DESIGN OF A SELF-REGULATING COMPOSITE BEARINGLESS BLADE WIND TURBINE

FINAL REPORT

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Design of a Self Regulating
Bearingless Blade Wind Turbine

ABSTRACT

A study was undertaken to design a 40-ft diameter wind turbine employing the UTRC/ERDA Self-Regulating Composite Bearingless Rotor (CBR) concept. The CBR concept was developed at United Technologies for rotary wing applications and is now in use on Sikorsky helicopters. The concept was further developed for wind turbine applications at UTRC under an ERDA contract in 1975-76. Successful wind tunnel tests were conducted during that contract, which demonstrated the self-starting and self-regulating features. The latest contract was to design a 40-ft system in the 5 kW - 15 kW power range. This effort included performance tradeoff studies, stress analyses of the blade and tower structure, a stability investigation, and engineering drawings of the complete system. However an overall cost analysis was not performed in this study.

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SUMMARY

The Composite Bearingless Rotor Concept, developed for helicopter applications at United Technologies Corporation, has been used in a wind turbine design to achieve a simple, low cost wind energy conversion system. In 1975-76 an Energy Research and Development Administration contract was executed to investigate the feasibility of this concept for wind turbines, and the findings of that investigation lead to the Department of Energy contract to design a 10 kW system, the results of which are reported herein.

The design consists of a 2-bladed downwind rotor, 40 ft in diameter, with two hub mounted pendula connected to each blade to achieve automatic blade pitch control. The pendula act under centrifugal forces to vary blade pitch as a function of rotor speed. The blade pitch (torsion) is produced by the twisting action of the inboard "flexbeam" region of the blade. Initially set at a negative pitch to provide a starting torque, the blade pitch then changes to an optimum performance position as windspeed increases. The rotor is connected via a step-up belt drive to a generator delivering electric power of variable voltage and variable frequency. At wind speeds up to 60 mph it is shown that the stresses in the rotor are well below the allowable stresses and that the action of the pendula causes partial stall of the blade, which limits inherently the turbine rotational speed. This feature allows the generator to be driven continuously by the turbine at all wind speeds. Speeds higher than 60 mph were not examined as part of this contract, however, a rotor brake would most likely be employed above 60 mph.

Within the scope of the design, a system has been arrived at, which is inherently self-regulating and self-starting, and its operation does not require attendance nor active tower mounted control systems. This report contains results of the performance, stress, and stability analyses and engineering layout drawings in reduced form of the complete system.

INTRODUCTION

The Composite Bearingless Rotor (CBR) concept was first demonstrated in helicopter model tests at the United Technologies Research Center (UTRC) nearly ten years ago as part of an ongoing research program to explore simple, low cost rotor systems. Most recently the CBR has been selected by the Sikorsky Aircraft Division for their UTTAS and S-76 tail rotors and has been successfully flight tested over the past year.

The use of the CBR concept in horizontal axis wind turbines was considered a natural application due to the stringent requirement of low cost for wind energy conversion systems. A contract was received from ERDA in 1975 to study the feasibility of the CBR for wind turbine applications and this work resulted in the wind tunnel testing of a dynamically scaled model in 1976. This work is reported in Ref. 1. A self-starting and self-regulating configuration was conceived and successfully tested during these tests. This provided the impetus to pursue a full scale demonstration of this concept, which has led to the current contract to design a 40-ft diameter demonstrator turbine in the 5 to 15 kW power range. This size was selected, with the assistance of the ERDA Wind Energy Office, to be the optimum size for low power requirements. The primary applications anticipated would be for power needs at remote locations, farming and irrigation, and individual home use; and the system would be designed for easy transportability and erection.

The objectives of this contract were to complete the design of the total system, to calculate performance and stability characteristics, and to prepare engineering drawings in sufficient detail to fabricate all components of the demonstrator system. Particular emphasis is placed on low initial cost, low maintenance, high reliability, and the ability to survive high wind conditions without disassembly or manual adjustments.

ROTOR DESIGN

Background

There were two principle considerations, in addition to the requirement for low cost, that determined the basic approach taken in the design of the 40-ft diameter rotor for this wind turbine system. The first was that the fundamental flatwise frequency be low (approximately 1.1 times rotor speed, or 1.1 P) so that the rotor stiffness be primarily made up by centrifugal stiffness thereby reducing the blade structure required to support rotor moments. This is also consistent with the model design which was successfully tested during the first year contract (Ref. 1). The second consideration is the problem of high wind speed conditions and the design trade-offs which would be necessary to achieve a rotor system capable of surviving all such conditions without the need for blade feathering or turbine shutdown. The major concerns here are, of course, that the stress levels in the blade and support structure be acceptable, but also that the blade edgewise frequency not drop close to 1P with the possibility of dynamic amplification due to 1P aerodynamic and gravity excitations.

Configuration Selection

The large scale ERDA wind turbine development programs have generally concentrated on rotors designed to achieve optimum performance. The blade configurations resulting from these programs, e.g., the Mod 0 and Mod 1, are highly twisted and tapered. It has been shown that such techniques are necessary to improve performance; that is, to approach as closely as possible the optimum power coefficient of 0.59. Realistically, no such design can exceed approximately 0.45. Since the overriding considerations in the design of the UTRC/ERDA wind turbine are cost and reliability, the logical first configuration to consider would be a single blade without twist or taper. Eliminating the single blade design due to the potential problems associated with aerodynamic unbalance, the initial configuration to be studied would be a 2-bladed untwisted rotor with constant chord. A review of the literature, for example Refs. 2 and 3, indicates that increasing the number of blades and/or solidity improves efficiency. For a fixed number of blades, increasing solidity, or chord, improves performance, however, the gains diminish as solidity increases. Thus, it becomes economically impractical to add chord when performance improvements become insignificant. Performance studies at UTRC have shown that only minor gains in performance are achieved above a turbine solidity of approximately 3 percent, which is low even for helicopter rotors. This results in an aspect ratio greater than 21 for a constant chord blade, which would result in a relatively flexible blade in the edgewise direction where centrifugal stiffness is minimal. In order to achieve an edgewise frequency greater than 1P for this high aspect ratio blade at overspeed conditions (tip speeds from 400 to 600 ft/sec) extremely high stiffness-to-weight blades would be required which would necessitate costly materials and fabrication processes. A nominal increase in chord would allow an easier design solution since the edgewise stiffness increases as the fourth power of chord. The increased chord would result in a lower cost blade and probably have a slight improvement in performance.

The structural aspects of the configuration, such as stresses are not considered here, because stresses can be reduced through minor expenditure in materials. Stresses reduce with the third power of material thickness, while material price is roughly proportional to thickness. Therefore, to some extent the chord can be changed without consideration of structural aspects. An aspect ratio of 16 was selected which corresponds to about 2 percent solidity per blade. Using this size as a standard, several variations in taper, twist, number of blades, and tip speed were made to determine performance trade-offs at the design wind speed of 12 mph. These results are presented in the following subsection.

Performance Comparison

The maximum power available in a 12 mph steady wind, assuming no wind shear, is 0.0089 kW/ft^2 . This translates to 11.1 kW for a 40-ft diameter circle. For a standard velocity duration curve, derived from interpolation of NASA Lewis RFP 3-499774-Q data, based on a 12 mph wind site, the maximum velocity attained is 33 mph and the total energy available for one year, assuming the generator is capable of operation up to 15 kW and assuming 36 percent overall efficiency, is 29,000 kW-hr. A discussion of the available energy at a 12 mph wind site is presented later, and a review of the wind characteristics in the Hartford, Connecticut area is presented in Appendix 1.

For a 12 mph wind speed, performance calculations were made for a 40-ft diameter rotor with a nominal solidity per blade of 0.02. The prescribed wake hover performance program, Ref. 4, was used to calculate performance of the various configurations presented in this section. A classical undistorted Goldstein wake was assumed, which is probable somewhat conservative, since actual wind turbine wakes expand behind the rotor, tending to improve performance over the undistorted columned wake. A cambered airfoil section recently developed by the Sikorsky Aircraft Division, and designated SC 1012, was used for this investigation. For this parametric variation study, it was assumed that azimuthal variations in inflow velocity would not influence comparative performance results, and thus effects such as wind shear and tower shadow were ignored. Figure 1 presents the results of the computer program showing the variation in shaft power output with blade pitch for several blade configurations. The maximum power is achieved with ideal twist and ideal taper (Ref. 5) which, for this condition, is shown to be 5.2 kW. Since tapered blades represent an expected cost increase, the reduction in performance with untapered blades was of prime consideration. Figure 1 shows a reduction of about 5 percent for the untapered ideal twist case which is minimal considering the added fabrication costs of approximately 20 percent for tapered blades. (This increment in cost was estimated from recent experience in filament wound fiber glass blades.) A further improvement in cost would be expected if a less sophisticated twist distribution or no twist were used.

A 7 percent loss in performance is shown for a linear twist of 10 deg over the entire span and a further reduction is shown for the untwisted blade over

WIND SPEED, $V_W = 12$ MPHBLADE CHORD, $C = 15$ INTIP SPEED, $\Omega R = 150$ fps (EXCEPT WHERE NOTED)IDEAL θ_1 IS HYPERBOLIC BLADE TWIST

Figure 1. Performance Comparison at Design Wind Speed

portions of the power curve, however essentially no loss is shown for the peak performance. Since the pitch schedule can be selected to operate at the peak performance for the 12 mph condition, it was decided to use the untwisted blade due to its favorable impact on cost. (If, on a production basis, twist was found not to effect cost then optimum twist would be used.) Variations in tip speed and number of blades were also made and these results are included in Fig. 1.

It is seen that the optimum tip speed at 12 mph is 150 fps. This corresponds to an advance ratio of 8.5 which has been shown in various studies and tests to be at or near the optimum advance ratio. Increasing the number of blades from two to four improves performance by about 10 percent, however studies have shown that 2-bladed systems are generally more cost effective, and thus rotors with higher numbers of blades would not be considered unless the stability study demonstrated a need for increased number of blades.

Blade Design

The blade consists of two basic components, the flexbeam extending from the nondimensional radial length $\bar{r} = 0.042$ to 0.190 , a transition region to $\bar{r} = 0.280$ and the blade proper from $\bar{r} = 0.280$ to the tip, see Fig. 2. The flexbeam serves as the member which undergoes the primary twisting and bending. Not only do the blade

Figure 2. Typical Blade Lay-out and Schematic of Primary Functions of Flexbeam

pitch angle changes occur over this region, but the major flatwise and edgewise flexing also take place here. Since large torsional deflections are experienced in the flexbeam, it is desirable to use a material of low shear modulus to reduce the torsional moment requirements. The 4.5-ft model tested during Phase I of this program used an unidirectional graphite/epoxy flexbeam which was shown to be satisfactory for the conditions tested. Initially in this design study, it was felt that a flexbeam of reduced length would improve the overall design. As a result, Kevlar 49 was studied as a replacement material for the graphite/epoxy flexbeam. This material has a shear modulus of approximately 50 percent of that of graphite/epoxy and therefore a corresponding reduction in flexbeam length would be expected. The material was used in the aeroelastic analysis and found to be satisfactory for the lower wind speed conditions, however for the high speed conditions, the reduced strength of the Kevlar 49 (approximately 50 percent of that of graphite/epoxy) resulted in inadequate safety margins for flatwise stress. This, coupled with the poor moisture resistance characteristics of Kevlar 49, resulted in the decision to use graphite/epoxy.

The dimensions of the flexbeam were initially selected by scaling up the model flexbeam (Ref. 1). However, the full-scale design was to include operation under high speed conditions which was not considered in the model design, and thus the flexbeam had to be reexamined in light of this new requirement. The flexbeam dimensions (length, thickness, and width) are determined primarily by four requirements: 1) tip clearance during start-up, 2) first flatwise mode frequency below approximately 1.1P at maximum rpm to minimize hub and tower moments, 3) edgewise

frequency safely above 1P at the highest tip speed expected, and 4) torsional stress safely below the material allowable at the maximum flexbeam twist. Tip clearance is principally a function of flatwise stiffness of the flexbeam. Cyclic pitch inputs are generated as a result of gravity effects on the pendulum motion (to be discussed later) and cause one-per-rev flapping of the blades. Under conditions of low rotational speed where centrifugal stiffness is minimal, the flapping deflections are determined mostly by the blades' structural stiffness. It was decided to limit the calculated tip deflections to at most one half the tip clearance of the undeflected lower blade with the tower guy wires (see "Specifications, Weight, and Drawings"). The flexbeam thickness was selected to produce a first modal frequency of approximately 1.08P at 500 ft/sec tip speed, and with this flexbeam configuration the start-up tip deflections were checked to ensure that they met the above criteria. The flexbeam width was sized by the high rpm condition where a first edgewise mode frequency of 1.5P was selected as the minimum acceptable frequency of operation. These requirements resulted in flexbeam dimensions of 0.7 in. by 7 in., and the flexbeam length was determined by the maximum twist expected at the high rpm condition where the pendulum has reached its in-plane position. Assuming a starting pitch angle of -16 deg, which is the minimum negative angle for adequate starting torque at zero rotor speed, and a high speed pitch angle requirement of +6 deg, this produces a total twist of 22 deg. Because of the critical nature of this degree of freedom, a conservative safety factor of 3 was used. With a stress allowable of 16,000 psi (Ref. 6) this places the maximum design stress at $16,000 \times 1/3 = 5333$ psi. Solving from a mechanical engineering handbook the empirical torsional equation of a slender rectangular cross section for the flexbeam length L, necessary to twist the beam 22 degrees while not exceeding the above maximum shear, gives 32.8 inches. Shear modulus G of unidirectional carbon epoxy is 0.65×10^6 psi.

The connection of the flexbeam to the blade at $r = 0.19$ would be nonadjustable for a production system, however, for the demonstrator wind turbine, a joint is provided to allow independent adjustments of the control strap and blade pitch. The final positions of these components would be determined during the demonstration tests.

A low risk design approach is being taken for the demonstration blade since the most important objective of this program is to first demonstrate the overall concept in full scale. Using a conventional rather than innovative blade design would represent the least interference in achieving this objective. A fiberglass spar and shell configuration was selected, which utilizes state-of-the-art technology and which is considered low risk. A cross section view of the blade is shown in Fig. 3. Also shown on this figure is a possible low cost design, which would employ an injection molding process. The fiberglass design is shown with a brass counterweight, which is incorporated to place the c.g. at the 1/4 chord. The need for 1/4 chord balancing has not been established analytically; however, again, the low risk approach has been taken for the demonstrator blade.

A UTRC blade frequency and mode shape computer program (named E159) was used to calculate frequencies and to provide the necessary mode shape information used in the UTRC aeroelastic analysis (named G400). The frequency diagrams are presented

Figure 3. Blade Cross-Section of Sikorsky Camber 1012 Airfoil

in Figs. 4 and 5. The torsional frequency shown on these figures includes the spring effect of the pendulum but does not include the mass effect. In using the fixed hub G400 computer program to initially design the blade, the pendulum could not be completely simulated. However, this was not considered a serious deficiency for this phase of the design, since the conditions studied with G400 were steady state. A complete simulation of the pendulum system is included in the multiblade finite impedance hub aeroelastic analysis (named F-762 WT) used for the stability investigation presented later in this report.

Pendulum Design

Three tools were developed for the design of the pendulum mechanism. The first was a simplified analytical model of the pendulum/flexbeam system based on kinetics, where the centrifugal and gravity forces acting on the pendulum weight are in equilibrium with the flexbeam twist. In the model, the strap connecting the pendulum with the flexbeam is considered rigid in edgewise bending. The second tool was a simplified full-scale experimental simulation of the pendulum-flexbeam mechanism. The geometry of the mechanism was simulated, and the torsional stiffness of the flexbeam was represented by a steel bar. The strap connecting the pendulum with the flexbeam was of the same material (spring steel) and dimensions as the full-scale strap. The third tool is again an analytical model of the pendulum/flexbeam system, but now the strap is described as an elastic system. This last tool is used in the stability analysis of the wind turbine as a whole, and it will be described in the section "Stability Analysis". The tools are developed in sequence to have the more complex tool confirm the simple one.

Figure 4. Frequency Diagram - Production Blade

Figure 5. Frequency Diagram - Projected Demonstration Blade

Analytical Design

The simplest analytical tool can be explained with Fig. 6, which shows the geometry of the pitch change mechanism. The wind turbine rotates in a counter-clockwise direction looking into the wind. The blade of concern is at azimuth position ψ and the corresponding pendulum is advanced an angle ϕ in the direction of rotation. The pendulum is deflected through angle β_p into the wind and perpendicular to the plane of rotation. The blade flaps through the angle β_b (positive being away from the wind), where the angle is assumed not to be effected by the strap forces. The bending of the flexbeam is approximated by deflections about a virtual hinge line. When the flexbeam is forced to twist over angle θ , the force which causes this twisting moment has its reaction at the attachment point of the strap with the pendulum. The strap force established moment equilibrium about the pendulum hinge with the centrifugal and gravity forces acting on the pendulum weight. The strap is assumed soft in torsion and flatwise bending, but stiff in edgewise bending. In the design of the mechanism the pitch schedule for good wind turbine performance is the first objective. However, as has been emphasized in Ref. 1, the natural frequency of the pendulum is important from dynamic considerations. This natural frequency ω of the pendulum is derived in Ref. 1 and equals

$$\frac{\omega}{\Omega} = \sqrt{\left(\frac{e}{t+a} + \cos \beta_p \right) \cos \beta_p}$$

Figure 6. Schematic of Pendulum/Flexbeam Design

This frequency ratio is independent of the pendulum mass and it is chosen to be approximately 1.5P to avoid coupling with 1P aerodynamic and gravity forces. Since the pendulum flap angle will change with rpm, the hinge position and the pendulum length are the parameters to be determined. The hinge location, strap position, pendulum length, and pendulum mass were determined as a result of many parametric variations using the simple analytical tool. For the 28.5-inch offset hinge and the 30-inch pendulum length, the pendulum natural frequency is approximately 1.4P. Its weight moment is 1500 inch-lbs as compared to about 300-inch lbs for the blade about its own elastic axis. The effective inertia of the pendulum as seen by the blade is further increased as a result of having the strap attached inboard of the pendulum weight. An additional feature of the pendulum, by virtue of its leading edge connection to the blade, is its ability to provide pitch-flap (δ_3) coupling. Such coupling can have a significant influence in minimizing blade responses due to gusts. This is a technique often employed in helicopter rotors to restrict blade flapping.

After many iterations the pitch control schedule with tip speed is determined in terms of the variables of Fig. 6, $e = 28.5$ in., $l = 19$ in., $a = 11$ in., $m = 50$ lbs, $\phi = 45^\circ$, and a constant precone angle of the blade $\beta_p = 10^\circ$. The virtual hinge line is estimated 2 in. outside the hub, which has a 10-in. radius. The flexbeam torsional stiffness is represented by a torsional spring of 1127 ft-lbs/rad along l_b . The above parameters describe accurately the physical system as shown in the engineering drawings presented at the end of the report.

The resulting pendulum and pitch schedules with tip speed are shown in Fig. 7 with envelopes for the extremes of the angles. When the blade is at azimuth angle $\phi = 45^\circ$, the pendulum is horizontal and gravity has no effect on the pendulum flapping angle. It is in this position that the initial pendulum flapping angle is defined ($\beta_{p_i} = 45^\circ$). At low tip speeds, where the centrifugal forces are small relative to the gravity forces, the pendulum experiences cyclic flapping motion due to the gravity forces causing opposite moments at different azimuthal positions. This is seen in Fig. 7 by the large excursions of both the pendulum and the blade pitch angle below a tip speed of 125 fps. Above this speed the centrifugal forces become large compared to the gravity forces and only minor perturbations occur. The significance of cyclic pitch on blade responses is shown in Fig. 8 where the pitch excursions shown in Fig. 7 were applied to three low speed conditions. The maximum negative tip deflection is shown to be 20 in. which is well within the tower clearance, and the cyclic stresses corresponding to the tip deflections shown in Fig. 8 are less than $\pm 10,000$ psi.

The blade pitch angle in Fig. 7 is determined by the amount of flexbeam twist plus the bias angle due to the blade mounting to the flexbeam (blade is untwisted). The rated performance design point, blade pitch equal to zero at 150 fps tip speed, and the required starting torque at zero tip speed and low wind velocities, lead to the bias angle of -16° at the blade root.

One of the major parameters to be chosen is the pendulum mass. The higher the mass the closer the pendulum is to its plane of rotation position for a given tip speed. This means that most of the pitch schedule is executed at low tip speeds. This is undesirable for two reasons. At start-up tip speeds, the blade

pitch can become too high too soon, resulting in inadequate accelerating torque. At high wind and tip speeds the blade pitch will not increase sufficiently to drive the blade into a stalled, energy dissipating condition. A very low pendulum mass is not desirable either because the tip speed would become too high in high winds before stall occurs. High mass is generally desirable either because the tip speed would become too high in high winds before stall occurs. High mass is generally desirable from the blade aeroelastics viewpoint since it represents high impedance to restrict blade torsional excursions. In all cases the schedule has to include $\delta_{75} = 0^\circ$ at 150 fps tip speed for optimum performance at 12 mph.

The pendulum parameters of Fig. 6 are designed for easy adjustment to facilitate changes during demonstration tests. This holds in particular for the major parameters: a (the strap attachment point), $l + a$ (the pendulum length), m (the pendulum mass) and β_{p1} (initial pendulum angle). These adjustments would probably be eliminated in a production design.

Details of the design can be seen in the engineering drawings (Drawings 1 through 5). To verify the above analytic results, a full-scale experiment was devised.

Experimental Verifications

The experiment is shown in Fig. 9. The phantom image of the hub with blade pendulum and strap is superimposed on the photo to explain the lay-out. The analogy with Fig. 6 can be noticed easily. The full-scale experiment is stationary and does not rotate. A calibrated square rod substitutes for the torsional spring stiffness of the flexbeam. Its inboard end is attached as a cantilever with a vise against a heavy vertical machinery table. The outboard end is instrumented for torsional deflections with strain gages. A single sheet of $1/8 \times 4 \frac{1}{2}$ in. C-1075 heat treated spring steel forms the connecting strap between the blade and pendulum. The hinge offset, the pendulum length, pendulum mass, and strap attachment point agree with the full scale. The pendulum is hinged with pillow blocks relative to the table and its motion occurs in a vertical plane.

Vertically above the pendulum is positioned an overhead-crane scale and a very long nylon line to the pendulum mass. The pull of the hoist line indicated on the crane scale simulates the centrifugal force on the pendulum mass. With the hoist control the pendulum flapping angle can be adjusted, while keeping the hoist line vertical. The substitute flexbeam will twist with the pendulum flapping as is indicated in Fig. 7 with a dashed line. By virtue of the long nylon line, the hoist line pull is nearly constant for small deflections of the pendulum flapping angle. This allows for approximate measurements to be taken of the natural frequency of the system. The results of the experiment are shown in Fig. 10. The torsional deflections of the substitute flexbeam have been held under 21° because of the elastic limit of steel. The pendulum angle is measured directly with an inclinometer. The substitute flexbeam twist is determined from the strain gage readings displayed on a visicorder and from use of data from previous torsion calibrations of the flexbeam. The hoist pull is translated into a "Derived Tipspeed" by approximating the pendulum by a rotating 50 lbs weight and centrifugal force equal to the hoist pull.

Figure 7. Blade Pitch Schedule

Figure 8. Tip Deflections at Start-up

Figure 9. Pendulum/Flexbeam Experimental Set-up

The frequency measurements are done for very small amplitudes, to eliminate as much as possible the spring in the long nylon line. The frequency is measured from the torsion strain gages. The pendulum angle and flexbeam twist show the smoothness of the mechanism with tip speed, and show a good agreement with the simple geometric results as given in Fig. 7. In Fig. 7 the experimental data are shown with dashed lines after having the twist biased with an angle of 16.3° , to match zero pitch at 150 fps. The agreement is good for all tip speeds. It must be observed from Fig. 7 that the departures from the simple results at higher tip speeds are due to the increasingly elastic behavior of the strap. The pendulum flaps to smaller angles and the flexbeam twists less than that expected due to the strap deflecting a small amount. In Fig. 10 the frequency response is shown of the pendulum/flexbeam mechanism without the rotational inertia of the blade about the feathering axis, which is negligible in comparison with the pendulum inertia. At low tip speeds the flexbeam natural frequency dominates, while at high tip speeds the spring caused by the centrifugal forces dominates. The effect of the nylon line on the frequency results becomes less negligible for higher tip speeds. It follows that the simple experimental tool has confirmed the design and established the mechanical integrity of the strap.

Figure 10. Test Results of Pendulum/Flexbeam Experiment

Figure 11. Velocity Duration and Power Density Curves, $V_{mean} = 12$ mph

Performance

The wind turbine performance can be divided into three general areas: startup, normal operating range, and high speed operation. The primary concerns during these operating regimes are that 1) the initial accelerating torque be sufficient to bring the wind turbine rpm up to operating levels, 2) the pitch schedule provide the correct setting to achieve maximum performance during normal wind speed conditions, and 3) the pitch settings be sufficiently high to produce separated (stalled) flow over a major portion of the blade under extreme wind conditions. This last operating regime provides a method for dissipating energy to avoid overspeeding. The performance characteristics for each of these regimes are reviewed in this section, following a brief description of the wind characteristics for a 12 mph wind site.

Wind Characteristics

Figure 11 shows a typical velocity duration curve of NASA Lewis for a mean wind speed of 12 mph at a reference height of 30 ft. The velocity duration curves depend on the site and the mean wind speed. However, for sites with the same mean wind speeds, the site effect is a lesser parameter. Therefore, a typical velocity duration curve is useful in expressing how many hours per year the wind velocity is at least the velocity indicated, regardless of the site. A typical feature of the curve is that the mean velocity, or higher, occurs for exactly half of the one year period. The velocity curve is raised to the third power and multiplied by a factor containing density to obtain the power density curve in watts/ft². The area under the power density curve is a measure of the total energy in the wind during one year at the typical site, and for this curve amounts to 78.2 kW-hr/ft².

At heights other than the reference height of 30 ft, the mean wind speed will vary due to the shear layer adjacent to the ground. Surface conditions cause a wind velocity profile which is found to be described by $v/v_1 = (h/h_1)^n$ where v and h are the wind velocity and height at an arbitrary point and v_1 and h_1 refer to the reference height. Exponent n depends on the terrain and is conservatively estimated for maximum shear at 0.2. The profile of the wind shear is shown in Fig. 12.

A further effect on the wind environment as seen by the turbine is caused by the tower itself. The 12 inches outer diameter support column is guyed with 1-1/8 inch diameter cables. The wake of the column at Reynolds numbers ($Re = 10.3 V$ with V in mph) below 400 shows a vortex street with a width of 1.25D (column diameter). This is less than the 15 inch chord of the blade. The corresponding velocity defect of 20 percent locally in the wake of the tower diminishes quickly within a few column diameters if Re exceeds 400 (see Ref. 7) and the width of the wake decreases. Since the guy wire diameter is much smaller than the column diameter, the above effects are negligible in the aerodynamic calculations of the turbine.

There are generally three constraints as to the conversion of wind energy. First, the wind turbine cut-in level (generator idling) of about 8 mph which corresponds to 3 watts/ft²; second, the generator power limit of 15 kW corresponding to 11.9 watts/ft²; third, the structural limit of the system as a whole under high wind speed conditions; however, this limit is not applicable here. These

Figure 12. Wind Shear Profile for Wind Velocities of 12 and 60 mph at the Hub

limits are shown in Fig. 11 and they reduce the recoverable energy to 43 percent of the original level. This does not consider system efficiency between the turbine shaft and the output power which would further reduce this percentage. In the event that the demonstrator wind turbine were erected at UTRC (Rentschler Airport, East Hartford), a brief study of the wind conditions in this region was conducted. These results are presented in Appendix 1.

Starting Characteristics

From two-dimensional airfoil data it is assumed that sufficient blade lift can be generated at an angle of attack of 75 deg to overcome the static friction in the system to initiate rotation. As rotation begins, the section angles of attack will reduce rapidly with a corresponding increase in lift-to-drag ratio, and thus an increase in accelerating torque. Figure 13 shows the accelerating torque of the rotor as a function of wind speed for conditions when the blades are static ($\Omega R = 0$) and during normal rotation. For the curve where the tip speeds are indicated, the blade pitch begins at -16 deg at $\Omega R = 0$ and increases, according to the schedule of Fig. 7, to $\theta = 0$ deg at the design point of $\Omega R = 150$ ft/sec at 12 mph.

Figure 13. Start-up Torque Characteristics

Normal Operating Range

Typical wind turbine performance curves showing output power versus blade pitch for constant advance ratio generally indicate maximum performance at about $\theta = 0$ deg. Beyond the peak performance, the blade sections have exceeded their maximum lift-to-drag ratios and eventually begin to stall as pitch is further increased. The performance curves for the subject design, as calculated using the analysis of Ref. 4, are shown in Fig. 14. The peak performance points are shown for the 12, 15, and 18 mph cases; however, only the high pitch portions of the 27 and 60 mph curves are shown. Power is shown to drop precipitously as pitch is increased due to the increasing extent of blade stall. The operating line is shown as a dashed curve. This curve is simply the locus of points which satisfy the pitch schedule of Fig. 7. The variation of tip speed with wind speed is shown on Fig. 15, and it is expected that the tip speed will not exceed the maximum shown, 500 ft/sec. The rotor thrust and torque are shown in Fig 16 where the torque is shown not to exceed the generator rating of 15 kW. the results shown in Figs. 13 to 16 were generated assuming no wind shear or tower shadow. Wind shear effects are believed to have a negligible effect on rotor performance since the wind variation at the height encountered by the blades is relatively linear and therefore should not influence performance when using the average velocity at the center of the rotor. The tower shadow was neglected since the frontal area is only about 2 percent of the disk area and would not influence blade aerodynamics significantly. However, wind shear was used for the loads calculations presented later in this report.

Figure 14. Performance Characteristics

Figure 15. Tip Speed Variation with Wind Velocity

Figure 16. Rotor Thrust and Torque Characteristics

High Wind Speed Operation

Under high wind speed conditions the wind turbine rotational speed would increase in proportion to wind speed if the pitch angle were left at its optimum value of 0 deg. This would result in tip speeds approaching 1000 ft/sec and rotor thrust levels as much as fifty times the thrust at the design point. Also, coalescence of critical frequencies would be difficult to avoid due to the wide range of rpm. Needless to say, it would not be cost effective to design for these conditions, and thus another solution was sought.

Wind turbines such as the DOE Mod 0 have variable pitch blades, which can be controlled remotely and feathered to 90 deg to avoid overloading. Such a control, however, represent an additional expense and requires an operator or sensing system to activate it. The primary goal with the UTRC/ERDA wind turbine is to achieve a low cost system which is completely self-regulating and requires no operator or sensing equipment.

The principal reason for blade failures in high wind speeds is the fact that the blades simply are not sufficiently stiff to restrict their responses. It has been shown that adding structural stiffness is not the answer since excessive weight and dimensions would be required. Such an approach would probably result in a more expensive system than one having sophisticated control system to feather the blades. However, blade stiffness can also be achieved through the centrifugal force field, by increasing rotational speed. For example, at 150 fps tip speed the equivalent flatwise spring of the blade is about 8000 ft-lb/rad, however at

500 fps the flatwise spring increases to about 40,000 ft-lb/rad, five times stiffer. In other words, for the same aerodynamic load, the tip deflection at 500 fps would be only one fifth of the deflection at 150 fps.

The technique of increasing stiffness through higher rpm must be examined closely for possible instabilities arising from lowly damped blade modes coalescing with tower frequencies or excitation frequencies. The notion of allowing the rotor to increase rpm naturally at its optimum blade pitch setting of 0 deg was first studied. As wind speed increases, the tip speed would be expected to increase proportionately (assuming the generator could not absorb more than 15 kW) until the blade drag rise due to Mach number effects balanced the driving torque. However it was found that in a 70 mph wind this did not occur until about 850 fps tip speed, which would place the first edgewise frequency at or near 1P, which would not be acceptable. Increasing edgewise stiffness would compromise the total design, and in addition, at 850 fps excessive tower forces would be experienced, which would require a large and costly structure. If the maximum tip speed could be held to approximately 500 fps, thereby still providing adequate rotor stiffness, the edgewise frequency would be safely above 1P (see Fig. 5) and the thrust load on the tower would be about one-third the load at 850 fps.

It was decided to attempt to achieve torque equilibrium at 500 fps by increasing the blade section drag coefficient by means of stalling. If the pitch schedule with rpm was such that the blade angles of attack over a portion of the blade exceeded the stall angle, the drag force over this region would increase drastically and immediately reduce the driving torque. The rpm would then stabilize at a point, where the driving torque would equal the drag torque plus any output torque being absorbed by the generator. The performance results were given in Fig. 14, where the operating line was shown to intersect the high speed power curve at a pitch angle of about 5 deg. Due to the negative aerodynamic pitching moments on the stalled blade sections, it is necessary to schedule the pendulum to provide a pitch angle of 6 deg at 500 ft/sec. This is then reduced to 5 deg under the influence of the aerodynamic pitching moment. Speeds beyond 60 mph were not investigated, however it is expected that the tip speed would not vary significantly from 500 ft/sec. Since higher wind speeds would initially increase the angle of attack of the blade sections, causing greater stall, the greater negative aerodynamic pitching moment brought on by the higher angle would reduce the pitch, thereby returning the angle of attack nearly to its original value. The net effect would be only a minor change in torque.

Greater insight into the impact of operating at a pitch setting of 5 deg in the overspeed condition is gained by comparing the blade aerodynamic characteristics at this condition with the design condition at 12 mph. This comparison is presented in Fig. 17 where the radial distributions of drag coefficient, Mach number, and angle of attack are shown for the two conditions. The significant increase in these aerodynamic parameters at the overspeed condition is evident from this figure.

Figure 16 showed the thrust levels for wind speeds up to 60 mph. Normal operating thrust at 150 fps tip speed is approximately 400 lbs, and the maximum thrust attained would not be expected to exceed 5000 lbs at 60 mph. Above this speed deeper stall is experienced with little change in thrust.

Figure 17. Blade Section Characteristics at 12 mph and 60 mph

A special case of high speed operation of the wind turbine at low wind speed would follow from a failure in the step-up gearing, such as belt failures. At constant windspeed the wind turbine will accelerate because of the reduced load. Figure 15 will no longer apply because none of the output power is absorbed. However, since blade stall is primarily an angle of attack phenomena and not a dynamic pressure phenomena, all the above will hold. The blade rotation will be limited through stall, but now at a much lower thrust.

Blade Loads and Stresses

The G400 blade aeroelastic analysis (Ref. 8) was used to determine the rotor load and stress characteristics. In this study this analysis utilized two flatwise modes, one edgewise mode, and one torsion mode to simulate the blade structure, and solved the fully coupled equations of motion for the separate modal responses. For these calculations, the hub was assumed fixed (except for rotation) and the inflow was made up of the sheared wind velocity and a steady induced velocity determined from momentum theory. The hub degree of freedom was not considered critical to the blade design parameters unless the stability analysis should reveal a dynamic magnification due to coupling effects. This will be covered later in the stability section of this report.

Since the blade is considered "low stiffness" relative to many wind turbines employing large amounts of taper, the critical stress location is at the root of the blade where the major curvature occurs. The blade was designed to withstand the loads which occur under the high wind conditions. The procedure taken was to determine the blade loads assuming nominal stiffness and mass distributions which provided the desired frequency characteristics, and then to design the critical load paths accordingly. For a production design the critical load point would be the blade root. However, for the demonstration design, in addition to the blade root, a somewhat complicated joint is required at the flexbeam/control strap/blade interface which must also carry the blade moments.

The low wind speed conditions are shown in Figs. 18 and 19 where the azimuthal variation in blade root stress is shown for wind speeds of 12, 15 and 18 mph. The 1P cyclic stresses are due to the wind shear and gravity excitation; however, the stress levels are low. The maximum vibratory stress is less than ± 5000 psi which is only a small fraction of the graphite/epoxy allowable of 55000 psi (Ref. 9). The most critical aspects of operating in a partially stalled condition is blade stability and stress characteristics. The ability to simulate the highly unsteady environment of stalled flow is extremely difficult, and in this preliminary study only 2-dimensional steady-state airfoil data were used. It is believed that the aerodynamics in these high speed stall conditions are sufficiently complex that no available analysis would be capable of predicting blade responses with a high degree of confidence, and that in the final analysis only full-scale tests can establish the feasibility of such operation. An indication of the general stability and stress levels can be gained, however, by an analysis using steady-state data, and it was only this level of effort that was planned under this contract. The

Figure 18. Flatwise Stress Time Histories

Figure 19. Edgewise Stress Time Histories

results of the G400 analysis using steady-state data are presented in Fig. 20. The maximum vibratory stress is in the flatwise direction; however, the magnitude is only $\pm 20,000$ psi which is well below the allowable of $\pm 55,000$ psi (Ref. 9). These allowable stresses follow from fatigue tests and imply an infinite fatigue life. The radial distributions of flatwise and edgewise bending moment for this

Figure 20. High Speed Stress Characteristics

Figure 21. Steady Blade Moment at 60 mph

condition are shown in Figs. 21 and 22 where the steady and vibratory moments are presented versus blade station. The peak edgewise moment is determined by adding the steady and vibratory root moments which results in a moment of 86,000 in.-lb. The major concern for this condition was the blade joint which is located from approximately 50 in. to 70 in. For the demonstrator blade this joint is necessary

Figure 22. Vibratory Moment at 60 mph

to allow for relative adjustments between the flexbeam, strap, and blade, however, the design resulted in several bolted connections. The design shown in the engineering drawings was conservatively obtained by doubling the combined load of Figs. 21 and 22 to account for gusts and then applying factors as normally practiced in the Design Branch at Hamilton Standard. The maximum load in the blade proper occurs just outboard of the joint. This load, at 70 in., amounts to 49,000 in.-lb and converted to stress is equivalent to 5800 psi at the trailing edge. The blade tip deflections for the 60 mph condition are shown in Fig. 23. The maximum flatwise deflection towards the tower guy wires is only 8 in., which reduces the clearance from 48 in. to 40 in. The sign conventions for tip deflections, as well as stresses, are: positive flatwise -- away from the tower, and positive edgewise -- in the direction of rotation.

The blade torsional stresses are negligible compared to the stresses in the flexbeam where essentially all torsional deflection occurs. The 45 deg fiber alignment in the fiberglass portion of the blade results in an extremely rigid structure in torsion. The maximum torsional moment occurring at the most inboard point of the fiberglass blade as a result of summing the outboard moments is approximately 800 in.-lb. For the fiberglass spar and shell structure this would produce an insignificant shear stress of only 400 psi in the 0.050 in. wall of the shell. This moment can also be seen as that which is required to untwist the flexbeam from its inputted value of 6 deg (i.e., the pitch setting imposed by the pendulum at 500 fps tip speed) to a value of about 3.5 deg as seen in Fig. 24 at an azimuth of 210 deg. The pitch deflections at the lower speeds are also presented on this figure.

Figure 23. Tip Deflections at 60 mph

Figure 24. Blade Pitch Variations

SYSTEM DESIGN

Tower

In addition to providing an adequate support for the wind turbine itself, the tower must satisfy other requirements. Its structure should cause the least obstruction to the wind, thereby minimizing its aerodynamic input to the rotor. The natural frequencies of the structure should not coalesce with turbine frequencies, and the handling of the structural parts during shipping and erection should be accomplished without the need for extraordinary support equipment.

The height of the tower follows from the simple shipping requirements that no part shall be longer than the turbine radius. Two of those parts make the total height of the tower equivalent to the turbine diameter. A tower of that height is most economically designed as a guyed column. This is depicted in the engineering drawings which show the layout of the wind turbine and the tower layout. The structural integrity of the support structure is totally determined by the high wind condition and the erection procedure forces. At the high wind condition of 60 mph, the maximum thrust developed by the turbine is 5,000 lbs. To account for gusts and over-speeding, it is assumed the total thrust can reach 10,000 lbs. With the guy wire angle at 110° with the tower, the single cable load is 52,400 lbs and the tower compression load is 53,900 lbs (including nacelle weight). When the wind turbine moves through the initial arc of the erection as shown in Fig. 25, the

Figure 25. Erection Procedure

lifting cable force is 16,700 lbs, while the tower compression is 24,450 lbs. The lifting cable pulls asymmetrically relative to the tower, thereby keeping the guy wire, which is in line with the hinge line of the tower base, under tension for stability. Using a safety factor of two for the guy wire strength, the 50 ton 1-1/8" 6 x 19 wire center galvanized steel bridge strand cable was chosen. The three guy wire sets make the tower an isotropic structure. To remain so under load, no cable should become slack under the highest load. This requires a pretension in the cables of 50,000 lbs. However, it can be seen, that this reduces the safety factor to one. It is assumed that the cables behave as linear springs. To eliminate very heavy ground anchors, the pretension requirement is relaxed to 5,000 lbs, which makes the structure isotropic to just above the rated wind speed. With a safety factor of five on the stability of the structure, which is more critical than the strength, two pieces of 12" OD 1/4" wall carbon steel with 125 number ASME plate steel flanges comprise the central compression member of the structure. Weather protection is obtained through paint.

The natural frequencies of the support structure have been calculated in two ways. The frequency of a simplified model with a concentrated turbine and nacelle mass on top of a single stage guyed tower is derived in the midterm report of this project, Ref. 10. The frequency of the isotropic structure in bending was calculated to be 2.4 cps. The final nacelle design had different mass and inertia characteristics and a different guy wire configuration, and therefore a more accurate analysis of the frequencies was necessary.

The more accurate computer analysis, NASTRAN, is described in Ref. 11. The structure under study is divided into finite elements of bars and rods. The bars allow bending and elongation (central column) and the rods allow only axial elongation (guy wires). Half of the mass is at each end of the element. The frequencies were computed to be 4.14 cps for for-and-aft deflections along the turbine axis, and 4.25 cps for lateral deflections of the nacelle with the nacelle free in yaw. In the NASTRAN analysis the turbine mass is lumped together on the axis of rotation. This increases the inertia of the turbine and therefore the calculated frequencies are lower than in the case where turbine blade elasticity is considered. Frequencies and mode shapes with elastic blades are not calculated because the results would depend on the blade position in the NASTRAN analysis which makes the results difficult to interpret. At 500 fpa tip speed or with 1P corresponding to 3.98 cps, the tower frequency is sufficiently displaced from the lowly damped lateral frequency to prevent coupling. The proximity of the 1P with the fore-and-aft frequency is not expected to pose a problem since there is significant aerodynamic damping in this mode. The next higher natural frequencies of the tower are 13.39 cps (for-and-aft) and 14.17 cps (lateral). For the above mentioned four natural frequencies the normalized mode shapes for unit deflection are given in Fig. 26. The low frequency mode shapes of longitudinal and lateral motions are nearly the same. The mid-tower guy wire will experience less periodic stresses than the top guys. The next higher frequencies give longitudinal and lateral mode shapes which are less equal. Because the moment exercised by the nacelle on the top of the tower, which is felt primarily in the longitudinal motion, the longitudinal mode shape has the different shape. Clearly the effect of the mid-tower guy wires is shown in the mode shape.

Figure 26. Tower Mode Shapes for Unit Deflection

The design of the tower foundation is based on one compression anchor under the central column and three tension ground anchors at the guy wire ends. The compression anchor is designed for 53,900 lbs concentrated load on top and a reaction pressure at the 4' x 4' of 23.3 psi, which is lower than the allowable bearing capacity of firm fine sand, compact inorganic silt, firm sand-clay soils, or medium clay (see Ref. 12). Soil compacting or any other soil except loose fine sand will cause higher allowables. It is assumed that the underlying stratum has an equal or higher bearing capacity than the soil on which the foundation rests. The tension anchor whose shafts have the same 11° top angle as the guy wires have to be firm for 50 tons. The standard building code solution for piles in uplift requires three deep piles under different angles, which converge on the surface. However, such solution is not justifiable for this application. In general, no unique solution can be suggested for all soil conditions. Inexpensive solutions to consider are: stock anchor with folding flukes, which is driven in and then pulled out the length of the fluke, ground plate with rod, which has to be dug in, or a concrete anchor block with rod.

The guy wire arrangement is chosen to reduce the slenderness aspect of the central column. The spreaders at the mid-tower location preserve the rotor clearance in case of whipping cables and also divide the top guy wires in equal parts, whose lengths are equal to the length of the mid-tower guy wire. The top guy wire and mid-tower guy wire are attached with a three bolt plate to the eye of the turnbuckle. The opposite eye of the turnbuckle is hinged to the ground anchor shaft.

Hub, Shaft and Main Bearing Design

The absence of heavy blade retention hardware due to the continuous carbon epoxy flexbeam leads to a simple clamp-like hub, made of two parts, the inner part

having a flange which is mounted to the main turbine shaft (see engineering drawings). This main shaft length is determined by the turbine blade clearance near the tower wires. To reduce the bending moments in the shaft the supporting ball bearings are located as close as possible to the location of the forces acting on the shaft. At the hub end of the shaft, acts the turbine thrust force, the in-plane turbine force, the pitching moment, and the axial driving moment. The bearing nearest the hub works as a free floating self-aligning bearing and therefore reacts only with forces perpendicular to the turbine shaft. The other end of the main shaft reacts also to the forces from the step-up gear pulley. These forces are the weight of the pulley and the synchronous belt tension. The bearing nearest this pulley is self-aligning and reacts the forces of the turbine moments, except the driving moment, the turbine thrust force, and the pulley forces. In the following, the structural aspects of each of the above components will be discussed.

The two part hub is mounted to the main shaft with two bolts through a sleeved flange. The octagonal flange can be forged steel for high volume production, but as a single item here it is machined from solid stock. The flexbeam is clamped against this flange with a grooved hub cover, and the hub cover is a machined part. On the flange side of the hub cover is machined the recess, which leads the locally fiberglass wrapped flexbeam through. The fiberglass wrapping protects the carbon epoxy fibers against surface roughness and damage while fitting, lowers the tolerance on the recess dimensions, and serves as rounding for the edges of the recess ends. Across the other face of the 1.5" steel plate hub cover is mounted the pendulum hinge offset bar. Through the center of the offset bar, through the center of the hub cover and also through the center of the flexbeam is mounted a 7/8" bolt, which centers the whole assembly during mounting. The hub cover itself is centered against the flange with the bolts of the hub through high tolerance holes. The bolts are loaded in shear because of the driving torque and in-plane turbine force and in tension because of the pitching moment of the turbine thrust.

From the G400 aeroelastic analysis results of Figs. 21 and 22, it was shown that the maximum edgewise and flatwise moments occur at the root of the blade at 60 mph wind speed. The calculated flatwise moment is 19,000 in.-lbs, the edgewise moment is 86,000 in.-lbs and the thrust is 5,000 lbs. With the allowable stress in carbon epoxy of 55,000 psi, the maximum nondestructive flatwise moment on one blade can become 28,600 in.-lbs. The large edgewise moment is nondestructive and can be higher based on flexbeam strength. However, this is much of a theoretical value, because in the analysis a constant rpm is assumed. The above edgewise moment is nearly all oscillatory because the steady driving edgewise moment is low because of blade stall. The oscillatory part of the edgewise moment sees the shaft rotating at constant speed with infinite inertia. In reality, the inertia of the shaft, hub, drive and generator will forestall such edgewise moments, unless resonance occurs. Moreover, the key of the first pulley in the step-up drive shears at the maximum belt drive torque of 45,000 in.-lbs.

The hub cover is attached to the shaft flange with eight 7/8" bolts and nuts. It is conservatively assumed that not four but the two bolts next to the flexbeam restrain the flexbeam. The loads follow from: differential tension because of the 22° flexbeam twist, tension due to the thrust load, tension due to the maximum flatwise moment and shear due to the above edgewise moment. Maximum shear is 1800 psi while at the worst, bolt tension is 12,900 psi.

The two bolts, which attach the hub flange to the shaft, experience shear only. The shear at the interface of the flange should be just larger than the allowable shear of the key at the other end of the shaft (not yet counting shear due to the thrust of 5,000 lbs). The sum of the maximum driving torque possible and the thrust gives 19,300 psi shear at one bolt (shaft diameter 3-15/16").

The position of the main shaft bearings can be seen in the nacelle components drawing. To determine realistic forces on the bearings, more forces and moments have to be applied to the hub. More conservative than the flatwise moments is the assumption of a temporary deflection of 7.5° of the turbine tip path plane. The thrust vector component causes a force perpendicular to the shaft of 1070 lbs and a moment about a shaft of 14,600 ft-lbs. Considering further the weights of the turbine and the shaft pulley (vectorially summed with the belt pull) the forces on the two main bearings can then be calculated. Because of the rotating nature of some of the moments and forces, several combinations need to be considered. The maximum radial force on the bearing nearest the hub is 2500 (660) lbs. The forces on the other bearing are 3520 (1400) lbs radial and 5000 (1050) lbs axial. The bracketed quantities apply to the 18 mph normal operation of the wind turbine and should be withstood in continuous operation for 20 years. The high wind speed quantities have an assumed duration of 50 hours per year (see Fig. 11).

The main shaft is loaded heaviest at the bearing near the hub. The stresses in the shaft at the bearing location are caused by the blade edgewise moment which produces torsion, blade thrust, shaft bending due to hub moments, and the inplane force associated with tip path tilt. Since the torsion loads are most critical and bending loads next, a hollow shaft is chosen. Not only is material more efficiently used, but the strain gauge wiring can be accommodated as well. Based on the above conservative forces, a carbon steel C-1015 tube of 4" ID and 0.625" wall is chosen. Locally the outer diameter is turned down to 3-15/16" OD to assure concentricity and to fit the bearings (see next paragraph). With this outer diameter and the reduced wall thickness, the stresses in the shaft at the hub bearing location are 11,000 psi tension and 36,200 psi shear. It follows that the torsion would drive the shaft just to the yield limit. It should be realized, however, that the applied torsional moment follows from the in-phase two edgewise moments at 60 mph. This torsion is only counteracted by the 45,000 in.-lbs of the pulley key. Under this torsional load the maximum shear would be reduced to 4930 psi.

The bearings are nearly totally determined by the shaft outer diameter of 3-15/16". The hub bearing is intended to be free-floating and self-aligning. When the nacelle bends, the axial distance between the hub bearing and the pulley bearing changes. If the hub bearing were not free floating, this would lead to axial bearing loads. The thrust axial load is transferred to the nacelle and then to ground along the shortest path by the pulley bearing, which is self-aligning. The commercially available pillow blocks for this shaft diameter are too heavy and too voluminous to be considered. To reduce mounting errors and to decrease spare parts inventory two identical bearings are chosen: namely, the SKF bearing #23122K + SNW3122 + W33 spherical roller bearing. The first number is the standard bearing number. The second number is the adapter number. The use of an adapter eliminates the pulling of a bearing over long distances along high tolerance surfaces. The adapter allows to bypass small surface irregularities and eliminates special tools, which is highly desirable under field conditions. The third number indicates a

lubrication feature of the bearing. It would appear that the choice of identical bearings eliminates the free-floating characteristic of the hub bearing, however, this is not the case since the free-floating characteristic is approximated in a mechanical way, as will be discussed later. The application of the manufacturer's guidelines to the loads on the bearings with grease lubrication yields; hub bearing 37 years of life under 60 mph winds, pulley bearing 5 years of life under 60 mph and 90 years under standard conditions. The over dimensioning of the bearings is merely due to the shaft diameter. The bearing's maximum permissible misalignment is ± 2 degrees. Weather protection for the bearings is obtained through seals, which ride on the shaft and which are connected flexibly to the mounting blocks.

Step-Up Drive Design

In Ref. 10 the commercially available solutions for the step-up gearing have been listed and it appears that the synchronous belt drive is the cheapest and the lightest solution. The belt drive provides very smooth and quiet running and it can absorb appreciable shock. Of the belt drives the synchronous belt drive is most efficient because it has no slip and the deformation energy for the flat belt back is minimal.

The desired gear ratio follows from the performance analysis covered earlier. At 12 mph the turbine tip speed is 150 fps. At 18 mph the turbine tip speed is 250 fps while extracting power, this equals 120 rpm. The design speed of the generator is 1800 rpm. Therefore, a gear ratio of about 1:15 is required. This is theoretically possible in one step, but the centerline distance of the pulley becomes impractical to have enough teeth engaged at the smallest pulley. Also the stability of the belt becomes a problem with large centerline distances. For simplicity, ease of repair, spare parts inventory, etc., the gearing is made in two equal steps of 1:3.69, so that the total gear ratio becomes 1:13.62. The layout of this drive is shown in the nacelle component drawing at the end of this report. The 96 tooth pulley is attached to the main shaft. The synchronous belt is also wrapped around the small 26 tooth pulley at the intermediate shaft. At the intermediate shaft is also attached another 96 tooth pulley. Its belt drives a 26 tooth small pulley at the generator shaft. The 15 kW generator centerline coincides with the main shaft centerline.

From the Wood's Company mechanical power transmission equipment literature follow the 1.3 service factor for generators and the 0.4 additional service factor for step-up ratios greater than 3.5. The total design horsepower becomes then 45.6 HP. The belt pitch is determined by the power and two speeds of the first and the second step-up. The 7/8" pitch XH belt satisfies both conditions. The desired gear ratio was established with 96 XH to 26 XH tooth pulleys. A number of belts fit around the pulleys for specific centerline distances. For structural reasons of the nacelle the centerline distance of 27.57" is chosen. This determines the belt number 112 XH. The drive belt width follows from horsepower and rpm data. The first, as well as the second step can be equipped with the 4" belt width. The Wood's Company 112 XH 400 belt drives the 96 XH 400 and 26 XH 400 pulleys. The pulleys are mounted on the shaft with split conical bushings for easy assembly. The bushings can be standard or reverse mounted. In the engineering drawing it can be seen that the small 26 XH 400 pulleys are equipped with

flanges to keep the belt in track. The small pulleys with their bushings have to be removed to change belts. The belts can pass around the generator and the structure. Belt changes are the only maintenance on the drive system and they are expected to be less frequent than annual.

The forces on the shafts due to the drive are the drive torque, the pulley weight, and the belt force. The torsion and bending of the main shaft at the pulley location have been considered. Acting at the first step-up side of the intermediate shaft are the 36 lbs pulley weight and the 960 lbs belt force. The maximum torque which the pulley can handle is 11,000 in.-lbs, which corresponds with 77 HP versus the desired 45.6 HP. Torsion and bending due to the belt force require a carbon steel C 1015 shaft of 2-1/2" OD. The shear is then 14,400 psi in the intermediate shaft. Acting at the second step-up side of the intermediate shaft are the 146 lbs of the 96 XH 400 pulley weight and the 260 lbs belt force. Since the belt force is smaller and opposes the pulley weight, the intermediate shaft is over-designed for these conditions. Since the generator can be driven by V-belts, which have higher belt forces, the generator shaft bearing is not critical.

The vectorially summed loads on either side of the intermediate shaft cause only radial loads in the two pillow blocks chosen to support the intermediate shaft. The pillow blocks are self-aligning and one of the two is free floating to eliminate axial loads due to mounting tolerances. The first step-up pillow block has a 1995 lbs radial load and the second pillow block has 935 lbs load. For simplicity of assembly, and to have identical base-to-centerline heights, both pillow blocks are chosen from the same series. Each is a spherical roller bearing pillow block by SKF with pillow block number SYR 208-3 (free floating) and number SYR 208-12 (fixed). The maximum permissible misalignment is 1.5 degree. The highest loaded bearing with grease lubrication has a life expectancy of 12 years under continuous 15 kW power conditions and a 7.6 years life under turbine over-speed conditions (intermediate shaft at 700 rpm) and assuming that 15 kW is still generated. The fixed or free-floating characteristics have no effect on bearing life in the absence of axial loads. It should be understood, that all bearings in the design are shielded against the elements, either by their standard shields or by external shields between bearing housing and shaft.

To minimize the length of the intermediate shaft, while still leaving space for mounting a clutch or brake to the large pulley in the first step-up, the pillow blocks are mounted with the shaft locking collars at the pulley side. This makes it necessary to reverse mount the bushings of the intermediate shaft pulleys, i.e., when removing the pulley the bushing needs to be removed first.

In case this demonstrator wind turbine is being used for long periods or is unattended, it is desirable to enclose the step-up drive system to prevent hail, freezing rain and large particles to come between the belts and the pulleys. Otherwise weather proofing is not necessary.

Emergency or repair shutdown of the mechanism applies only to the rotor, because it is necessary for the system to weather-vane. From the following section follows that the generator can slow down the turbine in wind speeds slightly above the rated speeds, through field excitation. Furthermore between the main shaft sprocket and generator sprocket, space is kept to mount the Warner Electric fail safe brake ER-1225 of 400 ft-lbs static torque.

Generator

The aerodynamic torque of the turbine is brought in equilibrium with the torque of the electro-magnetic fields in the generator. These fields revolve faster than the turbine by the constant gear ratio of 13.63 of the two step synchronous belt drive. Because of this, the generator performance will be shown as a function of the turbine tip speed or rpm.

Based on manufacturer's data and additional engineering data provided for this particular application, and for low maintenance, a brushless generator was chosen from the General Electric Company. The unit of concern is of the Type SJ, Brushless type generator #5SJ256AC213, 1800 rpm, 3 phase, 60 Hz, continuous, 40 degrees ambient, two bearings, dripproof, 240/480 Volts, 12 wires, class F insulation, voltage adjustment ± 10 percent, base mounted, and weighs 390 lbs. This generator will sustain operation down to 1/3 rpm at a constant kVA per Hz slope. The 260 lbs belt drive load at 2" from the generator is within the overhung limit. Voltage regulation in the control package on the generator allows for adjustment of the torque required curve. The linearity of the standard voltage regulation is better than 5 percent. However, even open voltage regulation will result in about 48V output because of residual magnetism in the exciter field. Therefore even at idle the generator requires a small input torque, which is higher than the bearing torque. The power and torque curves of this generator are given in Fig. 27 as functions of the generator rpm. The torque curve end points follow from manufacturer's engineering data, and are connected by a straight line because of the constant kVA per Hz operation. The power output curve follows from the generator rating and the assumed constant conversion efficiency of 82 percent. The curves shown are the best performance curves. It can be seen that the low speed power requirement is much exceeded. However through voltage regulation or through load management via a resistor bank the power output of the generator can be matched with the available turbine power. The generator will give variable voltage and variable frequency.

The torque to drive the generator for best performance as a function of tipspeed is shown together with the available driving turbine torque in Fig. 27. It is assumed that the belt drive is 98 percent efficient, which means that the torque seen by the generator is the turbine torque multiplied by the efficiency and divided by the gear ratio. It can be seen that the torque absorbed by the generator is larger than the torque produced by the turbine. This is due to the performance requirements of the wind turbine at a low, as well as a high, wind speed. The shape of the turbine torque curve is a result of the optimization of the turbine for the low wind velocity of 12 mph (150 fps tipspeed), which compromises the performance at higher wind speeds.

In Fig. 27 is also indicated the percentage of generator rpm corresponding with the tip speed. All generators are designed with the NEMA 25 percent safe overspeed capability, which occurs at a tipspeed of 346 fps. At this point there still is a safety factor on the mechanical integrity of the generator, the square root of which corresponds with the multiplier for the maximum possible generator rpm. The maximum possible tipspeed of the turbine is 500 fps. This is possible with a safety factor of 2.1 on the mechanical integrity of the generator. Since the generator wiring does not have the current carrying capability at these high speeds, the generator should not be excited under overspeed conditions. However this

Figure 27. Generator Performance

generator has been driven commercially at 3600 rpm after the addition of bracings to the field windings, which moves the NEMA safety condition to twice the rpm.

The generator is delivered standard, drip proof, with two sealed bearings, and moisture proof wiring. A totally enclosed fan cooled edition of the generator or a fungus proof wiring edition is not justifiable for the conditions experienced at the Hartford test site.

The basic schematic for the generator-exciter is shown in Fig. 28. The mechanical energy of the generator shaft is converted into electrical energy (3 phase) and consumed by the electric load, which may have varying impedance. A Delta-load rather than a Y-load is sketched because of its larger residual impedance in case of an element breakdown. Field power needs to be supplied to the exciter. This supply can come from the generator itself via a regulator or compensator or from an external direct current field power supply. The simplest form of

Figure 28. Generator-Exciter Schematic

supply would be by external field power. However it is in general desirable to have a feedback loop from the alternating current output to the exciter field. In such a feedback loop a compensator maintains a broad band of voltage regulation. It senses output current through a current transformers and line voltage through potential transformers. The total a-c feedback power is rectified to supply the exciter field. A voltage regulator works in addition to current and voltage feedback signals with a preset reference. A regulator offers a relatively flat regulation of the voltage over the load range and the reference voltage level is adjustable.

Nacelle Design and Yaw Mechanism

The design of the nacelle is intended to be simple and not to require close tolerances. The layout of the components is shown in the attached engineering drawings. All components are mounted on a U-shape bent 1/4" steel sheet which forms the backbone of the nacelle. Along the top surface are mounted the main shaft and the generator. The intermediate shaft of the step-up drive is mounted against the box beam, which is locally formed by welding the same U-channel to the original nacelle

part. Because of the size of the large pulleys in the drive, cut-outs have been required. The structure between the main shaft and the generator has been fortified with webbing. The hub bearing is mounted near the shear center of the channel end and because it is a floating bearing, the channel is subjected to bending primarily. The main shaft thrust bearing is mounted near the yaw axis to shorten the load path.

The yaw mechanism is simply a double supported hollow shaft. The aerodynamic forces on the turbine blades, the weight, and the inertia forces on the nacelle components and turbine parts are brought in equilibrium by the reaction forces on the yaw bearings. Two spherical roller bearings provide the yaw degree of freedom. Both bearings are contained in a single unit which is attached with a flange into the top of the tower. The top bearing carries the nacelle weight. There is a special arrangement for the attachment of the yaw shaft to the U-channel. The flanges bolted to the yaw shaft have a width which is smaller than the inside dimension of the U-channel. By properly locating spacers between the shaft and the U-channel, an asymmetry between turbine axis and yaw axis can be obtained if desired. The parallel walls of the U-channel further allow the turbine shaft pitch angle and the turbine overhang (center of gravity location) to be varied by simple remounting of the nacelle relative to the yaw shaft. If any asymmetry of the yaw axis and turbine axis is required to compensate for alignment with the wind or to help to cancel the driving torque component about the yaw axis in case of a pitched-up nacelle, then a very simple lightning protection is possible. Since the yaw axis and the turbine axis would not intersect, a lightning rod can be connected through a small hole in the nacelle and through the hollow yaw axis to the grounded tower in a stationary fashion. Until the nacelle location is determined in the field, lightning protection is not further resolved, as it does not pose a problem.

The yaw shaft is structurally critical at the top bearing location of the mechanism. Because of the rotating forces and moments, the most severe condition occurs only once per revolution. These are: 4598 lbs axial thrust, 7500 lbs shear and 26,800 ft-lbs bending moment. Because of the manner in which the flanges are mounted on top of the shaft, the wall thickness of the shaft is important. This determines the choice of a hot finished seamless carbon mechanical tubing of 6.5" OD with a 2 in. wall. To fit the bearings, the OD has to be turned down, which leads to the maximum tension of 16,500 psi.

The bearings are primarily determined by the yaw shaft diameter. To cause a more equal radial loading in the bearings, a 10 in. separation between the bearings was chosen. Both bearings are the same self-aligning spherical roller bearings manufactured by SKF. The designation is #23130K + SNW 3130. The first number is the bearing number, the second number is the adapter mounting number. Maximum permissible misalignment is 2°. The thrust bearing is the highest loaded. Since the yaw axis makes few if any revolutions per year, the standard SKF life expectancy formulas do not apply. The axis will oscillate in yaw about a point, which primarily changes with wind direction. This might lead to brinelling of the bearing race. Consultation with the manufacturer yielded a life expectancy of over 10 years in the absence of lubrication. From the drawing it can be seen that the bearings are adapter mounted to the yaw shaft, but can both slide freely in the tower mount.. This is to make the lower bearing floating. During normal operation, the top bearing will have positive thrust, while upside down (during assembly) the lower bearing will take the weight load after a slight axial displacement of shaft and nacelle.

For the nacelle attachment to the yaw shaft, the 6.5" OD 2.5" ID shaft has been machined with two parallel faces 4" apart (see drawing). Two one-inch flanges are bolted to the shaft with four 7/8" bolts, which are 3.82" apart tangentially and axially. Shear stress is 49,500 psi. The reduced cross section of the yaw shaft has a maximum tension of 36,900 psi. Via spacers, each flange is bolted to a side of the 0.25" U-channel with 5/8" bolts, shear 29,500 psi. The local compression in the U-channel is 22,300 psi. The flanged mount fits with radial play into the top of the hollow central column. The tolerance on the ID of the tower is too large to allow a snug fit. The bending stress near the top bearing is 31,750 psi. The flange of the housing of the yaw shaft bearings has the same bolt pattern as the standard pipe flange.

The main shaft thrust bearing is mounted in a block, which attaches to supports nearly directly above the vertical parts of the U-channel. In the same manner, but without the thrust shoulder, the hub bearing is mounted. This bearing is free-floating relative to the block. A steady 600 lbs downwards and a rotating load of 2050 lbs are the maximum loads experienced by the bearing when the nacelle is fixed. For vertical radial forces on the bearing, the U-channel is mounted essentially as a cantilever. Horizontal forces on the hub bearing will cause yawing motion. Since the moment of inertia of the turbine assembly is about three times higher than that of the generator, the inertia forces for equilibrium will primarily come from the hub, pendulum, and turbine itself. The centerline of the main shaft is 6.25" above the cross section of the U-channel. The bearing forces act 4" above the shear center of the hub side of the U-channel cross section. The hub bearing mount is identical to the thrust bearing attachment to the U-channel. The critical cross-section of the U-channel is just aft of the yaw shaft attachment. In torsion, where the moments are absorbed by bending of the flanges, the U-channel shows less than 4° twist under maximum load, with 30,000 psi tension.

The generator weight of 390 lbs and its reaction torque of 82 ft-lbs are supported by a structure, which passes inside the drive belts. Rigidity, not strength of the support structure is the design criterion for the structure upwind of the yaw axis. The belt life and transmission efficiency are strongly determined by the ability to preserve the alignment of the pulleys and to keep the centerline distance constant. The centerline height of the generator is also 6.25" or less with its tolerance on the housing. The wiring from the generator are lead along the inside of the U-channel past the corners of the internal webbing to the hollow yaw axis.

STABILITY ANALYSIS

Two computer codes were used in the design of the wind turbine: G400 and F762BR. Both of these codes use a modal approach for simulating the rotor aerodynamics and obtain time-history type solutions. The former is a single-blade, hub-fixed analysis whose blade dynamic description uses recent state-of-the-art modeling techniques. The use of this analysis, however, required simplifying assumptions to be made with regard to the interaction of the blade and pendulum. This analysis was the primary tool for the preliminary design of the rotor blade from a stress standpoint.

The latter analysis, F762BR, is a comprehensive, multi-blade, finite impedance hub analysis which, however, uses a less sophisticated blade dynamic description. The purposes of this analysis were (1) to provide a check on the G400 computed blade stresses, and (2) to check for any aeroelastic instabilities arising from the presence of the pendula and the associated flex-straps, from the free yaw feature of the nacelle, and from the flexible tower. The remainder of this section will first describe the F762 BR analysis in greater detail, and then present the results of the use of that analysis.

Description of the F762BR Analysis

This analysis is a variant of an analysis originally developed under Sikorsky Division funding for the ABC (Advancing Blade Concept) helicopter development program. This earlier analysis already had multi-blade, finite impedance hub capability and was subsequently modified to analyze wind turbines. Those features which were specifically incorporated in the computer code to analyze the pendulum controlled composite bearingless rotor are as follows:

Control Pendula

The nonlinear (large swing angle) differential equations of motion for each of the control pendula were derived and coded. The pendula were assumed to be excited by all the inertia loads associated with rotor speed and hub/nacelle accelerations, by gravity loads, and by the elastic loads generated by the deflected curved flex-strap. Additional (optional) provision was made for either locking out the pendula motions or for ganging them together.

Elastic Flex-Straps

The general computational approach in the F762BR computer program is to calculate mass accelerating forces from known system deflection and/or displacements, in order to calculate accelerations. From the accelerations the time-history solutions are obtained by numerical integration to obtain new deflections to complete the solution in a step-by-step fashion loop. Consequently, the appropriate dynamic description required for the flex-straps is a stiffness matrix giving the complement of 6 loads (3 forces and 3 moments) at each end of the flex-strap due to as many relative deflections at the ends of the flex-strap. To this end, it was assumed that the flex-strap is a curved beam and possesses two modes of deflection:

bending in the radial direction (about the minimum bending modulus axis), and torsion about the circular arc elastic axis. These two modes were, furthermore, assumed to be uncoupled. Although the flex-strap was assumed to be rigid in the direction normal to the radii (the maximum bending moduli), nevertheless deflections in this direction are allowed due to effects of a twisted, curved elastic axis. The stiffness matrices were then obtained by inverting the appropriate influence coefficient matrices.

The influence coefficient matrix for radial deflections was obtained from a standard compendium of beam bending formulae (Ref. 13). The influence coefficient matrix for deflections due to torsion was derived from first principles using a differential equation for torsion which included both St. Venant and plate bending effects (see Ref. 8) and assumed a nominally circular arc torsion axis.

Vortex Ring State Modification to Time Variable Inflow

The operation of wind turbines is such that they can achieve blade lift coefficients far in excess of those typically encountered by helicopter rotors. As such, they are capable of operating in the "vortex ring state" (see Ref. 14). In such a state the usual momentum balance relationships used to obtain (uniform) inflow distributions break down. The time variable inflow description originally developed for this analysis, however, was derived using a differential equation version of the statement of momentum balance. Consequently, this differential equation had to be modified to incorporate the empirical relationship between inflow and induced velocity given in Ref. 14.

In addition to these specific features, several others had already been made to the original "helicopter" program to enable the analysis of more conventional wind turbine systems. These basic wind turbine modifications were already part of the analysis capability of UTRC at the inception of this study, and a detailed discussion of them is beyond the scope and intention of this report.

Dynamic Data Required for Analysis

Since the F762BR aeroelastic analysis models the features of this wind turbine system with more rigor than the G400 analysis used in the preliminary design, some of the input data for the former will be somewhat different, both in scope and value, from that for the latter. This subsection briefly describes the pertinent dynamic data required as input for the F762BR program.

Blade Torsion Mode

The explicit dynamic description of the control pendula relaxes the need for approximating the equivalent root torsion spring due to pendulum impedance. Consequently, the appropriate torsion modeling required to calculate the torsion frequency and mode shape requires that the flexstrap be completely disconnected and the blade stiffened in torsion only by the rotor blade flexbeam. This substantially softens the blade (alone) in torsion from what is shown in Fig. 4. For the nominal wind condition of 12 mph and tip speed of 150 fps this torsion mode was calculated to

have a frequency of 8.07 cps (6.76 P). This increased to 10.73 cps (2.70 P) at a tip speed of 500 fps. The remaining blade mode natural frequency variations with tip speed used in this stability study are essentially those shown in Fig. 5.

Flexstrap and Pendulum Geometry

The derivation of the torsion dependent flexstrap influence coefficients was made making the assumption of a nominally circular arc elastic axis. From the design details of the flexstrap selected and discussed earlier in this study, the following table presents the simplified geometric and mechanical data used in the stability analysis.

Flexstrap and Pendulum Data

Pendulum radial hinge location	28.5 in.
Pendulum axial hinge location	4.25 in.
Forward sweep angle of pendulum	45 deg
Flexstrap-pendulum attachment point from hinge	18 in.
Flexstrap radius of curvature	46.5 in.
Flexstrap-blade attachment pitch angle	-23 deg

Tower-Nacelle Dynamic Description

From the results of the NASTRAN calculations it was deemed that the rotor-tower dynamic interactions would be negligible for the elastic (finite frequency) natural modes. However, the inherent zero frequency yaw mode due to the free yaw bearing had to be included in the tower dynamic description. The adopted method for including finite impedance characteristics of the rotor hub in the F762BR program is to input inertia, damping and stiffness matrices for a "movable hub". All of these matrices are 6 x 6 to account for the 6 degrees-of-freedom of the hub. The inertia and stiffness matrices used in the calculations which closely simulate the first elastic bending modes of the tower, as well as the zero frequency yaw mode, are shown in Table I. Note that since zero hub damping was assumed, the damping matrix was omitted. The 6 elements in each row and column correspond, in order, to vertical (downward) deflection, lateral (starboard) deflection, clockwise (viewed from above) yaw rotation, nacelle (nose up) pitching rotation, axial (downwind) deflection, and rotation about an axis coincident with the rotor shaft.

Variable Inflow Due to Wind Shear

The F762BR computer program has the capability of inputting a discrete inflow distribution over the rotor disk which varies both radially and azimuthally. Consequently, for each wind speed an appropriate inflow distribution is calculated so that the inputted wind speed and variable inflow distribution combine to simulate the earth wind shear distribution. The wind shear model so selected is the typical power curve (See Ref. 15), whose exponent was taken to be 0.2.

TABLE 1
DYNAMIC MATRICES USED TO SIMULATE FLEXIBLE
TOWER WITH FREE YAW BEARING

Inertia Matrix

$$[M] = \begin{bmatrix} 1000. & 0 & 0 & 0 & 0 & 0 \\ 0 & 5.8512 & 336.567 & 0 & 0 & 0 \\ 0 & 336.567 & 25712.8 & 0 & 0 & 0 \\ 0 & 0 & 0 & 2 \times 10^6 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1000. & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \times 10^6 \end{bmatrix} \begin{matrix} (1b) \\ (1b) \\ (1a.-1b) \\ (1a.-1b) \\ (1b) \\ (1a.-1b) \end{matrix}$$

(sec²/in.) (sec²/in.) (sec²/rad) (sec²/rad) (sec²/in.) (sec²/rad)

Stiffness Matrix

$$[K] = 10^6 \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0.1 & 0.6 & 0 & 0 & 0 \\ 0 & 0.6 & 36.0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 14000. & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 7000. \end{bmatrix} \begin{matrix} (1b) \\ (1b) \\ (1a.-1b) \\ (1a.-1b) \\ (1b) \\ (1a.-1b) \end{matrix}$$

(/in.) (/in.) (/rad) (/rad) (/in.) (/rad)

Results of Analysis

The essential stability results for the nominal 12 mph wind condition are shown in Figs. 29, 30 and 31. Figure 29 presents the dynamic stress and bending moments for one of the blades over one rotor revolution. The blade strength indices selected were the bending stresses at the blade (flexbeam) root and the bending moments at the 50 percent span. This figure shows the essentially 1 P response, due mostly to gravity excitation, which are well within the high strength limits for the composite materials selected. Also shown in this figure is the negligible effect that the inclusion of a 50 percent tower shadow (reduction by 50 percent of the wind velocity behind the tower) has upon the results. All subsequent calculations were justifiably made disregarding the shadow. The 5 P content in the responses can be attributed to the close proximity of the first edgewise natural frequency.

Figure 30 shows the concurrent angular responses for the reference (No. 1) blade in torsion (elastic twist) and pendulum swing as well as the pendulum swing angle for the second blade. The pertinent information gleaned from this figure

Figure 29. Time-Histories of Blade Bending Stresses and Moments at 12 mph

Figure 30. Time-Histories of Blade Elastic Twist and Pendula Swing Angles at 12 mph

is first that, as expected, the pendulum functions analytically in the correct manner. That is, that the blade pitch angle is essentially slaved to the pendulum attached to it. The second significant result of this figure is that a substantial amount of unrequired cyclic pitch input (1 P content) is generated. This 1 P content appears to be generated by the 1 P gravity loads acting on the pendula and on the blade itself through ΔEI edgewise-flatwise nonlinear coupling excitation of torsion. This latter effect is dependent upon rotor thrust and built-in blade coning. One consequence of this self-generated cyclic pitch is the relatively large equilibrium yaw angle developed by the nacelle of -21.8 deg.

However, that the rotor and nacelle do, in fact, possess weathercock stability is shown in Fig. 31, where the nacelle yaw angle responses are given due to positive and negative 30 degree values of instantaneous wind angle changes (gust direction angles). Also, shown in this figure are the vibratory blade root

Figure 31. Effects of Gust Direction Changes on Nacelle Yaw Angle and Blade Root Vibratory Stresses at 12 mph

stresses developed on the rotor for each of these gust direction angles. It is significant to note the difference in character of the transient vibratory stresses for each of the gust direction angles. However, the responses are quite stable and the stresses are well within stress allowables. Although no explicit axial gust transient responses have been calculated, basically the sudden wind direction changes discussed above can be interpreted as sudden changes in the axial wind speed, which is a gust.

Table II compares the blade stresses, equilibrium nacelle yaw angles, and rotor thrust and power predicted for the 12 mph (nominal) and 15 mph low speed wind conditions. Operation at these conditions should be stable and present no operational difficulties:

TABLE II

LOW SPEED WIND CONDITION AEROELASTIC PERFORMANCE

Wind Speed (mph)	12	15
Tip Speed (fps)	150	200
Root Bending Stresses (psi)		
Median Flatwise	-7340	-10430
1/2 PTP Flatwise	2630	3015
Median Edgewise	-1300	-2740
1/2 PTP Edgewise	2280	2980
50 Percent Span Bending Moments (in.-lb)		
Median Flatwise	10	439
1/2 PTP Flatwise	270	279
Median Edgewise	-2030	-3980
1/2 PTP Edgewise	3340	4330
Nacelle Equilibrium Angle (deg)	-21.8	-11.7
Rotor Thrust (lb)	353	609
Power (kW)	4.96	7.45

CONCLUSIONS

The conclusions set forth below were arrived at as a result of analytical studies conducted under this contract. Some of these conclusions have been substantiated through model tests conducted under an earlier DOE contract, however final verification would not be achieved until full-scale testing is accomplished.

1. A potentially low cost wind turbine can be designed using the Composite Bearingless Rotor (CBR) Concept, wherein no active controls are required to achieve a system having self-starting capability, self alignment, and high speed survivability.
2. Over the wind speed range studied the UTRC Composite Bearingless Wind Turbine is stable under steady wind conditions, is self-aligning with wind direction changes, and does not adversely couple with tower motions when tower frequencies are greater than the rotor speed. (Low tower frequencies were not studied.)
3. The CBR wind turbine operates under steady and gusty conditions at calculated blade stresses below the infinite fatigue life stress levels of the blade materials.
4. High windspeeds, up to 60 mph, can be survived by properly scheduling blade pitch to produce blade stall, thereby limiting rotor rpm.
5. Except for the composite materials used in blade fabrication, all components of the CBR wind turbine can be purchased off-the-shelf or fabricated from conventional materials.
6. A simple, single-column support tower with guy cables is adequate for this size (40 ft diameter) wind turbine. It is feasible to have at least two tower segments for ease of handling.

SPECIFICATIONS, WEIGHTS, AND DRAWINGS

A summary of the important design parameters is given in Table III and a list of component weights and center of gravity locations is given in Table IV.

Reductions of the engineering drawings have been made and are contained at the end of this report. Due to the extent of the reduction required, it is difficult to extract detailed information from these copies, however sufficient information can be obtained to show the general features of the design.

TABLE III

CBR WIND TURBINE SPECIFICATIONS

General

Diameter	40 ft
Number of blades	2
Tip Speed	
at $V_w = 12$ mph	150 fps
15 mph	200 fps
18 mph	250 fps
Shaft Power (est.)	
at $V_w = 12$ mph	5 kW
15 mph	8 kW
18 mph	12 kW

Hub

Configuration	2 or 4 way
Material	Steel
Offset	10 in.

Flexbeam

Length	36 in.
Chord	7 in. (constant)
Thickness	.7 in. (constant)
Material	graphite/epoxy
Pretwist	-20 deg
Precone	10 deg
EI (Flat)	3.6×10^6 lb-in. ²
EI (Edge)	$360. \times 10^6$ lb-in. ²
GJ	$.489 \times 10^6$ lb-in. ²
Weight	9.7 lbs

Blade (Production Blade)

Configuration	Spar and shell(spar and foam)
Material	Fiber glass (graphite/epoxy and foam)
Chord	15 in.
Taper	none
Twist	none
EI (Flat)	3.73×10^6 lb-in. ² (3.75×10^6)
EI (Edge)	334.3×10^6 lb-in. ² (330.7×10^6)
GJ	13.13×10^6 lb-in. ² (7.68×10^6)
Weight	57 lbs (83)
c.g.	.25 c
e.a.	.355C (.22 c)
N.A.	.233C (.189 c)
Ip	.0158 lb-sec ²
k	4.33 in. (3.6)

TABLE IV

WEIGHT AND BALANCE

When the turbine main shaft axis and the yaw axis intersect, they form a plane of symmetry in which the r and x axis are chosen along the respective shafts to locate nacelle components. The origin is the intersection, r is positive towards the turbine and z is positive down along the yaw axis.

Component	Weight lbs	r inches	z inches
2 Turbine Blades	114.0		
2 Turbine Blade Flanges	70.0		
2 Flexbeam Flanges	40.0		
8 Flexbeam Fasteners	7.5		
2 Strap Brackets	21.5		
2 Control Straps	8.5		
1 Flexbeam	53.0		
1 Hub	65.0		
1 Pendulum Support Bar	58.5		
2 Pendulum Arms	40.5		
2 Pendulum Proper	88.0		
2 Pendulum Strap Joint	11.0		
Total	578.0	60	0
1 Hub-Shaft Attachment Flange	82.0	59	0
1 Main Shaft	149.0	20.5	0
1 Hub Bearing + Mount	97.0	52.0	0
1 Thrust Bearing + Mount	97.0	-12.5	0
1 U-Channel	760.0	10.0	4.6
1 Large Pulley, First Step-Up	146.0	-17.2	0
1 Drive Belt, First Step-Up	9.3	-17.2	9.6
1 Small Pulley, First Step-Up	35.5	-17.2	27.6
1 Intermediate Shaft	21.0	-22.9	27.6
1 Intermediate Shaft Bearing	16.0	-21.6	27.6
1 Intermediate Shaft Bearing	16.0	-24.4	27.6
1 Small Pulley, Second Step-Up	35.5	-28.5	27.6
1 Drive Belt, Second Step-Up	9.3	-28.5	18.0
1 Large Pulley, Second Step-Up	146.0	-28.5	0.0
1 15 kW Generator	390.0	-41.0	0.0
Total Weight Nacelle Plus Turbine	2587		
Center of Gravity		+10.85	+2.77

TABLE IV (Cont'd)

Component	Weight lbs	r inches	z inches
1 Yaw Axis Plus Flanges	202.0	0	20.6
1 Yaw/Thrust Bearing	35.5	0	21.5
1 Yaw Bearing	35.5	0	31.5
1 Mount Yaw Mechanism	<u>183.0</u>	<u>0</u>	<u>25.2</u>
Total Weight on Top of Tower	3043.6		
Center of Gravity		9.2	5.9
2 Flanged Pipes Tower	1484		
9 Guy Wire Sections	630		
3 Turnbuckles	129		
3 Spreaders	<u>100</u>		
Total Weight Wind Turbine	5395		

NOMENCLATURE

a	Distance strap attachment to weight of pendulum, inches
C	Blade chord, in.
C_D	Section drag coefficient
C/E	Indicating carbon/epoxy
c.g.	Center of gravity
C_P	Power coefficient, $P/1/2 \rho V^3 n R^2$
E	Indicating edgewise
e	Pendulum hinge offset, inch
e.a.	Elastic Axis
EI	Bending stiffness, lb-in ²
F	Indicating flatwise
F/G	Indicating fiber glass
G	Shear modulus, psi
GJ	Torsional stiffness, lb-in ²
I_P	Polar moment of inertia, lb-sec ²
K	Stiffness matrix
k	Radius of gyration, inch
kW	Kilowatts
L	Flexbeam length which twists, inch
l	Distance hinge to strap attachment of pendulum, inches
l_b	Flexbeam length outboard of virtual hinge line, inch
M	Mach number, Inertia Matrix
m	Pendulum weight, lbs

NOMENCLATURE (Continued)

N.A.	Neutral axis
P	Turbine shaft power, ft-lb/sec
PTP	Indicating peak-to-peak
R	Turbine radius, radius of gyration, ft or in.
r	Radial length, ft or in.
$\bar{r}$	Nondimensional radial length, r/R
T	Thrust, lb or indicating torsion
t	Thickness, in.
V_w	Wind speed, mph
α	Section angle of attack, deg
β_b	Blade flap angle, deg
β_p	Pendulum flap angle, deg
β_{p_1}	Pendulum initial flapping angle (no gravity), deg
β_{p_0}	Pendulum static flap angle, deg
δ_3	Pitch-flap coupling ratio
θ	Elastic twist of flexbeam, deg
θ_1	Blade twist between root and tip, deg (ideal twist is a hyperbolic distribution; if not specified, distribution is linear)
θ_{75}	Blade pitch at $0.75R$, deg
σ	Shear stress, psi
ψ	Azimuth angle blade ($\psi = 0$ vertically down), deg
ϕ	Azimuthal leading angle of pendulum, deg
Ω	Rotational speed, rad/sec
ω	Natural frequency of pendulum, rad/sec

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APPENDIX I

WIND CHARACTERISTICS AT BRADLEY FIELD

Weather data at Rentschler Airport are taken only intermittently and therefore the records of the U.S. Weather Bureau at Bradley Field were used instead. These two sites should experience similar wind speeds since their topography is the same and they are separated by only 15 miles.

Table A1 shows the meteorological data for four years, 1950, 1960, 1970, and 1975, and also the normals, means and extremes from a ten-year period ending at each of those years. Over the past 25 years the annual averages, as well as the 10 year averages, show close agreement. It shows also the pattern that the second, third, and fourth months of the year have the highest averages. The average hourly speeds for the best months are slightly below the desired levels; however, one should account for the daily pattern as well. At night the wind velocities are considerably lower than the peak velocities, which occur around noon. This effects the 24 hour average in a decreasing sense. The daytime averages are therefore in general above the day average hourly velocities of the month. Since both Rentschler Airport and Bradley Field have the same open grass field shear layer the velocities can be multiplied by 1.15 to correct the 20 feet anemometer height of Bradley Field to 40 feet.

LATITUDE 41° 56' N
 LONGITUDE 72° 41' W
 ELEVATION (ground) 169 feet

TABLE: A 1

Cup - anemometer height: 20', in
 center of field
 Wind movement in mph

HARTFORD, CONNECTICUT
 BRADLEY FIELD

METEOROLOGICAL DATA FOR THE INDICATED YEAR

Month	Wind				
	Average hourly speed	Prevailing direction	Fastest mile		
			Speed	Direction	Date
Jan.	9.7	N	38	N	16
Feb.	9.3	N	57	N	9
Mar.	10.8	NE	17	N	8
Apr.	9.1	N	34	N	4
May	7.4	SE	37	NE	7
June	7.9	S	34	SE	17
July	5.8	SE	34	SE	12
Aug.	6.0	SE	28	N	29
Sept.	6.0	NNE	24	N	13
Oct.	5.1	N	25	N	20
Nov.	5.9	N	30	N	25
Dec.	5.0	N	30	N	5
Year	7.9	N	70	N	Nov. 25

Month	Wind				
	Average hourly speed	Prevailing direction	Fastest mile		
			Speed	Direction	Date
JAN	1.7	N	34	NE	16
FEB	9.9	SE	50	NE	27
MAR	9.3	SE	15	NE	25
APR	9.2	N	37	SE	13
MAY	7.3	N	33	SE	29
JUN	7.3	N	33	SE	29
JUL	4.4	S	24	S	1
AUG	4.3	N	19	NE	9
SEP	4.3	N	43	NE	12
OCT	7.3	SE	35	NE	24
NOV	7.3	N	30	N	10
DEC	4.4	N	31	N	12
Year	7.8	S	52	NE	Feb 27

Month	Wind				
	Direction	Speed	Average speed	Fastest mile	
				Speed	Direction
JAN	24	4.3	6.7	30	NE
FEB	28	7.3	6.0	36	NE
MAR	31	7.2	6.3	36	NE
APR	29	3.0	11.7	42	NE
MAY	31	7.1	5.3	39	NE
JUN	23	5.3	6.3	32	NE
JUL	20	2.9	4.1	28	NE
AUG	27	4.2	7.0	39	NE
SEP	24	4.9	7.4	39	NE
OCT	32	6.3	7.3	37	NE
NOV	34	3.0	7.4	39	NE
DEC	37	6.3	4.4	34	NE
Year	29	4.9	6.3	42	NE

Month	Wind				
	Direction	Speed	Average speed	Fastest mile	
				Speed	Direction
JAN	30	7.3	8.7	35	NE
FEB	29	4.4	6.4	35	SE
MAR	32	5.0	10.3	52	NE
APR	31	8.0	11.0	57	NE
MAY	29	2.5	7.0	86	NE
JUN	26	4.2	8.3	88	NE
JUL	20	4.4	6.4	88	S
AUG	22	2.9	7.2	84	NE
SEP	26	4.6	7.4	49	NE
OCT	31	2.7	7.7	35	N
NOV	23	2.9	7.7	30	NE
DEC	28	3.2	8.0	29	NE
Year	29	2.3	6.6	87	NE

NORMALS, MEANS, AND EXTREMES FOR 10-YEAR PERIOD ENDING AT INDICATED YEAR

Year	Wind				
	Month	Mean hourly speed	Prevailing direction	Fastest mile	
				Speed	Direction
1950	JAN	9.7	N	38	N
1951	FEB	9.3	N	57	N
1952	MAR	10.8	NE	17	N
1953	APR	9.1	N	34	N
1954	MAY	7.4	SE	37	NE
1955	JUN	7.9	S	34	SE
1956	JUL	5.8	SE	34	SE
1957	AUG	6.0	SE	28	N
1958	SEP	6.0	NNE	24	N
1959	OCT	5.1	N	25	N
1960	NOV	5.9	N	30	N
1961	DEC	5.0	N	30	N
1962	Year	7.9	N	70	N

Year	Wind				
	Month	Mean hourly speed	Prevailing direction	Fastest mile	
				Speed	Direction
1959	JAN	1.7	N	34	NE
1960	FEB	9.9	SE	50	NE
1961	MAR	9.3	SE	15	NE
1962	APR	9.2	N	37	SE
1963	MAY	7.3	N	33	SE
1964	JUN	7.3	N	33	SE
1965	JUL	4.4	S	24	S
1966	AUG	4.3	N	19	NE
1967	SEP	4.3	N	43	NE
1968	OCT	7.3	SE	35	NE
1969	NOV	7.3	N	30	N
1970	DEC	4.4	N	31	N
1971	Year	7.8	S	52	NE

Year	Wind				
	Month	Mean hourly speed	Prevailing direction	Fastest mile	
				Speed	Direction
1969	JAN	2.4	N	30	NE
1970	FEB	7.3	SE	36	NE
1971	MAR	7.2	N	36	NE
1972	APR	7.1	N	39	NE
1973	MAY	7.3	N	33	SE
1974	JUN	7.3	N	33	SE
1975	JUL	4.4	S	24	S
1976	AUG	4.3	N	19	NE
1977	SEP	4.3	N	43	NE
1978	OCT	7.3	SE	35	NE
1979	NOV	7.3	N	30	N
1980	DEC	4.4	N	31	N
1981	Year	7.8	S	52	NE

Year	Wind				
	Month	Mean hourly speed	Prevailing direction	Fastest mile	
				Speed	Direction
1979	JAN	2.4	N	30	NE
1980	FEB	7.3	SE	36	NE
1981	MAR	7.2	N	36	NE
1982	APR	7.1	N	39	NE
1983	MAY	7.3	N	33	SE
1984	JUN	7.3	N	33	SE
1985	JUL	4.4	S	24	S
1986	AUG	4.3	N	19	NE
1987	SEP	4.3	N	43	NE
1988	OCT	7.3	SE	35	NE
1989	NOV	7.3	N	30	N
1990	DEC	4.4	N	31	N
1991	Year	7.8	S	52	NE

1950

1960

1970

1975

GENERAL LAY-OUT OF WIND ENERGY CONVERSION SYSTEM

Drawing 1: General lay-out of wind energy conversion system.

78-02-178-2

LAY-OUT OF NACELLE AND COMPONENTS

Drawing 2: Lay-out of nacelle and components.

DETAILS OF HUB AND PENDULUM/FLEXBEAM MECHANISM

DETAIL OF BLADE ROOT ATTACHMENT

Drawing 4: Detail of blade root attachment.

DETAILS OF FLEXBEAM GEOMETRY AND FIBER LAY-UP