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Energy Research and Development Administration

Attention : Contract Administrator

Contract: E(11-1) 2614

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Summary

A review of the progress made during the first few months of the subject contract is presented herein. The objective of the one year program is to explore the feasibility of the Composite Bearingless Rotor (CBR) for use as a wind turbine and to evaluate several automatic control concepts designed to improve efficiency and cost effectiveness. The program consists of the design of a full-scale wind turbine, the design and fabrication of a dynamically scaled model, wind tunnel experiments, and analytical studies. A description of all aspects of the program is presented in Ref. 1. To date, the design work has been completed, the model wind turbine support tower has been fabricated and assembled with a generator. The model blades and other hardware are under construction. Completion of the hardware and initiation of the wind tunnel tests is expected in December 1975.

This interim progress report contains descriptions of the full scale and model designs, a review of the performance characteristics expected of the wind turbine, and an outline of the remaining tasks of the program. Appendices are included to provide detailed information on some of the design characteristics of the model wind turbine.

Technical Progress of Full-Scale Design

Much of the preliminary design information for the full-scale wind turbine was derived from the Lockheed/NASA design of Ref. 2 and from data obtained from Hamilton Standard. It was decided to use composite materials for

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the outboard portion of the blade, as well as for the inboard "flexbeam", which is required in the CBR concept. It was felt that since the trends in the helicopter industry are toward all composite blades for improved performance and lower costs, the wind turbine blade should also follow suit for the same reasons. A fiber glass blade comprising a wound elliptical hollow spar and a wound aerodynamic cover was selected. The flexbeam is carbon/epoxy as in the helicopter CBR designs. It has been shown that carbon/epoxy has improved modulus, strength, and fatigue characteristics over other composite materials for the CBR application. The blade planform, twist, and basic dimensions are given in Figs. 1-3. Pertinent design information is presented in Table I. Blade frequency information for the lower bending and torsion modes is presented in Fig. 4.

Model Design

Direct scaling of the full-scale blade structure was not possible since the large scale factor, 27.8, dictates model blade skin thicknesses less than a single-ply thickness. It was decided that the blade stiffness would be modeled using a single spar fabricated from carbon/epoxy and a nonstructural balsa airfoil. The spar consists of a tapered balsa core, serving as a mandril, and precured unidirectional carbon/epoxy sidewalls bonded directly to the balsa. This structure produces non-dimensional bending and torsion frequencies which duplicate the full-scale values. The wall thicknesses are given on Fig. 5. A NACA 00xx series airfoil is used and it varies from 35% thick inboard to 12% thick at the tip. The airfoil shape itself is shaped from lightweight balsa and the final blade assembly is wrapped with one layer of Monocote.

The inboard portion of the blade consists of the flexbeam, the pre-set pitch mechanism, the feathering mechanism, the rpm sensing weights, and the pitch-flap coupling mechanism. Drawings of these components are shown in Figs. 6 and 7.

The wind turbine support tower is shown in Fig. 8, and a close-up of the generator and other tower components are shown on Fig. 9. Also visible on this photograph is the rpm pickup, the slipring clamp, and below the generator support plate, the azimuth gear, the potentiometer, and the pneumatic cylinder for releasing the yaw lock at a preselected azimuth position. The strain gage instrumentation wires run through the hollow generator shaft to the slipring assembly. From there the wires go through the center of the yaw plate and support cylinder and down the tower to the tunnel floor. A Bosch GR 15x generator was selected which

delivers 420 Watts at 14V at about 3000 rpm. The full-scale speed of the model is about 1250 rpm. Although this rotational speed is considerably less than the design speed of the generator, for the power levels expected the generator should be an adequate torque matcher.

At the low rpm the generator has been calibrated by measuring the power input and the power output. The input power is the product of rotational speed and the torque on the generator at that speed. With this torque the rpm of the windmill can be controlled. By controlling the power output of the generator, which depends on the loading resistor of the generator, the input torque is determined.

Performance Calculation

Although the turbine's rotational speed can be controlled within limits by governing the load on the turbine, for the control mechanisms under study here, it is a pitch change policy that is looked for. It is therefore essential to analyze accurately the performance of the wind turbine. This is done with the UTRC Prescribed Wake Hover Performance Analysis which was developed to compute the blade circulation and inflow distribution and the corresponding integrated rotor performance, which is based on a prescribed wake geometry. Generality regarding the specification of the geometry was maintained in the computer program to permit the evaluation of a wide variety of wake geometries. The expanding wake of the wind turbine has been replaced by a cylindrical wake, awaiting further experimental data on wind turbine wakes. For the Prescribed Wake Analysis, each blade is represented by a segmented lifting line, and the wake is represented by a finite number of segmented vortex filaments, trailing from the blade segment boundaries. The blade and wake characteristics are assumed to be independent of azimuth position. Conventional strip theory based on two dimensional airfoil data is assumed applicable to compute the rotor performance characteristics. Taken into account are the Mach number effects on the two dimensional airfoil data and the rapidly changing characteristics with thickness ratio. The wind turbine is analogous to an unpowered rotor in vertical descent.

The program is divided into three parts. The first part transforms the wake geometry input to wake coordinates. The second part contains the computation of the wake influence coefficients at the blades, as defined by the law of Biot - Savart. It solves also the circulation matrix and the associated induced velocity distribution. In the third part, the performance characteristics are computed. A more detailed description of the program is contained in Reference 3.

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Some analytical results are shown in Figures 10 and 11. Figure 10 shows at a wind speed of 25 mph the effect of the $0.75R$ pitch angle on power output per blade versus tip speed. The rapid decline of the output with some values of tip speed and pitch angle is because of stall. Figure 11 shows at constant pitch the increase of the power output per blade with wind speed. It becomes clear that for a good power output, the tip speed needs to change with wind speed.

Plans for Remainder of Program

The remaining tasks of this program consist primarily of completing the model hardware, performing the wind tunnel tests, conducting an analysis of the test results along with correlation studies, and identifying future requirements to further evaluate the CBR concept for wind turbine applications. The specific tasks and estimated dates of completion are shown in the following diagram.

Ref. 1: Proposal P-N211

Ref. 2: "100-KW Hingeless Metal Wind Turbine Blade Design, Analysis and Fabrication", R. E. Donham J. Schmidt, AHS Annual Forum May 1975 Paper #S-998.

Ref. 3: A. J. Landgrebe: "The Wake Geometry of a Hovering Rotor and its Influence on Rotor Performance", AHS preprint #620, May 1972.

TABLE I
Design Parameters

<u>Full Scale</u>		<u>Model Scale</u>
$b = 4$	number of blades	$b = 4$
$R = 62.5'$	rotor radius	$R = 2.25'$
$V_{tip} = 150-300$	tip speed (ft/sec)	identical
$c_{tip} = 22"$	blade chord at tip	$c_{tip} = 0.79"$
$c_{root} = 55"$	blade chord at root	$c_{root} = 1.98"$
$r_{root} = 0.07R$	radial location root cut-out	identical
$r_{hub} = 0.05R$	radius, turbine hub	identical
airfoil sections: NACA 00xx		identical
tip: NACA 0012		
root: NACA 0035		

See page 6 for List of Symbols

LIST OF SYMBOLS

a	Factor
a _o	Length of half the principle axis of elliptical spar (inch)
aR	Fractional length of R (ft)
B	Width box spar (inch)
b	Factor; number of blades; thickness flex beam
b _o	Length of half the principle axis of elliptical spar (inch)
bR	Fractional length of R (ft)
cR	Fractional length of R (ft)
C.F.	Centrifugal force (lb)
c	Factor; chord of airfoil section; chord of flexbeam
d	Factor
dR	Fractional length of R (ft)
d()	Infinitesimal amount of ()
E	Modulus of elasticity in tension (psi)
G	Modulus of elasticity in shear (psi)
H	Height box spar (inch)
I _e	Edgewise area moment of inertia (inch ⁴)
I _f	Flatwise area moment of inertia (inch ⁴)
I _p	Polar mass moment of inertia (slug ft ²)
I _x	Area moment of inertia about y-axis (inch ⁴)
I _y	Area moment of inertia about x axis (inch ⁴)
J	Radial area moment of inertia (inch ⁴)
k	Coupling factor $\frac{d\theta}{d\beta}$
M	Moment (inch-lbs)
m	Mass (slug)
P	Power (HP)
R	Radius turbine (ft)
r	Radial length (ft)
$\bar{r}$	Nondimension radial length $\frac{r}{R}$
t	Flexbeam thickness (inch); airfoil thickness (inch); material thickness spar (inch)
t ₁	Material thickness spar (inch)
V _{tip}	Tip speed turbine (fps)
V _w	Freestream wind velocity (mph)
x	Axis, along airfoil chord or along principle spar; positive forward
y	Axis, perpendicular to x
β	Experimental factor for torsional rigidity; flapping angle (rad)
$\Delta()$	Finite amount of ()
δ, δ_1	Material thickness (inch)
δ_3	Angle (rad)
θ	Geometric pitch angle(rad); torsional deformation of flexbeam (rad)
θ_{75}	Pitch angle at 75 percent R
ρ	Density material (slug/ft ³)
ψ	Azimuthal angle (rad)
Ω	Turbine rotational speed (rad/sec)
ω	Frequency (rad/sec)
() _o	Initial or equilibrium state of variable

APPENDIX A:

Flexbeam Properties

The feathering feature of the turbine blade requires that the diagonal of the rectangular flex beam can rotate inside the elliptical fiberglass beam of the blade. The flexbeam extends from station 0.2R at one blade to the same station of the opposite blade and is enclosed by the hub out to 0.05R or $\bar{r} = 0.05$, if $\bar{r}$ defines the ratio $\frac{r}{R}$. The bending and twisting part of the flexbeam has the dimensions:

Bending stiffness at cross section $\bar{r}$:

$$\begin{aligned} EI_f &= \frac{1}{12} E b t^3 \\ &= 4.89 \cdot 10^8 (0.65 - \bar{r}) \\ EI_e &= \frac{1}{12} E t b^3 \\ &= 1.358 \cdot 10^{11} (0.65 - \bar{r})^3 \end{aligned}$$

where $E = 22 \times 10^6$ PSI for unidirectional carbon epoxy.

Torsional stiffness at cross section $\bar{r}$:

$$\begin{aligned} GJ &= G b t^3 \beta \\ &= G (-89.667\bar{r} + 54.563) \text{ lbs inch}^2 \end{aligned}$$

Of interest is $\frac{M}{\theta}$ of the beam as a whole. Here M is the torsional moment and θ is the rotation of the beams end. Consider section $d\bar{r}$ at $\bar{r}$, which rotates over angle $d\theta$ due to a moment.

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$$d\theta = \frac{d\ell}{GJ} M$$

or integrated:

$$\theta = \frac{MR}{G} \int_{0.05}^{0.2} \frac{dx}{J(x)}$$

Therefore

$$\frac{M}{\theta} = \frac{G}{R \int_{0.05}^{0.2} J^{-1} dx}$$

$$= 3.44 \cdot 10^5 \frac{\text{inch lbs}}{\text{rad.}}$$

APPENDIX B:

CALCULATION MODEL SPAR

At each location $\bar{r} = \frac{r}{R}$ of model turbine blade, four scaled parameters of the full-scale turbine have to be satisfied, namely:

flatwise bending	EI_f	= constant C_1
edgewise bending	EI_e	= constant C_2
torsional	$\frac{GJ}{I_p}$	= constant C_3
weight/length	w	= constant C_4

A solid bar has two dimensions; width and thickness. If the bending stiffnesses, torsional stiffness, and weight per length of the spar are equated to the above constants, then there are four equations. With two unknowns, these equations are only satisfied by chance, unless an assortment of materials is available with independently varying E and G . These materials are not known to exist. To satisfy four equations the cross section of the beam needs four characteristic measures. Of course there remains the question if a solution or a unique solution exists for four nonlinear algebraic equations.

A single sandwich construction or a spar with circular hole gives only three unknowns.

A double sandwich construction or a wrapped single sandwich gives four unknowns.

However the constant C_4 is so low in this applications, that no solution exists for known materials applied to these cross-sections.

Since the material near the center of the beam contributes little to the constants C_1, C_2 , and C_3 but contributes heavily to C_4 , a hollow spar could lead to a solution. Consider the cross section:

Evaluation of the four properties of the cross section:

$$E \left[\frac{t H^3}{6} + \frac{(B-2t) t_1^3}{6} + \frac{t_1 (H-t_1)^2 (B-2t)}{2} \right] = C_1$$

$$E \left[\frac{t_1 (B-2t)^3}{6} + \frac{H t^3}{6} + \frac{(B-t)^2 t H}{2} \right] = C_2$$

$$\frac{G}{\rho(I_x + I_y)} \frac{2 t t_1 (B-t)^2 (H-t_1)^2}{t (B-t) + t_1 (H-t_1)} = C_3$$

$$2\rho \left[Ht + (B-2t)t_1 \right] = C_4$$

For a given material with E , G , and ρ , the four unknowns H , B , t , and t_1 , can be calculated. Two computer programs have been written to automate the calculus of the four unknowns. Neither the Newton Raphson nor the least square error method converged except for initial guesses extremely close to the solution. The very strong nonlinearities in the left hand side of the equations cause divergence in most cases. The logic necessary for convergence was easiest obtained with persistent hand calculus. These solutions were presented in the main body of the report, Fig. 5.

APPENDIX C:

Derivation of RPM - BLADE PITCH Mechanism

Consider a mass m , in a centrifugal field as shown in the sketch.

Although the mechanism does not need to be specified here, moment M takes care of the equilibrium.

Suppose the mass m is spaced azimuthally over angle ψ with the torsional axis of the flex beam.

The moment of the mass about the torsional axis:

$$\begin{aligned} M_{\theta} &= -C.F. \times r \sin\psi \sin\beta \\ &= -\frac{1}{2}m(\Omega r)^2 \sin\psi \sin 2\beta \end{aligned}$$

At this point it is not relevant how this moment is transferred to the end of the flexbeam. Many feasible mechanisms are possible. Suppose the blade of the turbine needs to be pitched through angle θ .

$$M_{\text{beam}} = \left(\frac{M}{\theta} \right)_{\text{beam}} \theta$$

with

$$\left(\frac{M}{\theta} \right)_{\text{beam}} = 3.44 \times 10^5 \frac{\text{inch Lbs}}{\text{radian}}$$

This moment in the flexbeam provides the equilibrium for the moment produced by mass m . Moment equilibrium: $-\frac{1}{2}m(\Omega_0 r)^2 \sin\psi \sin 2\beta_0 - 3.44 \times 10^5 \theta_0 = 0$

Within the range of operation of the mechanism, it is assumed that ψ is constant. The perturbed moment equation becomes:

$$-\frac{1}{2}m \left[(\Omega_0 + \Delta\Omega)r \right]^2 \sin\psi \sin 2(\beta_0 + \Delta\beta) - \left(\frac{M}{\theta} \right)_{\text{beam}} (\theta_0 + \Delta\theta) = 0$$

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Because of the mechanical, yet unspecified connection between $\Delta\beta$ and $\Delta\theta$, it is assumed that:

$$\frac{d\theta}{d\beta} = k$$

substitute this in above equation:

$$-\frac{1}{2} m [(\Omega_0 + \Delta\Omega) r]^2 \sin \psi \sin 2(\beta_0 + \frac{\Delta\theta}{k}) - \left(\frac{M}{\theta}\right)_{\text{beam}} (\theta_0 + \Delta\theta) = 0$$

This establishes the relationship of $\Delta\Omega$ and $\Delta\theta$ at any operating point. It is assumed here that the gravitational and aerodynamic forces do not enter the moment equation. For simplicity only one mass m is considered. If another mass m is mounted at $-\beta$ and $-\psi$, those forces will generally cancel. Linearizing and neglecting second order terms gives:

$$\frac{\Delta\theta}{\Delta\Omega} = \frac{-mr^2 \Omega_0 \sin \psi \sin 2\beta_0}{\left(\frac{M}{\theta}\right)_{\text{beam}} + mr^2 \Omega_0^2 \sin \psi \frac{\cos^2 \beta_0}{k}}$$

k follows from geometry:

By good approximation:

$$L_2 d\theta = L_1 d\beta$$

$$k = \frac{d\theta}{d\beta} = \frac{L_1}{L_2}$$

Where the sides are given by

$$L_1 = \frac{\bar{r}}{\cos \psi \cos \beta} \quad L_2 = \frac{\bar{r}}{\cos \psi \sin \beta} \sqrt{1 + \left(\frac{\sin \beta}{\tan \psi}\right)^2}$$

Elimination of k yields:

$$\frac{d\theta}{d\Omega} = \frac{-m \Omega \bar{r}^2 \frac{2 \tan \psi \tan \beta}{\cos \psi}}{\left(\frac{M}{\theta}\right)_{\text{beam}} + m (\Omega \bar{r})^2 \frac{\sqrt{\tan^2 \psi + \sin^2 \beta}}{\tan \beta \cos \psi}}$$

Note that the above formula holds for one mass m . Use of two diagonal masses about the elastic axis of the flex beam, doubles the coupling term $\frac{d\theta}{d\Omega}$.

APPENDIX D:

PITCH-FLAP MECHANISM

The function of the blade pitch-flap mechanism is identical to the δ_3 hinge in rotary wing aircraft and it is intended to minimize the effects of gusts. In articulated helicopter blades there is a distinct hinge. However for hingeless blades like the wind turbine, there is no distinct hinge.

Any bending in the beam can be approximated by two rigid beam parts, which hinge about a representative point, called the virtual hinge point. For a nonrotating beam with a free end the virtual hinge point could be in the middle of the beam. However for a rotating beam the virtual hinge point generally lies within 10% of the point of attachment and moves inward as RPM is increased (as the fundamental first mode flapping frequency approaches 1P). The δ_3 effect is determined by a second hinge point at the connection of the pitch arm to the hub, Fig. 7. The line through the virtual hinge point and the second hinge point determines the sign and the magnitude of the flap pitch coupling.

Schematically Figure 7 is equivalent to:

From geometry follows: $\tan \delta_3 = \frac{a-b}{c}$

The relation between flapping and pitch follows from considering an arbitrary section at dR :

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If the blade flaps $d\beta$ degrees, then A goes up $(d-a) R \sin d\beta$.
 At the same time point B remains at the virtual hinge line. The airfoil section at radial location dR sees therefore a pitching of line AB of:

$$\begin{aligned} \sin (-d\theta) &= (d-a) \frac{R \sin d\beta}{AB} \\ &= (d-a) \frac{R \sin d\beta}{(d-a) R / \tan \delta_3} \\ &= \tan \delta_3 \sin d\beta \end{aligned}$$

With a small angle assumption:

$$\frac{d\theta}{d\beta} = -\tan \delta_3$$

Or approximately:

$$\frac{d\theta}{d\beta} = \frac{b-a}{c}$$

This relation is independent of the radial station of the airfoil section under consideration. The factor a is dependent on rpm. Factors b and c are determined by geometry.

FIGURE 1: PLANFORM OF WIND TURBINE BLADE

FIGURE 2: GEOMETRIC TWIST OF TURBINE BLADE

FIGURE 3: DIMENSIONS OF BLADE CROSS-SECTIONS

FIGURE 4: FLATWISE, EDGEWISE AND TORSIONAL FREQUENCIES VERSUS TIP SPEED

FIGURE 5: DIMENSIONS OF MODEL SPAR CROSS-SECTIONS

FIGURE 6: MODEL TURBINE BLADE ROOT DETAIL

FIGURE 7: TURBINE MODEL HUB AND BLADE ROOT DETAILS

FIGURE 8: SUPPORT OF WIND TURBINE SHOWN IN WIND TUNNEL

FIGURE 9: DETAIL OF INSTRUMENT ARRANGEMENT

FIGURE 10: CONSTANT WIND VELOCITY POWER OUTPUT PER BLADE

FIGURE 11: EFFECT OF WIND VELOCITY ON POWER OUTPUT