of 0 kg (0 lb). The test results match very well with the theory. The bottom of the test

curve is slightly flatter, which is probably due to extra weight added in the middle of the blade from the improper distribution of the sandbags. Fig. 3.29 shows a similar graph for 167.95 kg (369.5 lb) distributed load and 227.2 kg (500 lb) pre-tension load in the cables. The test curve generally has the same shape as the curve from the theory, but is displaced slightly lower. This difference is probably due to improper distribution of the sandbags. In general the test and theory agree well. The last graph of this section is of a total distributed load of 168.5 kg (370.75 lb) and a pre-tension load of 454.5 kg (1000 lb). Fig. 3.30 also agrees well with the theory. Again, the only real difference is at the blade location of 53.3 cm (21 in) to 68.58 cm (27 in) from the end. The error again is probably due to improper test load distribution.

Initial VAWT Outdoor Tests

The VAWT was taken outside to be assembled at the test site after the static load tests were finished. For ease of transportation the machine was disassembled and moved outside in sections. The shaft was placed on the roof of the instrumentation building along with its eight support legs. The stand, with its instrumentation, controls, generator, and starter motor, was then positioned in its place under the shaft inside the building and connected to the rotor shaft by an extension of the rotor shaft. Before the rest of the machine was assembled, initial spin tests were performed, using the motor to rotate the shaft.

Considerable effort was spent to assure the correct operation of the instrumentation and control systems. The guy wire adjustments were made, and set at 68 kg (150 lb) tension; this insured small guy wire vibrations with minimum tension to allow low thrust bearing forces and friction. The struts, cables, and blades of the machine were installed as spin tests progressed. Difficulties which developed included noise pickup from outside RF sources in the strain gage signals, inconsistent torque cell readings, and low torque cell sensitivity. These problems were solved by increasing signal cable shielding, repairing poor cable connections, and reducing the wall thickness of the torque cell from 0.3176 cm (0.125 in) to 0.1587 cm (.0625 in). New strain gages and wiring were installed on the torque cell.

Tests were also performed on the VAWT to evaluate the proper mechanical performance of the machine and its associated instrumentation system. The turbine was run by the electric motor during these initial tests.

A two-trace Brush recorder was used to monitor and record the wind velocity, which was measured with a cup-type anemometer, and also the torque readings from the torque cell. A d.c. tachometer generator, with a voltmeter attached was used to measure the angular velocity of the machine.

Performance measurements, per se, were not made during this initial testing of the VAWT. However, the method for determining the performance of a wind turbine is well known^{3.1} and will not be

repeated here. The measurement requirements to determine power coefficient are torque, rpm, air density, and wind velocity.

Some initial problems with vibrations from the shaft were encountered. It was initially uncertain what the higher limit of machine angular velocity would be. The reason for this was that mild vibrations in the shaft occurred between 120 and 125 rpm. After careful evaluation of vibration levels, it was determined that the machine could be run over 120 rpm. The machine has been successfully run to the motor's rpm limit of 143 rpm with vibrations damping out at values of rpm higher than 120-125.

During the performance testing a few of the strain gages on the blade were connected to the stepper switch and were monitored on a strain gage amplifier. This was done to study the actual loading effects on the blade during increases and decreases of rpm and wind velocity. The results of some of the tests were used for comparison with the static load test results for the same strain gages. Shown on the two graphs for strain gages B-4 and B-8 (Figs. 3.31 and 3.32) are strain vs. total distributed load on the blade. The graphs compare results for the static and dynamic tests at 0 lb load in the pretension wires. From the graphs it can be seen that the data compares very well, and both curves are fairly linear which gives an indication that the strain gages were functioning correctly. The good agreement adds confidence to the validity of the static testing method used on the rotor.

Conclusions

The WVU Vertical Axis Wind Turbine Test Model has been designed and constructed, its instrumentation calibrated, and some initial testing performed. The limited initial testing has shown acceptable system operation. A discussion of design concepts, construction techniques and instrumentation has been given and should be of aid to persons interest in designing similar facilities. Static load tests and some dynamic load tests were performed, and good agreement obtained.

The WVU Vertical Axis Wind Turbine Test Model is a well instrumented test machine which will allow a flexible and useful test program to be accomplished.

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Fig. 3.1. WVU VAWT test model.

Fig. 3.2. Stand with generator, starter motor, and instrumentation/controls.

Chord = 0.174 m (6.85 in)
Maximum Thickness = 3.69 cm (1.06 in)
15.5% Thick Symmetrical Airfoil
Maximum Thickness at 30% Chord

Fig. 3.3. Test model airfoil cross-section

Fig. 3.4. Diagram of forces on the blade.

Fig. 3.5. The blade supported by the two struts.

Fig. 3.6. Blade with two simple supports and pre-tension cables.

Fig. 3.7. Blade with three simple supports and pre-tension cables.

Fig. 3.8. Force body diagram of blade and its two equivalents.

Moment Diagram:

Deflection Curve:

Fig. 3.9. Moment and deflection diagrams.

Fig. 3.10. VAWT rotor with strain gage and load cell locations.

Fig. 3.11. Load cells with gage orientations.

Fig. 3.12. Shaft gage configuration.

B-1 through 8 on Top Half of Blade
 B-9 through 16 on Bottom Half of Blade

Fig. 3.13. Blade gages and their placement.

Fig. 3.14. Calibration curve for load cell #1.

Fig. 3.15. Torque load cell calibration.

Fig. 3.16. Rotor static load test results for load cell on tension cable L-2.

Fig. 3.17. Rotor static load test results for load cell on tension cable L-3.

Fig. 3.18. Rotor static load test results for load cell on center support wire L-5.

Fig. 3.19. Required force for blade pitch change vs. blade loading. (See text, pp. 54-55)

Fig. 3.20. Rotor static load tests for blade strain gage B-3.

Fig. 3.21. Rotor static load tests for blade strain gage B-2.

Fig. 4.22. Rotor static load tests for blade strain gage B-6.

Fig. 3.23. Rotor static load tests for blade strain gage B-4.

Fig. 3.24. Rotor static load tests for blade strain gage B-5.

Fig. 3.25. Rotor static load tests for blade gage B-8.

Fig. 3.26. Blade stresses at 24 in from the end of the blade (0.1b pre-tension load).

Fig. 3.27. Blade stresses at 62 in from the end of the blade (0 lb pre-tension load).

Fig. 3.28. Blade Stresses at 176.4 kg (388 lb) uniform load and 0 kg (0 lb) pre-tension load.

Fig. 3.29. Blade stresses at 168 kg (369.5 lb) uniform load and 227 kg (500 lb) pre-tension load.

Fig. 3.30. Blade stresses at 768.4 kg (370.75 lb) uniform load and 455 kg (1000 lb) pre-tension load.

Fig. 3.31. Rotor tests for blade strain gage B-4.

Fig. 3.32. Rotor tests for blade strain gage B-8

4. COST ANALYSIS OF VERTICAL AXIS WIND TURBINES

Introduction

At the present time, it is not possible to establish definite costs for various sized vertical axis wind turbines. Optimum turbine configurations have not yet been determined, and the exact power produced with such machines is rather uncertain. Optimization of various design parameters, such as machine size, structural materials, etc., remains to be accomplished. However, the cost of the WVU test model has been determined during its construction. Table 4-I summarizes these costs; Table 4-II gives a detailed breakdown of individual items. These figures do not include the research instrumentation unique to the test model, but do include the generator required to produce electrical power and the control system required to perform blade attitude changes. Several items included in this cost analysis were available without cost, such as the generator and low solidity (~ 0.11) conventional airfoil blades; purchase costs have been estimated and included in the analysis.

Cost of the WVU Test Model

Several items should be considered when reviewing the cost analysis. The test model is a singular unit, and no economy could

be attained by using mass-production techniques. The test model is a research machine, and so in many cases the machine is over-designed in the sense that parts were fabricated to provide versatility and large safety factors so that various future tests may be performed. Also, in order to expedite construction, many items were purchased from vendors where minimum time delays would occur, and, therefore, not necessarily at the lowest possible cost. Some extra costs were incurred due to the site (building roof and surrounding terrain) selected, which could be minimized with other sites.

High Solidity Blades

In order to initiate testing as soon as possible, surplus blades, which were originally designed for helicopter use, were installed on the test model. These blades consisted of a symmetrical airfoil section 15.5% thick with a 0.174 m (6.85 in) chord. Two such blades installed on the test model resulted in a rotor solidity of ~ 0.11 . These straight blades are of simple construction, consisting of a c-shaped extrusion which forms the leading edge of the airfoil shape, and to which is bonded an aluminum skin having the top and bottom surfaces riveted together at the trailing edge. Due to this simplicity of design, the costs were estimated on a unit weight basis as being \$5/lb. Even if

this estimate is considered to be low by a factor of two or three, the cost of these relatively small blades would still only be about 10% of the cost of the complete VAWT.

In order to test the aerodynamic performance of the VAWT and to compare results with theoretical calculations, higher solidity blades have been designed and are being fabricated. Fig. 4.1 shows the construction of these blades which involves conventional light aircraft type usage of a main spar, aluminum ribs, and riveted structure. The blade chord is 0.397 m (15.625 in) which gives the two bladed rotor a solidity of 0.26. Construction costs of these blades have not yet been completely determined, but the more extensive labor involved will result in higher costs per unit weight than the smaller blades. This cost will be determined in the near future.

Thus far, it appears that the cost of the actual blades for the straight-bladed rotor, will be much less than that for the curved blade Darrieus VAWT. However, for larger machines the relative percentage of the blade cost might be expected to increase. Also, the cost of the rotor for the straight bladed design must include the costs of blade support arms and other supports; thus final total costs may be comparable.

Circulation Control System

Fabrication of two circulation controlled turbine blades is

also underway. These blades will provide the same rotor solidity (0.26) as the larger symmetrical blades. Construction will be similar, with the sharp blade trailing edge replaced with the round contour and air slots required for the blowing of the circulation control compressed air. This added complexity will increase the blade cost. However, since the blades are straight, mass production techniques would significantly lower the blade cost.

The compressed air system required to provide the blowing air consists of a compressor, ductwork, an air supply "slip-ring" which pressurizes the main rotor shaft, and solenoid valves which control the timing of the air on-and-off cycle to the blade slots. This system is currently under construction, and complete cost breakdowns will be available in the near future.

Cost Estimation of an Operational VAWT Blade Design

It is recognized that for a large operational VAWT, the cost of the blades as a percentage of the total cost will probably be much greater than for the small test models utilized thus far. In order to obtain information which will lead to an evaluation of the economic feasibility of straight-bladed VAWTs, the Allegany Ballistics Laboratory of Hercules Incorporated is participating in the current research effort. The scheduled work includes a) a parametric blade analysis (weight and cost), b) a blade design analysis, c) a

system structural analysis, and d) estimation of facility costs.

The approach selected to accomplish this analysis is summarized in the following list of sequential steps:

1. Find the influence of VAWT operating conditions, material properties, and design features on blade weight and cost.
2. Screen various blade designs for structural efficiency and cost for both conventional and circulation controlled blade types.
3. Select a VAWT operating point and prepare prototype blade designs and analyses.
4. Prepare layouts for the VAWT tower, arms, etc., and design for structural adequacy.
5. Prepare cost estimates of the developed design using the best available performance predictions.

To date, step 1 has been accomplished and results will be given here. The design approach used for the allowable material load-strength relationship is shown in Fig. 4.2. The selected design parameters relating to this load model are the following:

Set:

3 σ Load:	10%	Based on 100 kw and larger units, incorporating high response load control and constant speed rotor design.
2 σ Strength:	10%	Fiberglass, Kevlar composites and metals.
	20%	Type ASMS graphite composite.
F-O-S:	2.0	Man rated, remote, statistical variation mostly removed.

- NMOL: 2.0- Times nominal aerodynamic load to account for
 3.0 gust and inertial (dynamic) effects.
 1.0 Times nominal centrifugal load.

Environment:

Max Operating Temp. = 120°F; Load Cycles = 10^8 ; Endurance = 10^5 hr.

The blade model used for analysis is shown in Fig. 4.3. The assumptions used are the following (symbols are defined in Appendix D):

1. Weight of spar is proportional to total weight of blade.
2. All spar weight is concentrated in the caps, A_c .
3. $\bar{A}_s$ is portion of skin that contributes to bending stiffness.
4. The weight of the ribs and other internal parts other than the spar is equal to the weight of the skin.
5. The blade is a simply supported beam with uniform loading.
6. The airfoil is symmetric and 12% thick.
- 7A. Failure is controlled by stress in the material (spars). Or
- 7B. failure is controlled by a deflection limit.

The equation developed for the mass (weight) of the blade is

$$m (W) = \frac{\beta_2 f_a f_s F_{ay}}{1 - \beta_2 \frac{f_c f_s F_{cy}}{RC}},$$

where for the stress criterion,

$$\beta_2 = \frac{\frac{1}{8} \frac{k_W}{k_T} \frac{\rho_c}{\sigma_c}}{1 + \beta_1 \frac{E_s}{E_c}} \frac{L^2 \rho_A V_\infty^2}{E_s};$$

for the deflection criterion,

$$\beta_2 = \frac{10}{384} \frac{k_W}{k_T} \frac{\rho_c}{E_c} \frac{L^2}{k_T \Delta_M} \frac{L^2 \rho v_\infty^2}{1 + \beta_1 \frac{E_s}{E_c}},$$

and $\beta_1 = \frac{1}{2} k_B \frac{\rho_c}{\rho_s} (k_W - 1)$.

The derivation of these equations will be given in detail in later project reports. It is important to note that two different criteria were used to define failure, the first based on material stress limits and the second based on blade deflection limits. The deflection criteria was investigated since the amount of blade deflection could be a limiting design factor where the blades have mechanisms which change the cyclic blade angle of attack; distortion of the blade shape could cause these mechanisms to become inoperative. An allowable blade deflection of 1% of the blade length was used to define the allowable limit. The blade sizes analyzed were larger than the WVU VAWT test model blades, and were selected as being more typical sizes for an operational design.

Typical results of the study are shown in Figs. 4.4 - 4.7 where blade weights and costs are summarized for the various materials analyzed. The blade weights and resulting costs are seen to be strong functions of the VAWT tip speed ratio. The

minimum gage material weights which are practical to fabricate are shown on the curves. Selection of a stress or deflection limited design would result in a different selection of materials (Figs. 4.4 and 4.5). The costs shown in Fig. 4.6 show that the more-conventional aluminum blade construction is only practical at the lower tip speed ratios, with the composite blades of approximately equal cost allowing machine operation at up to twice the tip speed ratios.

Figure 4.7 shows that the blade length has an appreciable effect on blade weight. This calculation, for the S-2 glass composite material, shows the high sensitivity of the deflection limited design to blade length.

Conclusions

A detailed cost analysis has been performed for the WVU VAWT test model, in order to gain insight as to what approximate costs would be likely for production models of similar configurations. The use of aluminum and composite material blade construction has been studied, and the initial results show promise that the use of the composite materials will be cost-effective, and will also allow machine operation at higher rotational speeds. The overall costs are on the order of those given for other wind turbine designs (see section 5 of this report). Exact performance data must be obtained for the WVU VAWT test model in order to allow a determination of the initial turbine cost per unit of power output.

When this is accomplished a meaningful cost comparison may be made with other machines.

TABLE 4-I

WVU VAWT TEST MODEL COST SUMMARY

<u>ITEM</u>	<u>MATERIAL</u>	<u>LABOR</u>	<u>TOTAL</u>	<u>% OF TOTAL</u>
Support Stand	\$ 486	\$ 638	\$1124	23%
Rotor *	744	712	1456	28
Blades *	164	60	224	4
Guy Wire System	492	181	673	13
Power Generating System and Control System	1073	450	1523	29
Assembly and Painting	33	160	193	4
	<hr/>	<hr/>	<hr/>	<hr/>
Totals	\$2992	\$2161	\$5153	100%
	<hr/>	<hr/>	<hr/>	<hr/>
% of Totals	58	42	100%	
	<hr/>	<hr/>	<hr/>	

*Two conventional symmetrical airfoil blades, 0.174 m (6.85 in) chord.

TABLE 4-II

WVU VAWT TEST MODEL COST BREAKDOWN

<u>ITEM</u>	<u>MATERIAL</u>	<u>LABOR</u>	<u>TOTAL</u>
SUPPORT STAND:			
Building pads (4)	\$ 110	\$ 85	\$ 195
Back up plates (4)	12	15	27
Bolts, nuts	8	-	8
Drill concrete walls	12	40	52
Ground wire	2	1	3
Ground rod	15	5	20
Stand leg angles (8)	86	20	106
Bearing casing and assembly	66	200	266
Bearings (2)	124	-	124
Bearing casing caps (2)	8	20	28
Bearing casing retaining rings	5	20	25
Bearing casing grease fittings	1	5	6
Bearing casing tabs (8)	10	10	20
Attachment bolts	10	1	11
Leg bottom angles (4)	10	5	15
Drill bottom angles/pads	-	15	15
Cut roof holes (3)	-	40	40
Roof flashings (3)	7	20	27
Roof sealing	-	136	136
	<hr/>	<hr/>	<hr/>
	\$ 486	\$ 638	\$1124
ROTOR:			
Main shaft	\$ 34	\$ 250	\$ 284
Shaft top extension	30	10	40
Top bearing cup	15	40	55
Grease fitting-top bearing	1	2	3
Arm shaft fittings (2)	40	40	80
Arm inner inserts (4)	8	10	18
Arms (4)	40	10	50
Arm outer inserts (4)	10	30	40
Arm bearing blocks (4)	40	40	80
Blade bearings (4)	86	-	86
Arm bearing block covers (4)	1	5	6
Arm bearing block fairings (4)	19	5	24
Arm fairing cement	4	-	4
Arm block guy fittings (4)	2	5	7
Bolts	4	-	4
Actuator rings (2)	50	20	70
Actuator ring bearings (2)	50	-	50
Actuator ring shaft sleeve	5	10	15

TABLE 4-II

(continued)

<u>ITEM</u>	<u>MATERIAL</u>	<u>LABOR</u>	<u>TOTAL</u>
ROTOR: (cont'd.)			
Pretension cables (4)	\$ 15	\$ 5	\$ 20
Pretension shaft fittings (2)	40	30	70
Blade end fittings, clamps (4)	10	30	40
Pretension turnbuckles (4)	6	10	16
Torsion wires (8)	20	-	20
Torsion wire struts (4)	8	-	8
Torsion wire clamps	2	-	2
Torsion wire turnbuckles	4	-	4
¹ Pitch actuator mechanism	100	100	200
Pitch actuator cables (4)	10	-	10
Actuator clamps, bearings	16	-	16
Actuator turnbuckles	4	-	4
Brake disk	20	20	40
Brake actuator	40	-	40
Brake actuator attachment	10	40	50
	<hr/>	<hr/>	<hr/>
	\$ 744	\$ 712	\$1456
BLADES (2):			
Bearing inserts (4)	\$ 2	\$ 20	\$ 22
End bolts (4)	2	-	2
² Center cuffs (2)	10	40	50
Blades (2)	150	-	150
	<hr/>	<hr/>	<hr/>
	\$ 164	\$ 60	\$ 224
GUY WIRE SYSTEM:			
Rotor top cap	\$ 30	\$ 40	\$ 70
Rotor top bearing	15	-	15
Guy poles (3)	120	10	130
Concrete	71	-	71
Pole installation	-	80	80
Pole guy ground screws (6)	45	10	55
Pole caps	15	-	15
Guy wire (220')	110	30	140
Guy wire clamps, thimbles	40	-	40
Ground rods (2)	30	10	40
Ground wire	1	1	2
Turnbuckles (10)	15	-	15
	<hr/>	<hr/>	<hr/>
	\$ 492	\$ 181	\$ 673

TABLE 4-II

(continued)

<u>ITEM</u>	<u>MATERIAL</u>	<u>LABOR</u>	<u>TOTAL</u>
POWER/CONTROL SYSTEM:			
Shaft extension	\$ 10	\$ 5	\$ 15
Rainshield	2	5	7
Base bearing	45	10	55
Generator stand	170	40	210
Generator speed increaser	60	80	140
Generator belts	15	-	15
³ Generator	200	-	200
Selsyn control/stand	50	100	150
Wind sensor/controls	100	200	300
Tachometer and drive	21	10	31
Electronic speed control	400	-	400
	<hr/>	<hr/>	<hr/>
	\$1073	\$ 450	\$1523
ASSEMBLY AND PAINTING:			
Crane	\$ -	\$ 130	\$ 130
Paint	33	30	63
	<hr/>	<hr/>	<hr/>
	\$ 33	\$ 160	\$ 193

NOTES:

1. Based on completed design.
2. Surplus item - cost based on \$5/#.
3. Surplus item - cost based on replacement.

Fig. 4.1. High solidity (0.26) blade under construction.

Fig. 4.2. Analysis material load-strength relationship.

Fig. 4.3. Blade model selected for analysis.

Fig. 4.4. Effect of material on blade weight; stress limited design.

Fig. 4.5. Effect of material on blade weight; deflection limited design.

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Fig. 4.6. Effect of material on blade cost; stress limited design.

Fig. 4.7. Effect of blade length on blade weight.

5. COST STUDY OF CURRENT WECS

Introduction

In order to fully evaluate the practical usefulness of any wind energy conversion system (WECS), it is necessary to determine the economics involved in developing and maintaining the system. Most WECSs are in various stages of development, and relatively few have been used long enough to accurately determine the initial and operating costs. As systems develop, these costs will be more accurately determined; at the present time, however, it is useful to evaluate the economic situation in the best way possible, even though the results are preliminary. This information offers assurance that innovative systems are not going to be prohibitively costly, and at least will be priced in the same order of magnitude when compared with some of the WECSs which have been under development for several years.

Reference 5.1, produced previously in the WVU study of innovative wind machines, contains a summary of 1975 prices of some machines available at that time. The approach taken was to attempt to calculate initial installed costs for commercially available WECS without energy storage systems, and to make reasonable estimates of costs of both historical WECS and those under present development. Thus the installed cost of constructing the WVU VAWT could be compared with these costs of other machines. Some other researchers

(e.g., Quinn, ref. 5.2) have carried such evaluations further and have attempted to derive generalized cost formulas for different sizes of turbines, and have considered financing, site, and other costs as well. Even though some of these first cost estimates are preliminary, the refined cost values and identification of critical cost items in future work will be much more accurate, and will evolve from today's economic analysis efforts.

Discussion

The results of the survey made of the costs and specifications of commercially available wind energy systems, and other projects such as the Gedser and Putnam machines, are shown in Table 5-I.

Due to the wide number of "systems" available today, the capital costs were calculated as the cost of the turbine and generator only. Where component prices were not available, estimates were made as to the amount to be subtracted for towers, storage, and controls. Generally, this was about 30% of the systems cost. No allowance has been made for site preparation and other related costs. The only turbines not offered with a generator are the farm types, and this additional cost was not calculated; therefore, for these machines the price per KW would be even higher than shown if a generator were included.

In order to allow for cost inflation since 1975, the 1975 and 1976 prices were increased 13% and 8% respectively, and thus should give a more realistic comparison with the turbines for which 1977

prices were available. These percentages were those obtained from the U.S. Labor Department's Consumer Price Index for the same period. While the estimates involved in obtaining the data shown will cause some errors, it is believed that some general conclusions may be drawn from the data.

Attempts were made to correlate the data in two ways. The first, Fig. 5.1, is a plot of the estimated 1977 capital costs in dollars/KW vs. turbine rated output in KW. A meaningful comparison between the points is difficult because of the great variety of differing designs. Two trends, however, can be noticed. The first is that, generally, the costs per KW decrease with increased rated output. Secondly, there appears to be 3 "clusters" of points on the graph. The upper left hand corner generally contains the "farm" type of machine. The center group generally contains the smaller commercially available horizontal and vertical axis turbines, and the right hand group encompasses the larger horizontal axis turbines and the older projects. The first two groups contain many machines which have been produced in quantity, while the larger machines have not. Thus, the larger "experimental" machines' costs are expected to be reduced if large quantities are produced. It may also be concluded, as noted by Quinn,^{5.2} that considerable price scatter exists, which is typical of infant industries.

The second graph, Fig. 5.2, is more interesting from the standpoint of correlation. The overall efficiencies of the systems, η , were plotted against initial capital cost. The efficiency was defined

to be the ratio between the power available from the wind and the power measured at the output terminals of the system. This latter figure was taken from the manufacturer specifications whenever possible. In the case of farm machines, no commercial figures are available for electricity generated; the output power was calculated from the pumping ability of the machine with a pump having a 0.05 m (2 in) diameter cylinder and operating at the rated wind speed given by the manufacturer. The selection of the cylinder size was arbitrary, but calculations showed that the effect of considering other sizes on the efficiency was negligible.

The plot (Fig. 5.2) shows several interesting trends, the first being the separation of the points by rated wind speed. The graph breaks up into 2 distinct regions; the first of 7-9 m/s, and the second of 10^+ m/s. Two exceptions are the NASA mod 0 machine, at a rated speed of 8 m/s, and the Kedco 1600, rated at 7 m/s. These machines have fairly high efficiencies, although designed for relatively low rated speeds. It can be seen that the lowest capital costs are attained by operating at higher rated speeds and efficiencies. A feature worthy of note is the close proximity of the only VAWTs in the survey (DAF 15 and 20) to the better horizontal axis machines. While not conclusive, this comparison gives hope that VAWTs may be developed to be competitive with the more established designs.

Conclusions

Within the framework of the limited analysis of wind turbine costs which was undertaken, several conclusions may be drawn.

Initial cost per KW output is seen to decrease with increasing turbine size and efficiency. Much refinement of the costs of some of the newer models (e.g., the NASA Mod 0 family) is underway, and the operating costs of these and other new machines must yet be determined.

The initial commercially available vertical axis turbines appear to be cost competitive with the horizontal axis designs. The cost of the WVU VAWT test model (see section 4 of this report) would produce values of about \$2000/KW at 2.3 KW output. With the expected cost reduction of a developed production machine, and possible output gains with circulation control blades, this cost seems competitive when compared with the turbines surveyed (Fig. 5.1).

Continuing cost analysis is necessary to give an accurate indication of both initial and operating WECS costs.

References

- 5.1. Walters, R. E., et al, "Innovative Wind Machines: Executive Summary and Final Report," ERDA/NSF/00367-76/2, Department of Aerospace Engineering, West Virginia University, June, 1976.
- 5.2. Quinn, B., "The Consumer's Cost of Electricity from Windmills." Eleventh Intersociety Energy Conversion Engineering Conference, State Line, NV, September 12-17, 1976, Vol. II, pp. 1746-1753.

TABLE 5-I
CHARACTERISTICS OF SOME WIND TURBINES

ID#	MANUFACTURER/MODEL		DIAMETER (FT)	RATED ¹ SPEED (MPH)		POWER IN (KW)	POWER ¹ OUT (KW)	EFFICIENCY	CAPITAL COST (\$/KW)	NOTES	ESTIMATEE 1977 CAPITAL COST (\$/KW)
1	Amerenalt	1500	8	12	27	5.02	1.5	.299	1766	2	1996
2		2500	8	15	36	11.93	2.5	.210	1140	2	1243
3	Windworks		12	11.2	25	8.99	1.87	.207	900	3	900
4	Kedco	1200	12	9.8	22	6.11	1.2	.195	1913	4	1913
5		1600	16	7	16	4.19	1.2	.286	2413	4	2413
6		1210	12	11.2	25	8.99	2.0	.222	1298	4	1298
7		1610	16	9.8	22	10.88	2.0	.184	1598	4	1598
8	D.A.F.	15	15	10.3	23	12.43	3.2	.257	607	5	656
9		20	20	10.3	23	19.43	6.0	.308	572	5	618
10	Aeromotor	6	6	3.9	20	1.12	.08	.071	5812	5,6	6277*
11		8	8	3.9	20	2.05	.167	.084	4131	5,6	4461
12		16	16	3.9	20	8.18	.895	.105	4731	5,6	5109
13	Baker	6	6	5.7	15	.485	.075	.154	6946	5,6	7502*
14		8	8	5.7	15	.863	.157	.182	4299	5,6	4643
15		12	12	5.7	15	1.942	.408	.210	3512	5,6	3793
16	Lempster	6	6	5.7	15	.485	.078	.161	5256	5,6	5676*
17		8	8	5.7	15	.863	.165	.191	3696	5,6	3992
18		12	12	8.9	20	4.6	.394	.086	4505	5,6	4865
19	Eunlite		13	11.2	25	10.55	2.0	.190	2000	2,3	2260
20	Grumman Windstream		25	11.6	26	43.89	15	.342	1100	4,3	1100
21	Vincharger		6	10.3	23	1.75	.2	.114	2625	4,3	2625
22	Electro GMB WVG50G		16.5	13.4	30	25.372	6	.202	1333	2,3	1506
23	NASA MOD 0		125	8	18	1880	100	.275	1600	4,3	1600
24	Eutnam		175	13.4	30	4600	1250	.272	650	4,3	650
25	Cedser		78.2	13.4	30	659	200	.305	400	4,3	900

NOTES: Costs are for turbine and generator only.

1. Manufacturer's specification, except for #s 10-18, where power out was calculated.
2. 1975 price
3. Estimated
4. 1977 price
5. 1976 price
6. Generator not included
7. Based on 8% increase from 1976-77 and 13% increase from 1975-77.

* Not on KW/\$/KW curve, Fig. 5.1.

Fig. 5.1. Initial cost per unit output vs. turbine capacity of wind turbines.

Fig. 5.2. Initial cost per unit output vs. turbine output efficiency.

6. CONCLUSIONS

A comprehensive analysis of straight-bladed vertical axis wind turbines (VAWTs) is underway. A numerical solution of the blade aerodynamics has been extended to investigate multi-bladed turbines. Initial results indicate that turbine performance is strongly dependent upon blade drag characteristics. A test model VAWT has been designed, constructed and prepared for outdoor testing. The cost of this test model has been determined in detail during construction, and the cost was found to be of the same order as that for horizontal axis wind turbines of the same output performance. Initial structural and cost analysis of a VAWT design using composite materials for blade fabrication has been accomplished. Results of this analysis have indicated that composite material blade construction may prove to be of lower cost than conventional (aluminum) blade construction. In order to attain operating conditions of higher tip speed ratios, the composite blades may be the required construction method.

7. RECOMMENDATIONS

Based on results obtained to date, it is recommended that the analysis of straight-bladed vertical axis wind turbines be continued. Specifically, more information is required on the theoretical aerodynamic analysis allowing performance predictions. Test data must be obtained to provide blade aerodynamic data for the theory, and to verify the predicted performance. Cost analysis and structural analysis should continue to better determine the competitiveness of the straight-bladed vertical axis wind turbine.

APPENDIX A

SYMBOL DEFINITION FOR BLADE STRUCTURAL ANALYSIS, SECTION 3

A	cross sectional area of blade, in
A_C	capture area of the rotor, ft
C	chord length of the blade, in
C_D	drag coefficient, dimensionless
C_L	lift coefficient, dimensionless
C_{F_X}	nondimensional force in X direction
C_{F_Y}	nondimensional force in Y direction
D	drag, lb
e	distance between neutral axis and outside fibers of blade, in
E	modulus of elasticity, psi
F_X	force in the X direction, lb
F_Y	force in the Y direction, lb
I_X	moment of inertia about X axis, in ⁴
L	lift, lb
l	blade half length, in
M	moment, in-lb
m	mass, slugs
R	radius of turbine, ft
R_a	center reaction, lb
S	axial component of pretension load, lb
V_∞	wind velocity, ft/sec

V_{rel}	velocity relative to blade, dimensionless
W	uniform load on the blade, lb/in
α	angle between V_{rel} and blade chord (total angle of attack), degrees
α_1	angle between relative velocity and radial axis of the machine, degrees
α_0	angle of attack between blade setting and the radial axis of the wind machine, degrees
Γ	torque, ft-lb
θ	angle between radial axis of machine and free stream, degrees
ρ	density, slugs/ft ³
σ_b	bending stress, psi
ω	angular velocity, rad/sec

APPENDIX B

COMPUTER PROGRAM FOR UNIFORM BLADE FORCE CALCULATIONS, SECTION 3

```
//ENGRAERO JOB (AS09332.C,9177,10),'ELKO'
//EXEC FORGO
//SOURCE DD *
    NAMELIST/DATIN/IDAL,A,B,RC,RVI,R
    .DATA DR/.017453/
10  .READ(5,DATIN)
    M=1.0
    G=32.2
    RHO=.002378
    C=R/RC
    CENT=(M/G)*WRVI**2)/(.5*RHO*C*R)
    WRITE(6,DATIN)
    WRITE(6,444)
444  FORMAT(///,' THETA      ',' ALFA1      ',' ALFA) ',' ALFA ',
.A ' CD ',' CL ',' VREL ',' A BLADE X ',' C BLADE X ',
B ' T BLADE X ')
    IALFA0=90+IDAL
24  .DO 10 ITHETA=5,355,10
    IF(ITHETA.GT.90.AND.ITHETA.LT.30) IALFA0=90-.IAL
    COST=COS(DR*ITHETA)
    SINT=SIN(DR*ITHETA)
    VREL2=1.-WRVI*(2.*SINT-WRVI)
    ALFA1=ATAN2(SINT-WRVI,COST)
    IF(ALFA1.LT.0.0)ALFA1=ALFA1+6.283
    .COSA=COS(ALFA1)
    SINA=SIN(ALFA1)
    ALFA=ALFA1+FLOAT(IALFA0)*DR
    .COSB=COS(ALFA)
    SINB=SIN(ALFA)
    .IF(ALFA.GT.6.283)ALA=ALFA-6.283
    IF(ALFA.GT.3.14159)ALFA=ALFA-6.283
    IALFA1=ALFA1/DR+.5
    CL=.1*ALFA/DR
    CD=A+B*CL**2
```


APPENDIX B (CONTINUED)

```

      IF (ABS(ALFA).GT..1607) CL=SIN(2*(ALFA-48dr00
      IF (ABS(ALFA).GT..1607.AND.ABS(ALFA).;..8727) CD=SIN(ALFA)
      IF (ABS(ALFA).GT..8727.AND.ABS(ALFA).lt.2.2689)
ACD=1.8*SIN(ALFA)-.15
      .IF (ABS(ALFA).GT.2.2689)CD=SIN(ALFA)
      LATA=ALFA/DR+ .5
      VREL=SQRT(VREL2)
      COSC=COS(IALFAO*DR)
      .SINC=SIN(IALFAO*DR)
      CFXA=VREL2*(-CL*SINB+CD*COSB)
      CFXC=-CENT*COSC
      CFXT=CFXA+CFXC
102  FORMAT(1H , 14,3(6X,14),3(2X,F.3),3(4X,F6.2))
      .WRITE(6,102) ITHETA,IAFA1,IALFAO,IATA,
A CD,CL,VREL,CFXA,CFXC,CFXT
100  CONTINUE
      WRITE(6,445)
445  FORMAT(//,' THETA ', ' A BLADE Y ', ' C BLADE Y ',
A ' T BLADE Y ', ' A MACH R ', ' C MACH R ', ' T MACH R ',
B ' A MACH T ')
      DO 200 ITHETA=5,355,10
      .IALFAO=90-IDAL
      IF (ITHETA.GT.90.AND. ITHETA.LT.270) IALFAO=90-IDAL
      COST=COS(DR*ITHETA)
      SINT=SIN(DR*ITHETA)
      VREL2=1.-WRVI*(2.*SINT-WRVI)
      ALFA1=ATAN2(SINT-WRVI,COST)
      IF (ALFA1.LT.0.0)ALFA1=ALFA1+6.283
      COSA=COS(ALFA1)
      SINA=SIN(ALFA1)
      .ALFA=ALFA1+FLOAT(IALFAO)*R
      COSB=COS(ALFA)
      SINB=SIN(ALFA)
      IF (ALFA.GT.6.283)ALFA=ALFA-6.283

```

APPENDIX B (CONTINUED)

```

      IF(ALFA.GT.3.14159)ALFA=ALFA-6.283
      IALFA1=ALFA1/DR+.5
      CL=.1*ALFA/DR
      CD=A+B*CL**2
      .IF (ABS(ALFA).GT..1607)CL=SIN(2*(ALFA-4*DR))
      IF (ABS(ALFA).GT..1607.AND.ABS.(ALFA).T..8727) CD=SIN(ALFA)
      .IF (ABS(ALFA).GT..8727.AN.ABS.(ALFA).LT.2.2689)
      .ACD+1.8*SIN(ALFA)-.15
      IF (ABS(ALFA).GT.2.2689) CD=SIN(ALFA)
      LATA=ALFA/DR+.5
      VREL=SQRT(VREL2)
      COSC=COS(IALFA0*DR)
      SINC=SIN(IALFA0*DR)
      CFYA=VREL2*(CL*COSB+CD*SINB)
      CFYC=-CENT*SINC
      CFYT=CFYA+CFYC
      .CW2RA=VREL2.*CL*SINA-CD*COSA)
      .CW2RC=CENT
      CW2RT=CW2RA+CW2RC
      .CWRA=VREL2*(CL*COSA+CD*SINA)
103  FORMAT(1H ,14,7(7X,F7.2))
      WRITE(6,103)  ITHETA ,CFYA,CFYC,CFYT,CW2RA,W2RC,CW2RT,CWRA
200  CONTINUE
      GO TO 10
      END

```

APPENDIX C

COMPUTER PROGRAM FOR BLADE MOMENTS, STRESSES, AND DEFLECTIONS, SECTION 3

```

10.      GET LIST(w,s,e,i,l);
15.      PUT IMAGE('')(f1);
16.      f1:  IMAGE;
X  SLOPE  DEFLECTION  MOMENT  STRESS
20.      reed:  DO x=0 TO 64 BY 1;
25.          ya=.53;
26.          a=1.18;
30.          k2=s/(e*i);
40.          k=sqrt(k2);
50.          ce2=(exp(2*1*k)+exp(-2*1*k))/2;
60.          se2=(exp(2*1*k)-exp(-2*1*k))/2;
70.          cel=(exp(1*k)+exp(-1*k))/2;
71.          sel=(exp(k*1)-exp(-k*1))/2;
71.1      gig=2*w*(se2/cel-se2)/k+w*1*1*k*se2;
71.2      trip=2*sel*sel-1*k*se2;
71.3      r=gig/trip;
71.4      A=r*sel/(s*k*se2);
140.      cex=(exp(k*x)+exp(-k*x))/2;
150.      sex=(exp(k*x)-exp(-k*x))/2;
155.      dy=-A*k*cex+w/(s*k2)*(-k*(sel*cex-cel*sex)/cel)+w/s*(1-x)+r/(2*s);
162.      y=-A*sex+w/(s*k2)*((cel*cex-sel*sex)/cel-1)+r*x/(2*s)+w/(2*s)*x*(2*1-x);
162.5     m=-s*A*sex+w/k**4*(k2*(cel*cex-sel*sex)/cel)-w/k2;
164.      stress=m*ya/i+s/a;
170.      PUT IMAGE(x,dy,y,m,stress)(f2);
171.      f2:  IMAGE;
-----
180.      END reed;
180.5     PUT LIST('');
181.      PUT LIST('CENTER REACTION=',r);
190.      END ;

```

APPENDIX D

SYMBOL DEFINITION FOR BLADE WEIGHT ANALYSIS EQUATIONS, SECTION 4

The symbols used in the blade weight (section 4) equations are defined as follows (also see Fig. 4.3):

C	blade chord, m (ft)
E_C	modulus of elasticity of spar, P_a (lb/ft ²)
E_s	modulus of elasticity of skin, P_a (lb/ft ²)
F_{ay}	normalized aerodynamic load coefficient, dimensionless
F_{cy}	normalized centrifugal load coefficient, m ³ /kg (ft ³ /slug)
f_a	safety factor on aerodynamic loads, dimensionless
f_c	safety factor on centrifugal loads, dimensionless
f_s	general safety factor, dimensionless
k_B	proportionality constant, dimensionless
k_T	ratio of blade thickness to chord, dimensionless
k_W	proportionality constant, dimensionless
L	blade length, m (ft)
$m(W)$	blade mass (weight)/unit length, kg/m (lb/ft)
R	blade orbit radius, m (ft)
V_∞	upstream wind velocity, m/sec (ft/sec)
Δ_M	maximum blade deflection, m (ft)
ρ_A	mass density of air, kg/m ³ (slug/ft ³)
ρ_c	mass (weight) density of spar caps, kg/m (lb/ft)
ρ_s	mass density of skin, kg/m ³ (slugs/ft ³)
σ_C	spar maximum bending stress, P_a (lb/ft ²)
σ_s	skin maximum bending stress, P_a (lb/ft ²)

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