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SELF-REGULATING COMPOSITE BEARINGLESS WIND TURBINE

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Division of Solar Energy

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Self-regulating Composite
Bearingless Wind Turbine

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Bearingless Wind Turbine

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Self-regulating Composite
Bearingless Wind Turbine

ABSTRACT

The Composite Bearingless Rotor (CBR) concept has been shown to have characteristics ideally suited for wind turbine applications. Originally developed for helicopters to reduce weight, costs, and complexity, the CBR eliminates blade bearings and hinges through the utilization of the unique structural characteristics of uniaxial composite materials. This rotor concept was further developed under an ERDA contract to provide a fully self-regulating and self-aligning wind turbine. Such a system was achieved without the need for auxiliary controls or sensors. These features allow self-starting for wind initiating from any direction and automatic pitch and yaw variations to optimize performance under all normal wind conditions. The work described in this report consists of the design of a 4.5 ft dynamically scaled wind turbine model and the testing of this model in the United Technologies low speed wind tunnel. Several concepts were investigated which were designed to achieve self-regulation. Of these, a system which consisted of a hub supported pendulum integrated with the CBR blade provided the features desired. Testing of this configuration consisted of start ups from any wind direction with wind speeds up to 30 mph. Stress and stability characteristics were investigated during the test program and the CBR wind turbine demonstrated low stress levels and highly stable response characteristics under all conditions tested.

INTRODUCTION

Much of the work in recent years in the field of wind energy conversion systems has been related to investigations of total system concepts and the general energy producing capability of rotors designed around current state-of-the-art aerodynamics. Many of these studies are reviewed in Refs. 1 and 2. The general theme which one obtains from these reports is that wind energy systems must be developed, installed, and operated at costs that are competitive with alternate forms of energy conversion. To achieve this, all high cost aspects of such systems must be identified and dealt with as efficiently as possible before proceeding to full-scale development. Significant progress can be made in this regard in the area of the principle component of a wind energy conversion system--the rotor itself.

A major activity in rotary wing research in recent years has been directed at the task of reducing costs. Progress has been made to achieve lower costs through the application of new rotor concepts, improved manufacturing and construction techniques, and the use of advanced composite materials. The experience gained in these areas can now be applied to other rotor or propeller systems, such as wind energy conversion systems, to achieve improvements in life cycle costs over that which would be expected for systems based on technology of a few years ago. In addition to cost reductions available through general advancements in rotary wing state-of-the-art, potential improvements specifically related to wind energy conversion systems become apparent. These are related to the control of such systems to maximize efficiency over the full range of operating conditions, which requires control of rotor axis alignment, rotational speed, and blade pitch. Further controls are desirable to guard against blade or tower failures resulting from gusts and excessive wind velocities. The objective of this study is to show that most of these controls can be achieved automatically (aerodynamically or mechanically) without the need for exotic sensing mechanisms or energy absorbing subsystems which would increase costs and reduce reliability. The specific rotor system studied is the Composite Bearingless Rotor (CBR) which has been under development at the United Technologies Research Center (UTRC) and the Sikorsky Aircraft Division for eight years. Full-scale main and tail rotors have been fabricated and tested and 30 percent reductions in weight have been achieved and cost reductions up to 30 percent have been projected for production designs. The Sikorsky UTTAS aircraft employs a CBR tail rotor and experience to date indicates a 50 percent reduction in maintenance over conventional tail rotors. The Composite Bearingless Rotor particularly lends itself to innovative concepts in automatic control by virtue of its unique aeroelastic characteristics. The objective of this research program was to experimentally examine the CBR as a wind turbine and several mechanisms designed to achieve a self-starting and self-regulating system.

Much of the investigation required under the contract consisted of tests of several configurations to achieve self-regulation. In some cases these were only of secondary importance, and therefore their results will be presented only as an appendix to this report. Additional detailed information concerning the test equipment design and fabrication also appear only in the appendices.

BACKGROUND

The self-regulating wind turbine described in this paper was made possible by employing the specialized features of the Composite Bearingless Rotor (CBR), developed initially for helicopter applications. It is important to be familiar with the basic CBR concept before discussing in detail its application to the wind turbine. An in-depth review of the concept is presented in the final contract report (Ref. 3) for the CBR feasibility study conducted at UTRC from 1971 to 1976, and a summary of the concept description is presented below. Additional background information presented here reviews the need for variable pitch and variable speed in a wind turbine if near optimum power is to be generated under variable wind conditions.

Composite Bearingless Rotor Description

The Composite Bearingless Rotor evolved as a result of the unique application of fiber reinforced composite materials. The nonisotropic nature of unidirectional composites, generally considered an undesirable feature, yields a structure which is decidedly softer in torsion than in a direction which loads the fibers axially. By employing a finite length of this material torsional deflections can be achieved to satisfy the pitch angle change requirements of a helicopter rotor or wind turbine while providing sufficient stiffness in the inplane and out-of-plane directions to satisfy hingeless rotor frequency requirements.

A schematic of the basic components of the CBR are shown in Fig. 1. The blade spars consist of unidirectional composite material as shown in Fig. 1a, where for a 4-bladed rotor, two single lengths are used, one for each set of opposing blades. The 4-bladed system simply consists of intersecting the two spars as shown in the figure. The dimensions of the spar are dictated by the edgewise and flatwise frequency requirements and the required torsional properties over the inboard, or flexbeam, portion of the blade. The aerodynamic cover is assembled from approximately the 20 percent radial position to the tip. This cover would normally be constructed of composite material with 45 deg fiber alignment to give maximum torsional stiffness. The cover is then bonded to the spar through a foam filler or honeycomb resulting in a torsionally stiff blade over the outer 80 percent as shown in Fig. 1b. The inner 20 percent (less the hub offset), the flexbeam, remains soft in torsion, and it is this portion of the blade that is elastically twisted to achieve pitch controls.

Figure 1: Composite Bearingless Rotor

The remainder of the rotor assembly consists of the control torque tube which is cantilevered inboard from the 20 percent station, Fig. 1c, and the two plates which make up the hub, Fig. 1d. Because of the carry-through feature of the blades, minimal hub retention hardware is required resulting in an extremely simple hub design. For the self-regulating wind turbine, control inputs to the torque tube are not required and therefore this portion would serve only as an aerodynamic surface.

Although the sketches of Fig. 1 show the spar extending to the tip of the blades, an alternate design could be used where the outer end of the flexbeam is attached to a blade of any type construction through a simple connector.

Variable Pitch/RPM Requirement

In order to self-start, the wind turbine must have blade pitch angles sufficiently low to prevent total section stall which would not provide the accelerating torque to bring the rpm to an operating level. The blades should thus be set in the static condition to the least negative (positive direction is leading edge away from wind) angle possible which will permit self-starting. This angle can vary depending on the blade twist and low speed Reynold's number but will normally be about -30 deg. As wind speed increases from zero, the turbine begins to

accelerate as a result of the lift produced on the blade as shown schematically (and ignoring induced velocity effects) by a typical cross section on the right side of Fig. 2.

Figure 2: Variable Pitch Requirement

As the rpm increases the blade section angle of attack increases from -30° as a result of the tangential velocity caused by the blade rotation. A more favorable angle of attack would result which could increase the C_L to near its maximum of about 1.2. Thus a greater accelerating force is produced and the rpm continues to increase. However, as rpm increases further, the angle of attack would tend to drop producing a lower section L/D which in turn would produce a lower rpm. This condition would represent a stable condition, however the rpm would be quite low and the power output would be well below optimum. This point on a typical performance plot can be found in Fig. 3 to fall below 10 percent efficiency whereas the optimum occurs at about 45 percent and is achieved at blade pitch angles of about 0° . This operating condition is depicted on the left side of Fig. 2. Thus, to achieve self-starting and also good efficiency at design operation, a pitch variation of approximately 30° is required. Figure 3 also demonstrates the requirement for variable rpm if optimum performance is expected. It is seen that the power ratio (efficiency) is a function of advance ratio, $\Omega R/V_W$, rather than ΩR , or rpm, alone. Therefore, with variations in wind velocity, rpm must also vary to maintain optimum power generation.

Figure 3: Wind Turbine Performance

FULL SCALE CHARACTERISTICS

The ultimate objective of this research program was to establish the feasibility of a self-regulating wind turbine employing the Composite Bearingless Rotor concept. The procedure taken to most convincingly establish this feasibility, within reasonable costs, is to test a scaled model which simulates, as nearly as possible, a full-scale system. In the ERDA study described in this report, this approach was taken wherein techniques developed for modeling helicopter rotors were used to simulate a full-scale wind turbine. The following paragraphs describe briefly the full-scale rotor and the design and fabrication details of the model.

Full Scale Blade Design

Much of the preliminary design information for the blades of the full-scale wind turbine was derived from the Lockheed/NASA design of Ref. 4 and from data obtained from the Hamilton Standard Division of United Technologies Corporation. The structural characteristics of a fiberglass blade comprising a wound elliptical hollow spar and a wound aerodynamic cover were selected. In the CBR wind turbine design the inboard aerodynamic cover is structurally distinct from the primary structural member, the flexbeam, which is located within it. The flexbeam is uni-directional carbon/epoxy as in the helicopter CBR designs, however other materials such as Kevlar, Boron or fiberglass could also be considered. Pertinent design information for the full-scale as well as the model rotor is presented in Table I.

TABLE I
DESIGN PARAMETERS

Full Scale		Model Scale
$b = 4$	number of blades	$b = 4$
$R = 62.5'$	rotor radius	$R = 2.25'$
$\Omega R = 150-300$	tip speed (ft/sec)	identical
$c_{tip} = 22'$	blade chord at tip	$c_{tip} = 0.79''$
$c_{root} = 55''$	blade chord at root	$c_{root} = 1.98''$
$r_{hub} = 0.05R$	radius, turbine hub	identical
$t = 2''$	flexbeam thickness	$t = 0.072''$
$b_{0.05} = 20''$	flexbeam width at hub	$b_{0.05} = 0.72''$
$b_{0.2} = 15''$	flexbeam width at blade	$b_{0.2} = 0.54''$
NACA OOX	airfoil sections	identical
	tip: NACA 0012	
	root: NACA 0035	

The full-scale dimensions are shown graphically in Fig. 4.

Figure 4: Dimensions of Full Scale Blade Cross-Sections

Full-Scale Performance Calculation

Although the turbine's rotational speed can be controlled within limits by governing the load on the turbine, for the control mechanism under study here, a pitch change schedule with rpm was desired. To determine the optimum blade pitch schedule, one must first calculate the performance of the wind turbine for various operating conditions. This was 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 (Ref. 5). The expanding wake expected of a wind turbine was replaced by a cylindrical wake, since experimental wake data for wind turbines were not available. 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. Mach number effects on the two-dimensional airfoil data and the rapidly changing characteristics with thickness ratio are accounted for in the analysis.

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 and solves the circulation matrix and the associated induced velocity distributions. In the third part, the performance characteristics are computed. A more detailed description of the program is contained in Ref. 5. Some of the analytical results

Figure 5: Full-Scale Performance, $V_W = 25$ mph

for the full-scale wind turbine are shown in Figs. 5 and 6. Figure 5 shows, at a wind speed of 25 mph, the effect of the $0.75R$ pitch angles on power output per blade versus tip speed. The rapid decline of the output with the higher pitch angles and the lower values of tip speed is caused by stall. For the $+6^\circ$ pitch angle there is no equilibrium at the highest tip speeds. Figure 6 shows at constant pitch the increase in power output per blade with wind speeds. From Fig. 6 it follows that for a given pitch angle, the optimum power at each wind speed requires variable tip speeds. It can also be seen in Fig. 6 that the locus of optimum power points generally coincides with a constant advance ratio curve. It thus becomes clear that for optimum power output, the tip speed must change with wind speed. It is also clear that the maximum power output occurs at relatively low or flat blade angles, between $\pm 2^\circ$.

Figure 6: Full-Scale Performance, $\theta_{75} = 0^\circ$

MODEL DESIGN

Blade and Tower Description

In scaling from the full scale diameter of 125 ft down to the model diameter of 4.5 ft, it is not possible to scale the full-scale blade structure since the

large scale factor, 27.8, would require model blade skin thicknesses less than the one-ply thickness of fiberglass. 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 with cured unidirectional carbon/epoxy sidewalls bonded directly to the balsa. A NACA OOOX series airfoil is used, which varies from 35 percent thick inboard to 12 percent thick at the tip. A segmented linear geometric twist was used to approximate the full-scale twist. The airfoil profile itself is shaped from lightweight balsa and the final blade assembly is wrapped with one layer of 0.002 in. Monocote. The calculated blade uncoupled natural frequencies are given in Fig. 7. The first torsional

Figure 7: Flatwise, Edgewise and Torsional Frequencies

frequency is presented for two conditions. The low frequency corresponds to an unrestrained flexbeam condition, that is, with no pitch control restraint, and the high frequency includes the restraint provided by a torque tube fixed at the hub and at the outer end of the flexbeam. The wind turbine model with its open truss support tower is shown in Fig. 8. In this photograph the model is shown with torque tubes installed for testing in the fixed pitch configuration. Data were obtained for this configuration to provide baseline data with which to compare the data obtained for the self-regulating configuration. The tower was fabricated using 1/2-in. steel tubing for the primary corner members and 1/4-in. tubing for the cross struts. A Bosch automobile generator was used which was mounted on a low friction yaw table which had provision for remote locking or unlocking.

Further details of the model design and test equipment are presented in the appendices.

Figure 8: Perspective View of Model Wind Turbine and Tower

Instrumentation

The instrumentation of the model consisted of edgewise, flatwise, and torsional strain gages, a yaw indicator, tower accelerometers, a tachometer with a one-per-rev indicator, and a 1 ohm rheostat to vary the output load of the rotor. The strain gage instrumentation wires passed through the hollow generator shaft, through the center of the yaw table and support tower to the tunnel floor. The data were recorded on an oscillograph and the rpm was displayed on a digital voltmeter.

EXPERIMENTAL PROGRAM

Test Procedures

Several control configurations were investigated during the wind tunnel program. These included baseline fixed pitch cases as well as configurations having automatic type controls. During the running of the fixed pitch cases it was found that for low

absolute pitch settings, low rpm equilibria were reached which resulted from stalled flow on the blades. If remote pitch control had been available this problem could have been overcome by reducing pitch to achieve unstalled flow and then increasing pitch slowly as rpm increased. To overcome the problem the generator was operated as a motor and the rotor was momentarily powered to achieve higher rpm and unstalled flow. As the unstalled equilibrium point was approached the power input was removed. The problem of stalled flow is aggravated in models of this size since the stall angles are significantly reduced as a result of the low Reynolds number. In cases where the self-starting mechanism was used, power was not required to achieve the high speed equilibrium since the blade angles operated below stall by virtue of the self-regulating feature.

The experimental program included tests of the self-aligning feature of the CBR wind turbine. In these tests it was necessary to preset the yaw position and then to release to simulate wind direction changes. A remotely controlled locking mechanism was provided for this purpose, and a 1/8-in. diameter welding rod extending from the generator to a hand operated pulley on the ceiling was used to remotely position the model before locking and releasing. The yaw control device had essentially no friction or inertia and thus, when free, had negligible effect on yaw response.

Model Tests

The wind turbine model is Mach scaled, so that full-scale tip speeds and wind speeds were retained. However, Reynolds number is reduced by the scale factor, which results in Reynolds numbers below the critical level of 10^5 . The effect is a reduction in blade section $C_{l_{max}}$ capability, which significantly lowers the start-up torque. The full-scale blade could be expected to operate unstalled up to angles of attack of 14 deg whereas the model blade stalls at about 6 deg. Since wind turbines generally operate most efficiently when the blade sections are operating at their maximum L/D's, and since this occurs at angles of attack of about 8 deg for high Reynolds number, low subsonic airfoils, the model blade is incapable of achieving reasonable performance. In addition, it was decided that the basic friction drag of the generator armature brushes represented such a large portion of the total energy output of the turbine that useful performance data could not be obtained. The test was therefore conducted with various resistor settings to adjust the load; however, no attempt was made to determine the actual power output.

The models tested consisted of several control configurations in addition to the fixed pitch configuration shown in Fig. 8. These were tested to explore various techniques designed to achieve coupling between rpm and blade pitch as well as coupling between blade flapping and pitch. The results of these tests are

reported on in the appendices, however only the fixed pitch and the final self-regulating configuration will be presented here.

Fixed Pitch Configuration

For fixed pitch testing a torque tube was used, which was supported at the hub with a universal joint and at the flexbeam end with a Thomas coupling. This device eliminates the torsional degree of freedom of the flexbeam but it does not effect the flatwise and edgewise degrees of freedom. The universal joint allows lagging and flapping motions, while the flexible connector (Thomas coupling) absorbs the changes of distance between the ends of the flexbeam during those motions. The blade pitch angle can be changed manually at the outboard end of the flexbeam.

For several blade pitch angles and wind velocities, the stabilized rotor speeds are given in Fig. 9. These results are with the turbine rotational axis aligned with the wind. At the low rotational speeds the blade pitch angle is not a significant parameter when the blades are stalled. This is indicated by the lowest curve in the figure. To obtain the higher rpm, unstalled equilibrium, power is momentarily added by using the generator as a motor. It is possible to achieve the high equilibrium by manually yawing the wind turbine 30-50 deg upstream thereby reducing the angle of attack on the advancing blade. This results in unstalling a portion of the disk and providing sufficient accelerating torque to increase the rpm to the high equilibrium condition. The turbine can then be returned to the unyawed position while maintaining rpm. Figure 9 shows that a fixed pitch equilibrium is possible at decreasing wind velocities for increasing blade pitch angles.

Sample stress time histories for fixed pitch are presented in Fig. 10. The flatwise and edgewise stresses are at the root end of the flexbeam ($\bar{r} = 0.07$) with the torsion stress at the midspan location of the flexbeam ($\bar{r} = 0.12$). Zero azimuth is with the blade pointed down and positive stresses correspond to flatwise deflections in the turbine thrust (downwind) direction, edgewise deflections in the direction of rotation, and torsional deflections are leading edge away from the wind so as to produce more positive flatwise deflection.

Under uniform inflow conditions, the flatwise stress level would show primarily a constant stress due to aerodynamic loading and also a one-per-revolution stress due to the effect of gravity on the coning angle. The edgewise stresses would show primarily a steady stress due to the driving torque of the turbine plus the one-per-revolution effect of gravity. However, since the model is Mach scaled resulting in the Froude number to be high by a factor equal to the scale factor, it is unlikely that the gravitational effects would be observable. The torsional stresses are zero in the fixed pitch configuration since the flexbeam is restricted in torsion.

Figure 9: Tip Speed Versus Wind Speed of Fixed Pitch Wind Turbine Model

The vibratory stresses are attributed primarily to the aerodynamic interference caused by the tower wake. This is a 1P excitation and results in reduction of the blade lift as it passes the tower. Figure 9 showed that the -4° pitch setting gave a higher rotational equilibrium speed than the -8° setting. The mean edgewise stress levels are thus higher for the pitch setting of -4° , reflecting the higher output torque. Edgewise stress also increases at constant pitch with increasing wind speed as shown in Fig. 10. The high frequency content of the edgewise stress is of no distinct origin.

Figure 10: Edgewise and Flatwise Stresses at Root of Flexbeam ($\bar{r} = 0.07$) in the Fixed Pitch Configuration

Hub-Mounted Pendulum Configuration

The system which was found to achieve most successfully the objective of self-regulation consisted of a pendulum attached directly to the hub and outboard of the pivot point connected through a flexible coupling to the leading edge of the outboard end of the flexbeam. Flapwise motions of the pendulum thus translate directly into torsional deflections of the flexbeam. The configuration is shown in Fig. 11. The flexbeam torsion spring and the spring due to pendulum flapping are continuously in equilibrium via the steel connector strap. The connector can flex to absorb the changes in distance between the pendulum and the end of the flexbeam due to coning angle and rpm. It should be noted that a full-scale blade would have the aerodynamic shell extend inboard almost to the hub to optimize the aerodynamics in this region. For convenience this portion was omitted in the model. Having the pivot point a finite distance from the center of rotation gives the pendulum a frequency above 1P, in this case about 1.5P. For a pivot located at the center of rotation, the pendulum frequency would be 1P and could adversely couple with the natural torsion frequency of the unrestrained blade, which is approximately 1P also. This would most probably result in an unstable system. However, a pendulum frequency of 1.5P is sufficiently displaced from 1P so as not to appreciably couple with the 1P frequency.

Figure 11: Configuration with Hub-mounted Pendulum

To achieve self-starting, the initial pitch angle must be set at approximately -30 deg. For a production design, this pitch setting would be built in relative to the untwisted flexbeam. For the wind tunnel model, a variable pitch capability was required and therefore a pitch bearing and locking device were located at the juncture of the flexbeam and the blade. The pendulum is likewise set statically at a large negative flapping angle. This angle depends on the kinematics of the pendulum, the spring rate of the flexbeam, and the geometry of their interconnection. The pendulum angle for the model wind turbine was set about -45 deg. As the wind speed increases from zero, an accelerating torque is generated by virtue of the large negative pitch angle. The development of rotational speed creates centrifugal loads in the pendulums, tending to move them towards an inplane position. The flexbeam spring opposes this motion and, for a given wind speed, an equilibrium position on each blade is reached where the moment produced by the pendulum is balanced by the twisting moment of the flexbeam. For light wind speeds this would occur at a relatively low rpm where little or no power generation would be expected. As wind speed increases further, rpm in turn increases, resulting in a higher blade pitch angle. The rpm-pitch schedule should be selected so that at design wind speed the advance ratio and pitch angle are approximately at their optimum, as indicated in Fig. 3. As wind speed increases beyond the design level, rpm increases proportionally, however the pitch angle would not increase beyond the angle corresponding to the pendulums in their inplane position. This angle should probably not exceed $+2$ deg.

When the pendulum is mounted statically on the downwind side of the turbine disk, a negative coupling of pitch and rpm is introduced which results in reduced pitch as rpm increases. Although this configuration was of secondary interest for this study, it was tested briefly to explore the capabilities of the pendulum concept as an rpm governor control. Such a device might be useful for systems where minimum rpm excursions are desired. A comparison of typical rpm schedules of this configuration with that of the self-starting configuration is presented in Fig. 12. The results demonstrate the reduced rpm excursions of the governor configuration.

Figure 12: Performance of Hub-mounted Pendulum Configuration

The self-aligning feature of the CBR wind turbine results from the inherent dynamic response characteristics of hingeless rotors. This arises from the phase lag of a simple spring-mass-damper system, wherein the mass experiences a finite time lag between the time of excitation and the time of response. The magnitude of the lag is a function of the frequency and damping of the system. In the case of the wind turbine which is yawed at an angle to the wind direction, the excitation results from the higher dynamic pressure of the blade rotating into the wind as compared to the blade moving away from the wind. These blades would be in their vertical positions when experiencing the greatest difference in pressure or lift. Their maximum flatwise

deflection would occur at some position during the next quarter of a revolution. This in turn would produce a moment at the blade root which feeds directly into the hub. This moment would produce both pitching and yawing moments on the rotor, however the only degree of freedom would be yaw. The turbine would respond to this moment and yaw in a direction to eliminate the differential lifts on the advancing and retreating blades. The motion would end when the blades are at equal lift which occurs generally when the turbine axis is aligned with the wind. In a free yawing system, such as that used for the CBR wind turbine, any wind direction change would create advancing and retreating blades and therefore the turbine would continuously self-align. The rate of alignment is proportional to the aerodynamic moment produced and inversely proportional to the inertia and rotational speed of the rotor.

Figure 13 shows the time history of the model at a wind speed of 25.8 mph where the static pitch angle was set at -31° . The turbine is initially yawed -90° . It is then released at $t = 0$, and initially the turbine weathervanes to the downwind position in a second order fashion. The tip speed then increases, and after about two seconds the pitch angle, under the action of the pendulum motion, begins to increase. This causes the continued acceleration through the start up. The torsional stresses change proportionally with the pitch angle. The edgewise and flatwise stresses show, at 14 seconds after release, a slight amplification at 6P which is caused by the coalescence of the first edgewise and the second flatwise natural frequencies. However the amplification is relatively mild and is quickly passed through.

Figure 13: Time History of Startup in Self-starting Configuration

The capability of the turbine in the self-starting configuration to respond to sudden wind direction changes is shown in Fig. 14. The turbine, rotating at its high rotational speed equilibrium, is released from a preset yaw angle. In a first order fashion the turbine tends to realign with the wind. The initial flatwise stress amplitude is due to the large helicopter type cyclic input because of the initial yaw angle. The results demonstrate that the CBR wind turbine in its self-starting configuration is self-aligning to the wind, self-regulating in rpm by means of automatic pitch changes, and has low vibratory stress levels during normal maneuvers of start up and alignment. Note that in Fig. 14 the turbine comes to a skewed yaw equilibrium because of the tower wake.

Figure 14: Response of Wind Turbine to Simulated Abrupt Wind Change

CONCLUSIONS

The conclusions set forth below were arrived at as a direct result of the model test program conducted under the ERDA Contract. Since the model was dynamically scaled it is expected that the conclusions are applicable as well to full-scale designs. However some variations in the full-scale characteristics might be expected as a result of the Reynolds number and Froude number difference, but the conclusions reached should generally be applicable. Reference to normal wind speeds means speeds up to approximately 30 mph.

1. The Composite Bearingless Rotor (CBR) can be used in conjunction with a hub-mounted pendulum to produce a self-regulating stable wind turbine. Self-regulating is defined as self-starting with an automatic pitch-rpm schedule to provide efficient power output at all reasonable wind speeds.
2. Wind turbines having hingeless blades of conventional stiffness, and free to yaw, have inherent dynamic moments which tend to maintain alignment of the turbine axis with the wind.

3. The CBR wind turbine in its self-regulating configuration operates well below stress allowables for normal wind speeds. This characteristic is independent of wind direction, initial yaw position, and initial rpm, and encompasses transient as well as steady operation.
4. For a free yawing wind turbine designed for operating downstream of the support tower, the rotor drag when nonrotating is normally sufficient to align the turbine to the wind for any initial wind direction.
5. The equilibrium yaw position for a free yawing wind turbine operating downstream of its support tower is skewed from the downwind position. This is due to unloading of the downward blade caused by the reduction in wind velocity behind the tower.
6. The CBR wind turbine with fixed pitch performs as a conventional wind turbine.

CONCLUDING REMARKS

The initial objective of the research program described herein was to determine a wind turbine configuration which demonstrated the potential for significant improvements in cost effectiveness over conventional designs. This particular study concentrated on the rotor itself and methods by which its initial and operating costs could be reduced. The study has shown that significant improvements could be expected by application of the Composite Bearingless Rotor (CBR), already established for helicopter applications as a simple low cost concept, and utilizing its specialized aeroelastics to evolve a completely self-regulating system. Such a system has been experimentally demonstrated using a small dynamically scaled model. An important milestone has been reached, however additional research is recommended before substantial investments are made in full-scale hardware. Two particular areas related to the operation of the rotor itself are: 1) the performance and self-regulating characteristics when operating in a more realistic Reynold's number environment, and 2) the self-aligning and self-regulating characteristics when coupled with a dynamically realistic tower. Design studies should also be conducted for a moderate size full-scale prototype, and that plans be established for an extended demonstration of the complete wind turbine system employing the CBR self-regulating concept.

NOMENCLATURE

b	Number of blades, flexbeam width, in.
C_l	Lift coefficient
c	Blade chord, in.
D	Drag, lb
E	Indicating edgewise
F	Indicating flatwise
J	Advance ratio, $\Omega R/V_W$
L	Lift, lb
R	Radius, ft or in.
r	Radial length, ft.
$\bar{r}$	Nondimensional radial length, $\frac{r}{R}$
T	Indicating torsion
V_W	Freestream wind velocity, mph
θ_{75}	Pitch angle at 75 percent R, deg.
σ	Turbine solidity, $bc_{.75}/\pi R$
Ω	Turbine rotational speed, rad/sec

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NOMENCLATURE FOR APPENDICES

a	Factor
aR	Fractional length of R (ft)
B	Width of box spar (inch)
b	Factor; number of blades; width flexbeam
bR	Fractional length of R (ft)
$C_{l_{max}}$	Maximum lift coefficient
C. F.	Centrifugal force (lb)
c	Factor; chord of airfoil section; chord of flexbeam
cR	Fractional length of R (ft)
D	Diameter
d	Factor
dR	Fractional length of R (ft)
d()	Infinitesimal amount of ()
E	Modulus of elasticity in tension (psi); Indicating edgewise
F	Indicating Flatwise
G	Modulus of elasticity in shear (psi)
H	Height of box spar (inch)
h	Hinge offset (ft)
I_e	Edgewise area moment of inertia (inch ⁴)
I_f	Flatwise area moment of inertia (inch ⁴)

NOMENCLATURE FOR APPENDICES (Cont'd.)

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), advance ratio $\frac{\Omega R}{V_W}$
k	Coupling factor $\frac{d\theta}{d\beta}$
k_θ	Torsional spring stiffness (ft lb/rad)
ℓ	Pendulum length (ft)
L/D	Lift drag ratio
M	Moment (inch-lbs)
M_θ	Moment caused by flexbeam twist (ft lb)
m	Mass (slug)
P	Power (HP)
R	Radius (ft)
r	Radial length (ft)
$\bar{r}$	Nondimensional radial length $\frac{r}{R}$
T	Indicating Torsion
$TE ()$	Trailing edge position () of pitch-flap coupling mechanism
t	Flexbeam thickness (inch); airfoil thickness (inch); material thickness, spar (inch)
t_1	Material thickness of spar (inch)
V_{tip}	Tip speed, (fps)
V_W	Freestream wind velocity (mph)

NOMENCLATURE FOR APPENDICES (Cont'd.)

w	Turbine blade weight per length
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); pendulum pitch angle (rad)
θ_{75}	Pitch angle at 75 percent R
λ_l	Wind turbine scale factor
ρ	Material density (slug/ft ³)
σ	Turbine Solidity
ψ	Azimuthal angle (rad); Yaw angle (rad)
Ω	Turbine rotational speed (rad/sec)
ω	Frequency (rad/sec)
$()_o, ()_i,$ $()_e$	Initial or equilibrium state of variable
$(\ddot{ })$	Second time derivative of ()

APPENDIX I

FLEXBEAM PROPERTIES

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. The full-scale flexbeam has the dimensions shown in the sketch.

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 $\bar{r}$:

$$\begin{aligned} GJ &= G b t^3 \beta \\ &= G (-89.6 \bar{r} + 54.5) \text{ lbs inch}^2 \end{aligned}$$

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 the angle $d\theta$ due to moment M .

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

or integrated:

$$\theta = \frac{MR}{G} \int_{0.05}^{0.2} \frac{d\bar{r}}{J(\bar{r})}$$

Therefore

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

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

For the model the properties are ($\lambda_\ell = 27.8$):

$$EI_f = 4.89 \cdot 10^8 (0.65 - \bar{r}) / \lambda_\ell^4$$

$$= 821.33 (0.65 - \bar{r})$$

$$EI_c = 1.358 \cdot 10^{11} (0.65 - \bar{r})^3 / \lambda_\ell^4$$

$$= 2.28 \cdot 10^5 (0.65 - \bar{r})^3$$

$$GJ = G (-89.6 \bar{r} + 54.4) / \lambda_\ell^4$$

$$= 1.68 \cdot 10^{-6} G (-89.6 \bar{r} + 54.5)$$

$$\frac{M}{\theta} = 3.44 \cdot 10^5 / \lambda_\ell^2$$

$$= 4.46 \cdot 10^2 \frac{\text{inch lbs}}{\text{rad.}}$$

APPENDIX II

MODEL DESIGN DETAILS

The inboard portion of the blade consists of the flexbeam, the preset pitch mechanism, the feathering mechanism, the rpm sensing weights, and the pitch-flap coupling mechanism. Drawings of these components are shown in Figs. A1 and A2. Not shown in these figures is the fixed pitch mechanism, which consists of a universal joint, supported at the hub, and a Thomas coupling at the outboard point of the flexbeam.

Several approaches were considered for the outboard portion of the blade spar which required specific mass and stiffness distributions for proper scaling of frequencies. A short study which examined these approaches is presented below.

At each radial location of the model 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	$\frac{GJ}{Ip}$	= 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 application, 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, however, solutions are obtained assisted by repeated hand calculations.

Detailed information of the model spar and blade properties, the blade aerodynamic twist, and the blade mode shapes is contained in Figs. A3 to A6.

APPENDIX III

ADDITIONAL EXPERIMENTAL RESULTS

In addition to the fixed pitch and hub-mounted pendulum data presented in the main body of this report, this appendix presents other results obtained during the course of the investigation. These include several mathematical derivations related to the kinematics and dynamics of the coupled blade/control pendulum system. A summary of the configurations and conditions tested during the experimental portion of this program is presented in Table AI.

Effects of Yaw

To determine the turbine behavior under skewed flow conditions, the model was positioned at various yaw angles in the tunnel and locked by means of the locking mechanism on the yaw table.

Figure A7 shows the fixed pitch equilibria of the turbine in a yawed position. With the wind turbine behind the tower and while looking upstream, the positive direction of rotation of the blades is counterclockwise. The positive yaw angle is defined by the advancing blade low whereas negative yaw angle is with the advancing blade high or above the tower. In comparison with Fig. 9 in the main body of the report, the yawed equilibrium has a lower rpm than the unyawed equilibrium. This is in general to be expected, since the mass flow through the rotor has been reduced by yawing.

Figure A7 shows also the asymmetry in behavior at equal yaw angle deflections from the downwind position. This is shown for a pitch angle of zero degrees. The positively yawed turbine has a lower equilibrium rpm because the advancing blade, which contributes most of the accelerating torque, passes through the tower shadow. The negatively yawed turbine has its advancing blade pass through undisturbed air and therefore, results in a higher equilibrium rpm.

Pitch-Flap Coupling Configuration

Analogous to the pitch-flap coupling hinge of a helicopter rotor (83 hinge), the wind turbine in the pitch-flap coupled configuration alleviates flapping responses caused by wind gusts by reducing blade pitch angle. The mechanical details of the model in this configuration are shown in Figs. A2 and A8. The flexbeam is free to flap and to lag, but its torsional degree of freedom is coupled to the flapping motion by the pitch-flap yoke. The sliding joint eliminates any forces which would follow from the distance changes between the pivot and the hub because of flapping and lagging deflections. The pivot is there to transfer only the torsional moments of the pitch-flap coupling. When the flexbeam deflects in flapping, the mechanism causes a torsional moment in the

flexbeam, which changes the blade pitch. The gain of the coupling between pitch and the flapping motion is inversely proportional to the distance of the sliding joint to the elastic axis of the flexbeam. The bracket, which constrains the joint can be mounted in four positions before or behind the flexbeam. Behind the flexbeam are those positions, which are on the trailing edge (TE) side of the flexbeam. The positions are numbered in decreasing gain, from TE1 to TE4 (Fig. A2). The TE positions decrease the pitch angle of the blade for increasing flapping angle.

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. A2. 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 A2 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 :

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 decreases with increasing rpm. Factors b and c are determined by geometry.

In the above, the rotation of line AB is neglected. AB rotates to satisfy the boundary condition of the yoke attachment at A and the straight line connection to the hinge point of the δ_3 yoke.

The performance of the unyawed turbine with the pitch-flap coupling is shown in Fig. A9, together with the corresponding fixed pitch results. The blade pitch angle in the pitch-flap coupled configuration is set at zero rpm. Therefore, the pitch angle of the blade in operation is smaller because of the negative gain of the coupling and the positive coning angle. At high rpm this coupling gain tends to zero and can become positive. For the cases where the coupling gain is negative, the equilibrium rpm is lower than for the corresponding fixed pitch gain, as shown in Fig. A9. It is also shown that at high rpm the coupling gain reduces and therefore, the fixed pitch and

coupled configuration results are similar. Figure A9 shows that the distance between the fixed point of the sliding joint and the elastic axis of the flexbeam (TE1, TE2, etc.) has only a minor effect.

Another performance aspect of the pitch-flap coupled configuration is the yawing moment about the turbine yawing axis. The aerodynamic input of the tower into the turbine is alleviated by the pitch-flap coupling. This follows from the yaw equilibrium of the turbine while yawing freely. As previously (Fig. 14) stated in the main text under the simulation of an abrupt wind change the yaw equilibrium is skewed with the wind direction. The new yaw equilibrium of the pitch-flap coupled turbine is typically reached at about half the yawing angle of the corresponding fixed pitch, free yawing case. This means a reduced cyclic input to the turbine due to yawing. This can be further reduced, and was demonstrated experimentally by extracting power from the turbine which reduces rpm thereby increasing the extent of inboard stall, and thus creating a clockwise yawing moment.

Time histories of the edgewise, flatwise, and torsional stresses during one revolution are given in Fig. A10. By comparing the torsional stress signals of the left and right hand column, it follows that the TE2 position has a higher gain than the TE4 position. The oscillatory torsional stresses for equal amounts of flapping are higher in the TE2 case and therefore, the amplitude of the blade pitch is higher as well. In the TE4 cases the edgewise and flatwise stresses are nearly identical to the fixed pitch cases. Because of the low damping of the torsional mode in TE2, the transient motions dominate the stresses.

Blade-Mounted Pendulum Configuration

The ability to use the centrifugal forces on a mass fixed to the blade to sense rotational speed is readily applicable to the CBR wind turbine, given its torsionally soft flexbeam. The sensor can simultaneously be the actuator. Figure A11 shows the model hub in such a rpm-pitch coupled configuration. The centrifugal weights of one flexbeam are mounted on opposite sides of the plane of rotation. When the rpm increases, the weights move towards the plane of rotation, thereby twisting the flexbeam to obtain a new moment equilibrium about the elastic axis of the beam and thereby changing the pitch angle of the blade. By mounting the centrifugal weight above or under the plane of rotation, a sign change of the gain of the rpm blade pitch mechanism can be obtained.

It can be shown that the natural torsional frequency of a free feathering flat rotating beam is equal to the rotational frequency, or 1P. Unless significant damping is provided, a rotating system of this type would be violently unstable if significant 1P aerodynamic excitation is present. The CBR wind turbine configuration with a blade-mounted pendulum resulted in a system very similar to this. The flexbeam torsional stiffness was relatively low compared to the dominate pendulum stiffness and thus, the system assumed a frequency nearly the same as the pendulum frequency, or 1P. In the torsional mode, very little damping is present. This, coupled with the 1P excitation due to the tower wake, resulted in an unstable system. Several low speed runs were made, and although some conditions appeared steady, small changes in wind speed would result in violent blade flapping. It was decided not to risk possible blade damage by further exploration of this configuration and thus, its testing was discontinued. A control system of this type may be feasible if the torsional frequency could be brought above 1P to approximately 1.5P or higher. This frequency is placed sufficiently away from the 1P excitation to avoid adverse amplification.

The following derivations are presented to indicate the coupling mechanism and the equation of motion of a pendulum fixed to the blade.

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.

Assume the mass m is positioned azimuthally, angle ψ , with respect to the torsional axis of the flexbeam.

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

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

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

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

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:

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

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

$$L_1 = \frac{\bar{r}}{\cos \psi \cos \beta}$$

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

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 flexbeam doubles the coupling term $\frac{d\theta}{d\Omega}$.

In formulating the equation's of motion of this system assume the arrangement shown in the sketch.

if $k_\theta = \left(\frac{M}{\theta}\right)_{\text{flexbeam}}$, then the torsional spring term in the equation of motion is

$$k_\theta \theta + 2 m \Omega^2 l^2 \sin \theta \cos \theta$$

The equation of motion becomes:

$$2 m l^2 \ddot{\theta} + k_\theta \theta + 2 m \Omega^2 l^2 \sin \theta \cos \theta = 0$$

or

$$\ddot{\theta} + \frac{k_\theta}{2 m l^2} \theta + \frac{\Omega^2}{2} \sin 2 \theta = 0$$

It should be noted that the term r does not enter the equation.

The perturbation equation about the equilibrium angle, θ_e , becomes:

$$\ddot{\theta} + \frac{k_\theta}{2 m l^2} \theta + \Omega^2 \cos 2 \theta_e \theta = 0$$

The last term in the equation represents the frequency contribution of the weights. At $\theta_e = 45^\circ$ the perturbational torsional stiffening is zero. For smaller angles of θ_e the frequency increases, but cannot exceed a frequency of one per revolution. The added stiffness due to the flexbeam stiffness, k_θ , increases this frequency slightly but this stiffness is necessarily low due to the large required pitch variations and therefore has little influence. At best, a system with a natural frequency close to 1P is neutrally stable. From experiments it became clear that the system damping is not sufficient to keep the amplitudes from growing rapidly.

The steady state of the mechanism is expressed by the spring constants:

$$\frac{k_\theta}{m\ell^2} \theta_o + \Omega^2 \sin 2 \theta_e = 0$$

where θ_o is the pitch excursion from θ_i at $\Omega R = 0$ to θ_e at the operational condition,

or
$$\theta_i = \theta_e + \frac{m\Omega^2 \ell^2}{k_\theta} \sin 2 \theta_e$$

This relationship is presented graphically below.

For the mechanism to be sensitive to rpm changes, k_θ must be small. At positive θ_i ($d\theta/d\Omega < 0$ tending to keep ΩR constant) and rpm, the blade is stalled which prevents start up. For self starting, $d\theta/d\Omega$ must be positive since the initial start up pitch must be a large negative value. The initial blade pitch is negative and as the rpm increases, the pitch increases ($d\theta/d\Omega > 0$) to the operation value.

Operation at High Wind Speeds

Under high wind conditions wind turbines are normally locked in a nonrotating position and the blades feathered to minimize aerodynamic loading. It is conceivable that wind turbines and support towers could be designed to withstand winds of any expected intensity while rotating, and such a capability would be highly desirable. However, it was not the objective of this study to explore this aspect of wind turbine operation. Tests were made in the nonrotating, feathered configuration at several wind speeds up to 60 mph, and the data acquired consisted of blade stresses as well as observations concerning blade stability. The model wind turbine was tested first with the blades freely feathered and the turbine free in yaw. At relatively low speeds a flatwise oscillation would begin and couple with the free pitch degree of freedom of the blade. The onset of the oscillations did not seem to be sensitive to the yaw angle of the turbine. As wind speed was increased, the flatwise and pitching oscillations increased and the blades would rotate at low rpm in an unsteady fashion. At 27 mph the flatwise deflections caused the blades to nearly contact the tower at which point the test was stopped. The blades were then locked into the feathered position and the test repeated. Under these conditions blade oscillations were negligible, and the turbine demonstrated highly stable weathervaning characteristics, significantly more stable than in the free feathering mode. The experimental steady flatwise stress results are shown in Fig. A12 along with the calculated results. It should be noted that since the model blade feathering motion takes place at the outboard end of the flexbeam, blade edgewise deflections due to wind drag result in flatwise deflections of the flexbeam. The flatwise stresses in the flexbeam are shown to increase rapidly with the wind velocity, but remain well below the stress allowable. Small torsional oscillations about a zero mean existed but were negligible and not included in Fig. A12.

APPENDIX IV

MODEL WIND TURBINE THEORETICAL RESULTS

The UTRC CBR Aeroelastic Analysis (G400), Ref. 6, was used to calculate the model wind turbine performance and blade stresses. The blade natural frequencies and mode shapes used in the analysis were obtained from program E159 and presented earlier in Figs. 7 and A6. The calculations were performed assuming constant inflow and thus, no velocity defect due to the tower wake was used. Also, gravity effects were ignored due to the nonsimulated Froude number. The resulting stresses therefore, have no harmonic content. The results for two fixed pitch test conditions, $\theta_{75} = -4$ deg and $\theta_{75} = -8$ deg, are presented in Fig. A13. Thrust, power output, and steady flatwise stresses are presented for a range of wind speeds from 10 mph to 30 mph. Edgewise and torsional stresses for the conditions studied were negligible (less than 1000 psi) compared to the flatwise stresses and are not presented. At each pitch setting the tip speed was held constant for all wind speeds. Under these conditions the blade section angles of attack generally increase directly with wind speed and therefore, blade lift and stresses follow in the same manner. This can be seen by the relatively linear increase in these quantities in Fig. A13. The steady stress values taken from the experimental results presented in Fig. 10 are also shown as data points on Fig. A13. As indicated, the theory overpredicts the steady stress levels for the two points shown. Two possible causes for this difference are: 1) the torque tube used for the fixed pitch configuration produces a flap restraining moment on the blade as coning increases. This would tend to reduce coning and in turn the steady flatwise stress. This effect was not simulated in the analysis, and 2) the analysis assumes constant inflow which may place the center of lift of the blade further outboard than that resulting from the actual inflow distribution. If the center of lift moved inboard, the root moment and stresses would decrease. Although the analysis is capable of accepting variable inflow input, the generation of such data was beyond the scope of this contract.

APPENDIX V

NATURAL FREQUENCY OF ROTATING MASS WITH HINGE OFFSET

Presented in this appendix is the derivation of the natural frequency of a simple pendulum operating in a rotating environment.

$$C.F. = m\Omega^2(h + l \cos \beta_e)$$

The external moment M_θ caused by the twisted flexbeam establishes the pendulum's equilibrium at angle β_e

Equation of Motion:

$$M_\theta - ml^2\ddot{\beta}_e - m\Omega^2 l (h + l \cos \beta_e) \sin \beta_e = 0$$

$$\ddot{\beta}_e + \Omega^2 \left(\frac{h}{l} + \cos \beta_e \right) \sin \beta_e = \frac{M_\theta}{ml^2}$$

The perturbation equation becomes:

$$\ddot{\beta} + \Omega^2 \left(\frac{h}{l} + \cos \beta_e \right) \cos \beta_e \beta = 0$$

The natural frequency of the above offset pendulum is:

$$\Omega \sqrt{\left(\frac{h}{l} + \cos \beta_e \right) \cos \beta_e}$$

It should be noted, that this frequency is independent of mass m . A practical choice is to design now for the natural frequency 1.5P

$$1.5^2 = \left(\frac{h}{l} + \cos \beta_e \right) \cos \beta_e$$

Solving for the offset parameter, h/l , gives,

$\cos \beta_e$	0	10°	20°
h/l	1.25	1.30	1.45

A design where the hinge offset and the pendulum length are approximately of equal length seems reasonable. Although the natural frequency is not dependent on mass, the spring effect, M_0 , is dependent and must be considered when the pendulum is acting to oppose the torsion spring of the flexbeam.

APPENDIX VI

EFFECTS OF TOWER WAKE

Because of the influence of the tower wake on the stresses in the flexbeam, as shown in Fig. 10, variations in the tower blockage were made in an effort to quantify this effect. Figure A1⁴ shows the result of this study for three geometries. The original open truss tower, see Fig. 8, which is called Geometry 3, has been wrapped to obtain the circular tower, Geometry 1. Geometry 2 is called the flat plate tower, where the upstream face of the square tower was blocked.

At the low speed rotational equilibrium of the rotor, only the flatwise peak-to-peak stresses are significant. Without a tower, there would be a steady stress. As follows from Fig. A1⁴ the open truss tower gives the lowest stress amplitude. At the high speed equilibrium of the unyawed rotor the edgewise moments become more important, in particular for the flat plate tower. The differences in stress amplitudes between the flat plate tower and the open truss tower is of an order of magnitude. It should be kept in mind that these stress amplitudes do not contain realistic gravitational effects. At the high speed equilibrium of the yawed turbine, the flatwise stress amplitudes remain about the same, while the edgewise stress amplitudes are somewhat reduced. Yawing the turbine reduces the tower effect but adds a one per revolution cyclic stress component due to the advancing and retreating blade effect. The total stress amplitude remains the same.

Not only the momentary stresses but also the stresses integrated over one revolution are of importance. From the high speed yawed rotational equilibrium of the flat plate tower configuration and the open truss tower configuration, the turbine is released to simulate a sudden wind direction change. The yawing motion, shown on the right side of Fig. A1⁴ is well damped and of a first order character. However, the motion does not stop at the downwind position of the turbine, but instead increases positively. The flat plate tower case proceeds beyond a yaw angle of 40° , after which the rotational equilibrium breaks down. In a second order fashion the turbine returns to the downwind position, typical for the low speed equilibria. For the open truss tower the high speed equilibrium exists under yawed conditions. The figure shows that with the turbine in the downwind position, there exists a considerable yawing moment, which is caused by the tower wake. In a simplified manner, the phenomena can be explained as follows: The turbine is very inert and the turbine responds 90° later. This causes the turbine to yaw in a positive direction, until the yawing moment is balanced by the aerodynamic moment about the yaw axis. The above explanation clearly does not hold when the rotational

speed of the turbine is much lower. The explanation must then be given with the phasing of the response of the blade to the input of the tower. Experiments have shown that at low rpm the free yawing turbine misaligns itself with negative yaw angles. This most probably is a result of the larger pulling load on the upward blade compared to the downward blade which has less load and coning due to the reduced velocity. This effect is greater than the phase lag effect of the downward blade which is the major contributor at the higher rpm. When the blade is completely stalled, the turbine aligns directly to the wind.

TABLE AI
SUMMARY OF TEST CASES

Run#	Blade Pitch	# Cases	Yaw Angle	RPM	Config.	Remarks
1-9	-17° → -2°	33	0°	low	θ const.	<u>Fixed Pitch Configur.</u> some cases turbine is driven
10-26	-18° → 0°	222	0°, free, yawed	high/low	"	releases from preset yaw angle
28	0°	13	" " "	high	"	" " " " "
27	-4°	11	0°, 15°, 30°	high/low	"	study of tower wake
34-36	-8°, -4°, 0°	32	0°, free	high	TE 3	<u>Pitch-Flap Coupling</u> few cases power extracted
38-40	-8°, -4°, 0°	31	0°, free	high	TE 2	" " " "
41-43	-8°, -4°, 0°	28	0°, free	high	TE 4	" " " "
37	0°	1	0°	low	TE 1	vibrations because of high gain
45	-8°	4	0°	high/low	TE 4	<u>Blade-Mounted Pendulum</u> Pendulum unstable without δ ₃
53-55	-19°, -17°, -15°	81	0°, ±15°, ±30°	high	Pendulum angle -15°	<u>Hub-Mounted Pendulum</u> self-starting
56	-4°	10	0°, ±15°, ±22.5°, -30°	high	" " +15°	rpm-control
57, 58, 60	-33°, -31°	30	0°, ±15°, ±30°, ±90°	high	" " -52°	self-starting
62, 63	-4°, 0°	19	0°, ±15°, -30°	high	" " +42°	rpm-control
64	free	25	0-180°	~0	θ free	<u>Severe Wind Conditions</u> Pitch and flexbeam oscillations
64	-90°	10	0°, free	~0	θ constant	Stable flatwise deflections

FIGURE A1: MODEL BLADE DETAIL

FIGURE A2: MODEL HUB AND BLADE ROOT DETAILS

FIGURE A3: DIMENSIONS OF MODEL SPAR CROSS-SECTIONS

FIGURE A4: WEIGHT DISTRIBUTION AND SECTION PROPERTIES OF MODEL BLADE

FIGURE A5: GEOMETRIC TWIST OF TURBINE BLADE

FIGURE A6: MODE SHAPES OF MODEL BLADE

FIGURE A7: TIP SPEED VERSUS WIND SPEED OF FIXED PITCH YAWED WIND TURBINE MODEL

FIGURE A8: CONFIGURATION WITH PITCH-FLAP COUPLING

FIGURE A9: EFFECT OF PITCH-FLAP COUPLING ON TIP SPEED

$$\theta_{75} = -8^\circ$$

COUPLING TE2

$$V_w = 18.9 \text{ mph}$$
$$\Omega R = 200 \text{ fps}$$

COUPLING TE4

$$V_w = 23.6 \text{ mph}$$
$$\Omega R = 298 \text{ fps}$$

COUPLING TE2

$$V_w = 23.6 \text{ mph}$$
$$\Omega R = 179 \text{ fps}$$

COUPLING TE4

 $V_w = 23.6 \text{ mph}$
$$\Omega R = 209 \text{ fps}$$

FIGURES HAVE COMMON TIME SCALE OF 2 IN. = 0.1 SECOND

FIGURE A10: STRESS CHARACTERISTICS WITH PITCH-FLAP COUPLING

FIGURE A11: CONFIGURATION WITH BLADE-MOUNTED PENDULUM

FIGURE A12 : FLEXBEAM STRESS IN NONROTATING FEATHERED MODE
AT HIGH WIND SPEEDS

$\theta_{75} = -4^\circ$ $\Omega R = 216$ FPS

$\theta_{75} = -8^\circ$ $\Omega R = 193$ FPS

FIGURE A13: CALCULATED MODEL PERFORMANCE AND STRESS CHARACTERISTICS

FIGURE A14 : THE EFFECT OF THE TOWER WAKE ON THE PEAK TO PEAK STRESSES IN THE FLEXBEAM. FIXED PITCH CONFIGURATION.