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Initial Structural Response Measurements and Model Validation for the Sandia 34–Meter VAWT Test Bed

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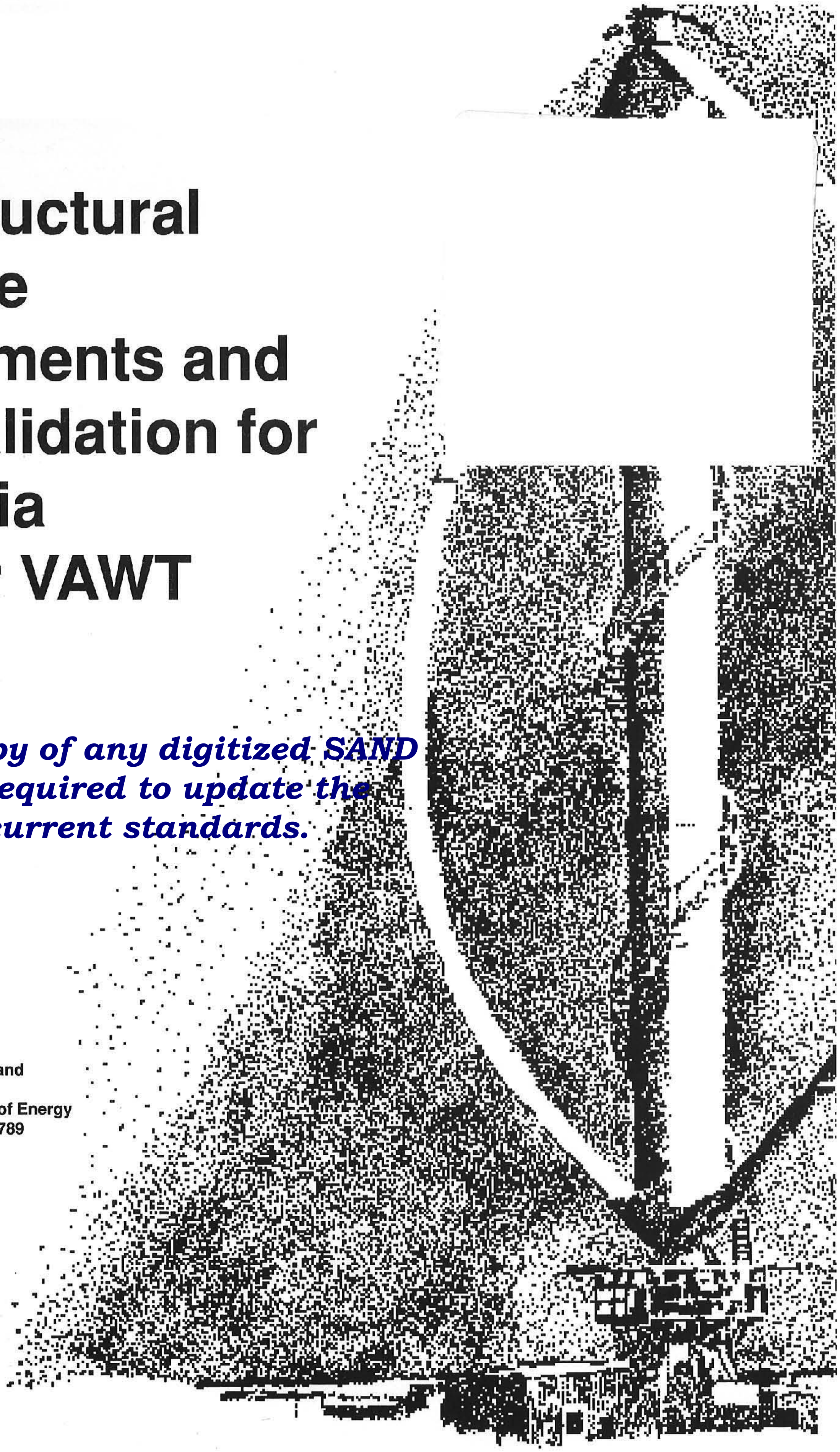

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INITIAL STRUCTURAL RESPONSE MEASUREMENTS AND MODEL VALIDATION FOR THE SANDIA 34-METER VAWT TEST BED

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ABSTRACT

Sandia National Laboratories (SNL) has designed and constructed a 34-meter diameter vertical axis wind turbine (VAWT) Test Bed. The machine will be used to advance research in aerodynamics and structural dynamics, improve fatigue life prediction capabilities, and investigate control algorithms and system concepts. The Test Bed has extensive instrumentation including 70 strain gauges to measure blade and tower response. Immediately after the blades were mounted, blade gravity stresses were measured and a modal test on the stationary rotor performed to determine zero rpm modal frequencies. Assembly and start-up tests are complete, and testing is in the machine characterization phase. Structural resonance surveys will fully characterize the modal frequencies and mode shapes of the rotor, drive train and guy cables. Measured gravity stresses, centrifugal stresses, and modal frequencies are compared to predicted values.

INTRODUCTION

The 34-m VAWT Test Bed is a variable-speed, research turbine recently designed and constructed by Sandia National Labs. The design of this 500 kW machine incorporates the results of recent VAWT research in aerodynamics and structural dynamics. The turbine will be used to evaluate the advances in these areas and to contribute to developments in other areas such as fatigue life predictive techniques, stochastic wind modeling, variable-speed control algorithms, and system design concepts. The construction of the Test Bed, which is located at the U.S.D.A. Agricultural Research Service facility, 15 miles west of Amarillo, Texas, has been completed, and testing is underway. The blades were mounted on the tower in November, 1987, and the first rotation with blades occurred on February 4, 1988.

Assembly and start-up tests, Phase I of the Test Plan (1), are finalized. Phase I included testing of the power system and controller, verification of the blade strain gauges, testing of the transmission and drive train for vibrations and power losses, set-up and checkout of the brake system and implementation of the data acquisition system. Immediately after the blades were mounted, blade gravity stresses were measured and a modal test performed on the stationary rotor. The modal test measured stationary modal frequencies, which have been compared to predicted frequencies. This allows the predicted fanplot (a plot of modal frequencies versus rotation rate) to be used more accurately as a guide for identifying resonant frequencies during rotation and provides validation of the finite element model used to obtain the predictions.

The turbine testing has progressed into the machine characterization phase - Phase II of the Test Plan. This testing will determine the structural response and aerodynamic performance over the entire operating range. It will include further evaluation of the variable speed controller, understanding its capabilities, and upgrading the control algorithm to reflect knowledge gained during the characterization. Before running for long periods of time at different rotation rates and wind speeds, a resonance survey was conducted by operating for short periods at several rotation rates from 6 through 40 rpm in low, medium, and high wind conditions. This survey characterizes the modal frequencies of the rotor, drive train, and guy cables and allows for the determination of mean stress levels and approximate cyclic stress levels. The evaluation of structural response and turbine performance during these short periods has been followed by operation for longer periods of time to collect enough time series records to create statistically reliable, averaged data (2).

This report describes the major aspects of the measured structural response data obtained up to and including the resonance surveys. (The turbine has rotated from 6 to 40 rpm in winds up to 17 mps). After a general description of the Test Bed structure and the finite element model employed in the structural design, measurements of gravity stresses, centrifugal stresses, and modal frequencies are shown and compared to predicted values.

GENERAL DESCRIPTION OF THE TEST BED

The 34-meter Test Bed is rated at 500 kW at 37.5 rpm in a 12.8 mps (28 mph) wind. Figure 1 shows a photograph of the turbine in its completed form. The rotor is 34 meters (110 feet) in diameter and has a height-to-diameter ratio of 1.25. The turbine, which has a total height of 50 meters (165 ft), is supported at the top by three sets of double guy cables each of which is 63.5 mm (2 1/2 in.) in diameter. Each cable pair is tensioned to 826 kN (186,000 lb) and is anchored at ground level to a reinforced concrete tiedown block measuring 4.3x4.3x6.1 m^3 (14x14x20 ft^3). The tower, or column, is an aluminum cylinder, 3 meters (10 ft) in diameter, constructed of 13 mm (0.5 in.) rolled plates, butt-welded together. Two blade mounts attached to the tower at each end are box-like structures that provide blade attachment surfaces and transition the tower diameter down to bearing shaft diameters.

Each blade is step-tapered with five sections constructed of extruded 6063-T6 aluminum. The top and bottom root sections are straight with a 1.22 meter (48 in.) chord using a NACA 0021 profile. The equatorial section, with a 17.1 meter (675 in.) radius of curvature, has a 0.91 meter (36 in.) chord with an SNLA 0018/50 profile. The intermediate sections, with a 30.0 meter (1180 in.) radius of curvature, have a 1.07 meter (42 in.) chord with an SNLA 0018/50 profile. The SNLA 0018/50 airfoils are part of a series of natural laminar flow airfoils developed at Sandia specifically for use on VAWT's (3). Because all three chords are too large to be fabricated from a single extrusion, each blade section is made of two or three single extrusions bolted together in the spanwise direction. The seven different extrusions that make up the three profiles were successfully extruded to specification during October, 1986 by Consolidated Aluminum, Inc. of Madison, Illinois. The 36-inch and 42-inch chord sections were bent to the proper radius around dies designed specifically for the Test Bed. The extrusion-to-extrusion bolts were retorqued to specification after the blade bending. The extrusion-to-extrusion assembly and blade bending process were performed by Flow Industries of Kent, Washington.

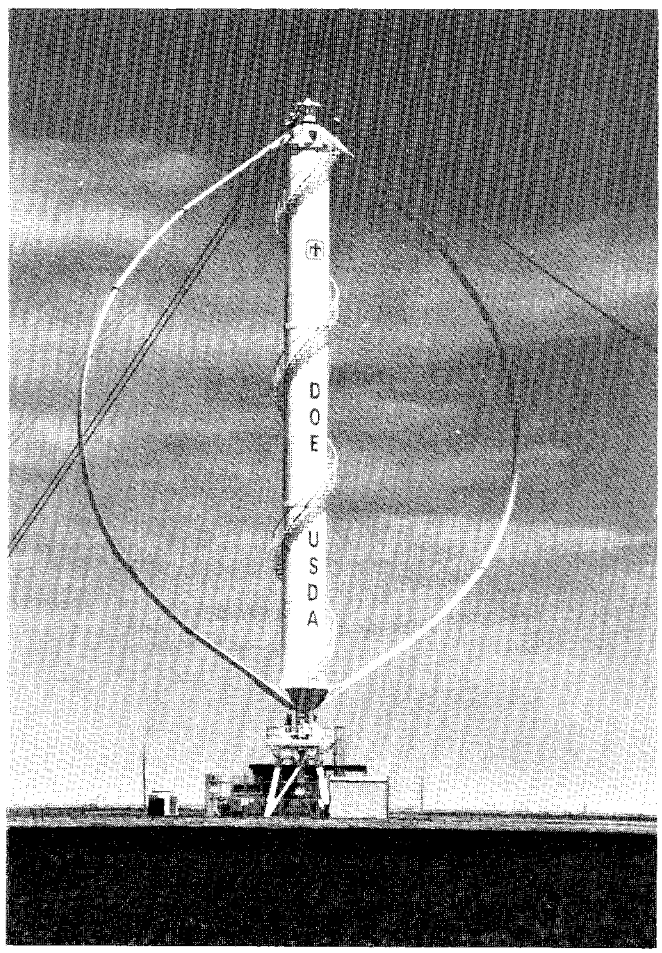

Fig. 1. 34-Meter Test Bed

The blade shape approximates a 37.5 rpm troposkien and contains "kinks", or slope discontinuities, of 6-7 degrees at the blade-to-blade joints as shown in Fig. 2, a schematic of the blade shape geometry. Because of the multi-sectioned, step-tapered characteristics of the Test Bed blades, the normally-used straight/single-curve/straight approximation to a troposkien was not employed. A better approximation resulted by implementing the different radii of curvature along the blade and the slope discontinuities at the blade joints where mass is concentrated. Design calculations (4) show that these implementations reduce the largest mean flatwise stresses approximately fifty percent. This mean stress reduction extends the estimated fatigue life by a factor of two to four.

THE FINITE ELEMENT MODEL

The structural design process (5) of the Test Bed incorporated the finite element code NASTRAN to determine the turbine dynamic response. Gravity stresses were also predicted with NASTRAN using a static solution (Rigid Format 24). Predicted modal frequencies were computed with NASTRAN using an eigenvalue solution methodology (FEVD) developed at Sandia (6). In this methodology the stiffness matrix, representing the turbine structure when subjected to centrifugal and gravitational loading, is determined by running a series of geometric nonlinear analyses (Solution 64). The complex eigenvalue analysis (Solution 67) is then used after including rotating coordinate system effects (Coriolis and softening matrices). The mean stresses due to gravitational and centrifugal loading at a particular rpm are output by Solution 64. The complex mode shapes and their respective frequencies of vibration are determined by Solution 67.

A forced response analysis code, called FFEVD (7), determines the vibratory stresses. In FFEVD the stiffness, Coriolis and softening matrices are computed just as in FEVD and input to the complex frequency response analysis (Solution 68) along with the steady (per-rev) wind loads determined by the double streamtube code CARDAA (8). Solution 68 then computes the structural response at each per-rev frequency.

Figure 3 shows the finite element grid for the 34-m Test Bed. It consists of 42 CBEAM elements for each blade, 25 CBEAM elements for the tower, and 2 CBEAM's for each of the four "mini-struts." (The NASTRAN beam element, CBEAM, has properties that include two-dimensional bending, extension, and torsion.) These ministruts model the large blade mounts. The guy cables that restrict motion at the top of the turbine are represented by two orthogonal, horizontal springs, a torsional spring, and a

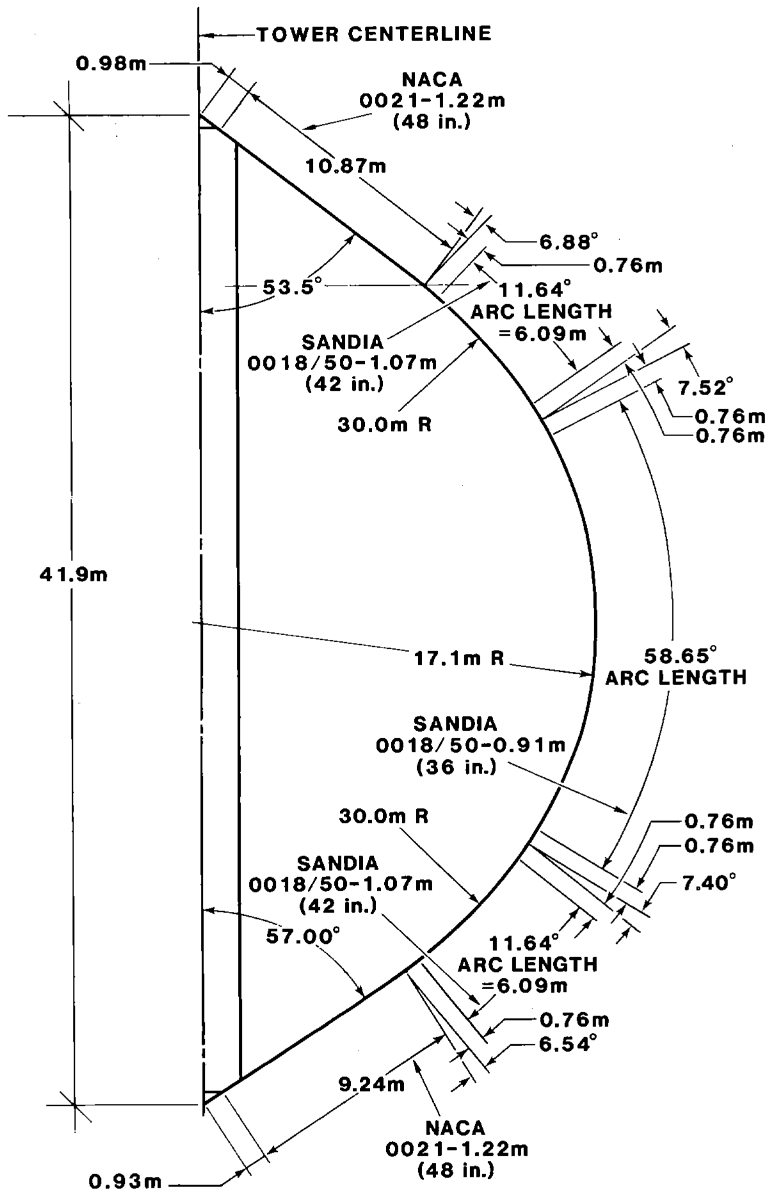

Fig. 2. Blade Shape Geometry

vertical load acting down through the tower. At the bottom of the rotor, at grid point 1, the restriction of motion by the support stand is represented by two orthogonal, horizontal springs. A torsional spring at grid point 1 represents the torsional stiffness of the drive train. Concentrated masses are added at several grid points along the tower to account for the entire tower mass. The mass of the blade joints is distributed evenly along the relatively short joint elements. As the turbine was constructed, each component was measured and weighed. The concentrated masses and properties of the finite elements were then updated in the model to reflect these "as-built" conditions.

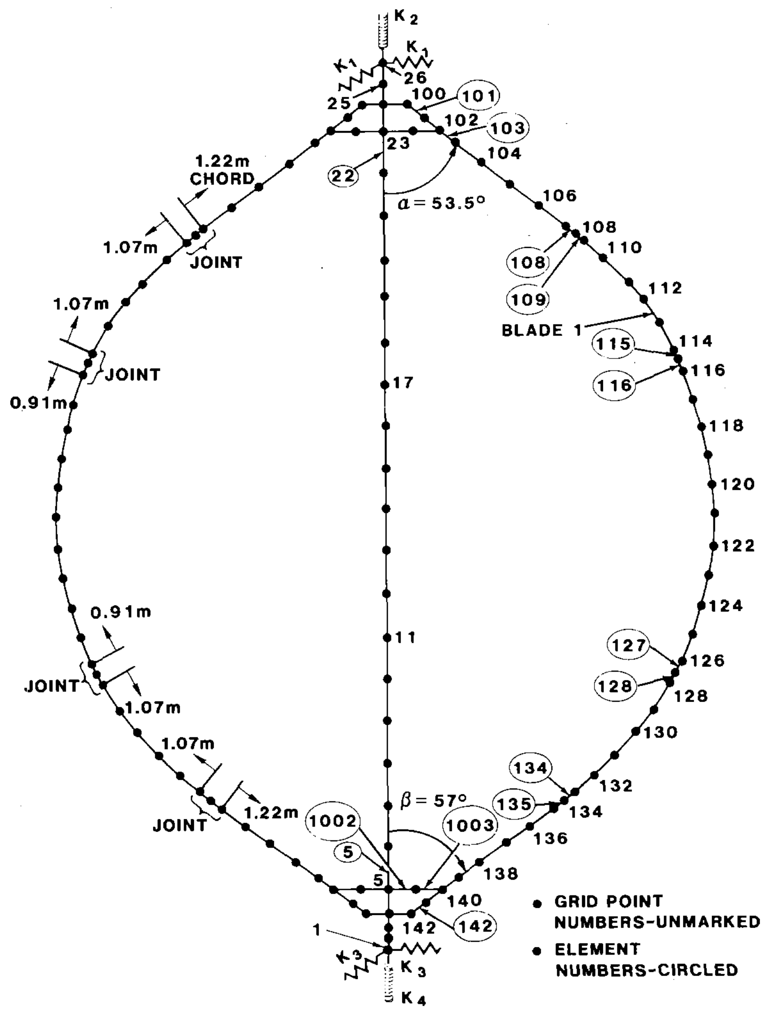

Fig. 3. Finite Element Grid

For blade one, elements 101-108 and 135-142 represent the blade root sections, elements 109-115 and 128-134 the intermediate sections, and elements 116-127 the equatorial section. Blade two is identical in structure to blade one. For comparisons with test results, an additional boundary condition was added to the model at grid point 2 to represent the brakes in the engaged condition. This condition was modelled as two orthogonal, horizontal springs and a torsional spring (torsion about the vertical axis) equal to the appropriate stiffnesses of the support stand.

GRAVITY STRESSES

The Test Bed has been instrumented with 70 strain gauges that measure axial, lead-lag, and flatwise response in the blades and torque and bending in the tower (9). Brake strain gauges and damage gauges that measure crack growth of a pre-cracked aluminum coupon also have been installed. When the blades were first instrumented, the flatwise gauges were validated by hanging known weights from each blade section, recording the strains and comparing them to calculations. The entire set of gauges functioned correctly, and the measured strains agreed with predictions to within 2 percent. Following the gauge validation tests, the blade sections were transported to the turbine site and bolted together while at the same time pulling instrumentation cabling through two blade cavities. The blade cavities containing the cabling were filled with structural foam to prevent cable movement. Before the blades were mounted, the gauges were rechecked, recalibrated and rezeroed. Immediately after the blades were mounted, the gauges were monitored to measure gravity strains.

Figure 4 compares the measured flatwise gravity stresses to the predictions. Stresses along the blade from the top to the bottom are plotted left to right on the x-axis. Positive stress corresponds to tension on the outside of the blade. The patterns of stress distribution for the measured and predicted are very similar; the values agree well at the ends of the blades where fatigue is a major concern. Discontinuities in the stress distribution occur at the joints because of the change of blade stiffness at those locations. The analytical values show an offset along the middle portion, however, indicating an underprediction at the tension side and an overprediction at the compression side of up to 10 MPa (1440 psi). This is also the area where the only significant differences between blades one and two occur. These differences may be due to small errors in the blade shape geometry in the model. Gravity stresses are especially sensitive to small angle changes at the blade-to-tower connection. The blade section lengths were carefully measured in the field, and the blade sections (including the instrumentation cabling, paint and aerodynamic smoothing compound) were weighed several times. A survey of the turbine structure will be performed to attempt to determine the exact "as-mounted" blade attachment angles.

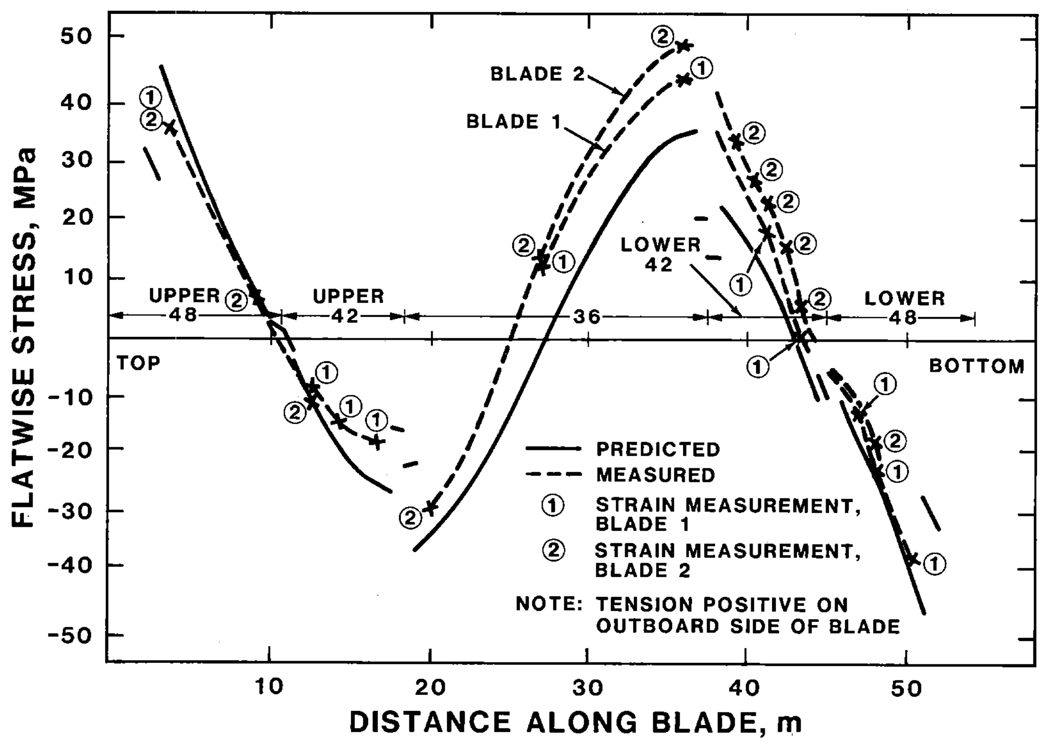

Fig. 4. Flatwise Gravity Stress Distribution

MODAL FREQUENCIES

After the blades were mounted on the tower, a modal test was performed on the stationary rotor by Sandia's Modal and Structural Mechanics Division (7544). During this test, frequency response functions were measured using both wind and step relaxation (snap releasing) excitations. Accelerometers were temporarily attached to the blades, tower and cables. These measurements were used to determine the mode shapes, their frequencies of vibration, and modal damping values. Reference 10 describes the details of the Test Bed modal test.

Table I compares the measured modal frequencies with those determined analytically. The mode number and name are listed in the first two columns (Figure 5 shows the shapes of these modes when viewed from three orthogonal directions). Columns three and four of Table I show the modal frequencies for the stationary rotor measured after a snap release and during wind excitation, respectively. Column five lists the predicted values, and column six lists measured frequencies obtained from amplitude spectra plots of strain gauge data taken during the the wind-excited modal analysis. There is excellent agreement between the measured and predicted frequencies for the first eleven modes. At the higher modes the predicted modal frequencies do not agree as well with the measurements. The two- dimensional finite element model appears to have a limit on its ability to predict these higher modes. However, in general, the higher modes have lower deformation and less energy associated with them.

Mode Number	Mode Shape	Modal Analysis: Snap Release	Modal Analysis: Wind Excitation	Predicted	Strain Gauge Spectra
1	1FA-First Flatwise Antisymmetric	1.04	1.06	1.05	1.06
2	1FS-First Flatwise Symmetric	1.04	1.06	1.05	
3	1Pr First Propeller (Rotor Twist)	1.35	1.52	1.56	1.50
4	1B-First Blade Edgewise	1.81	1.81	1.72	1.82
5	2FA-Second Flatwise Antisymmetric	2.06	2.06	2.07	2.14
6	2FS-Second Flatwise Symmetric	2.16	2.16	2.14	
7	1TI-First Tower In-Plane	2.49	2.50	2.46	2.50
8	1TO-First Tower Out-of-Plane	2.60	2.61	2.58	2.61
9	3FA-Third Flatwise Antisymmetric	3.45	3.50	3.49	3.51
10	3FS-Third Flatwise Symmetric	3.45	3.50	3.51	
11	2Pr Second Propeller	3.59	3.59	3.52	3.59
12	2B-Second Blade Edgewise	-	4.06	3.90	4.05
13	2TI-Second Tower In-Plane	-	4.69	4.33	-
14	2TO-Second Tower Out-of-Plane	-	-	4.57	-
15	4FS-Fourth Flatwise Symmetric	-	5.09	5.25	-
16	4FA-Fourth Flatwise Antisymmetric	-	5.30	5.37	-
17	3Pr Third Propeller	-	-	5.71	-

Table I. Stationary Modal Frequencies - Brakes Engaged

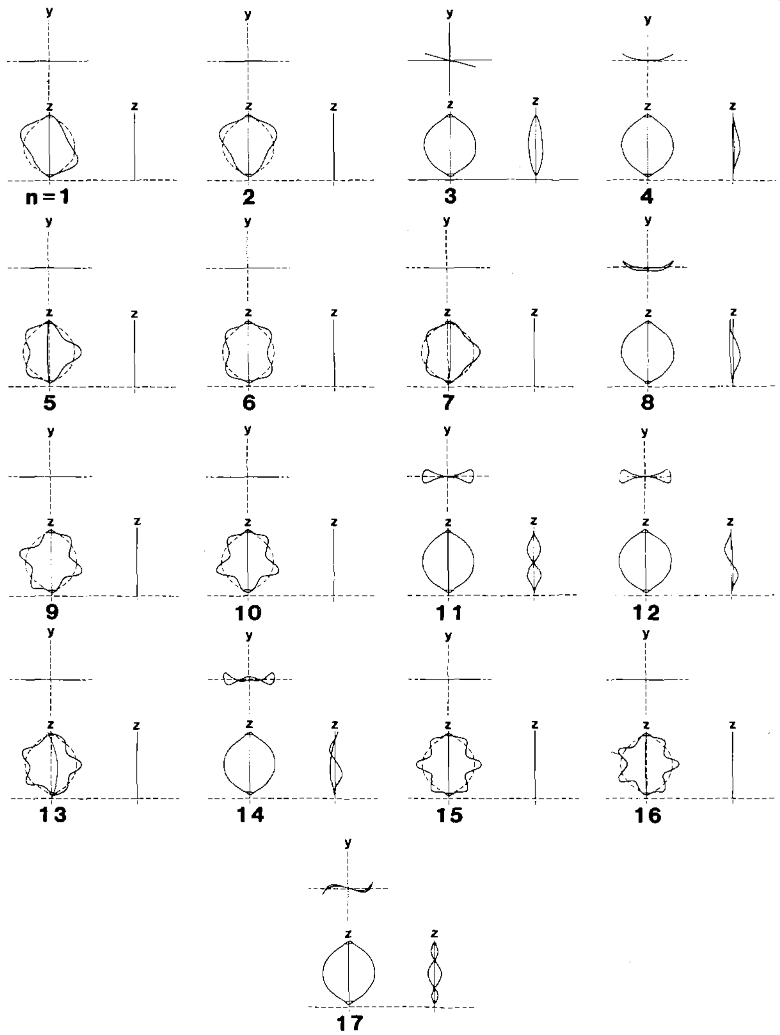

Fig. 5. Mode Shapes

The first propeller mode is approximately 1.5 Hz with the brakes engaged and 0.2 Hz with the brakes released. By adding or changing elastomeric pads (known as hockey pucks because of their size and shape), the drive train stiffness can be adjusted and the frequency of the first propeller mode changed. The first three flatwise modes (both antisymmetric and symmetric), the first tower in-plane and out-of-plane modes, and the second propeller mode were all predicted within two percent of the measured frequencies. The first blade edgewise mode is underpredicted by 0.09 Hz (approximately 5 percent) indicating that the blades are actually slightly stiffer in the lead-lag direction than modeled or that the blade-to-tower attachment should be more rigid in that direction. The close agreement between predicted and measured frequencies for the first eleven modes is an indication that the finite element model does an excellent job in representing the Test Bed - a complicated three-dimensional structure.

The measured frequencies resulting from the two methods of excitation - snap releasing and wind excitation - are very close. Only the first propeller mode shows any significant difference in frequency between the two methods. The reason for this difference is not clear. However, the snap release puts more energy into the system and possibly alters the brake boundary condition slightly. As shown in column six, the peak frequencies from the strain amplitude spectra agree closely with those measured by the modal test.

The first two guy cable frequencies, modes n=1 and n=2, were designed to vibrate at 0.81 Hz and 1.62 Hz, respectively, at a nominal cable tension of 826 kN (186,000 lb), assuming a uniform cross-sectional area the entire cable length. This design puts the first cable mode below 2P (two per-rev) and the second mode between 2P and 4P for the entire operating range. The first four cable frequencies were measured at 0.81, 1.27, 2.05, and 3.02 Hz. The first cable mode occurs at the frequency to which it was designed (0.81 Hz), however, the second measured mode is significantly lower than the second design mode. This is due to an interaction between the cables and heavy cable attachment hardware at the lower cable connection. There is much less motion associated with this mode than the n=1 mode. As the cable fanplot (Fig. 6) indicates, this resonance is the second mode crossing the 2P line. A new guy cable model that includes the ability to model the variation in mass distribution has recently been developed and correctly predicts the four measured cable modes.

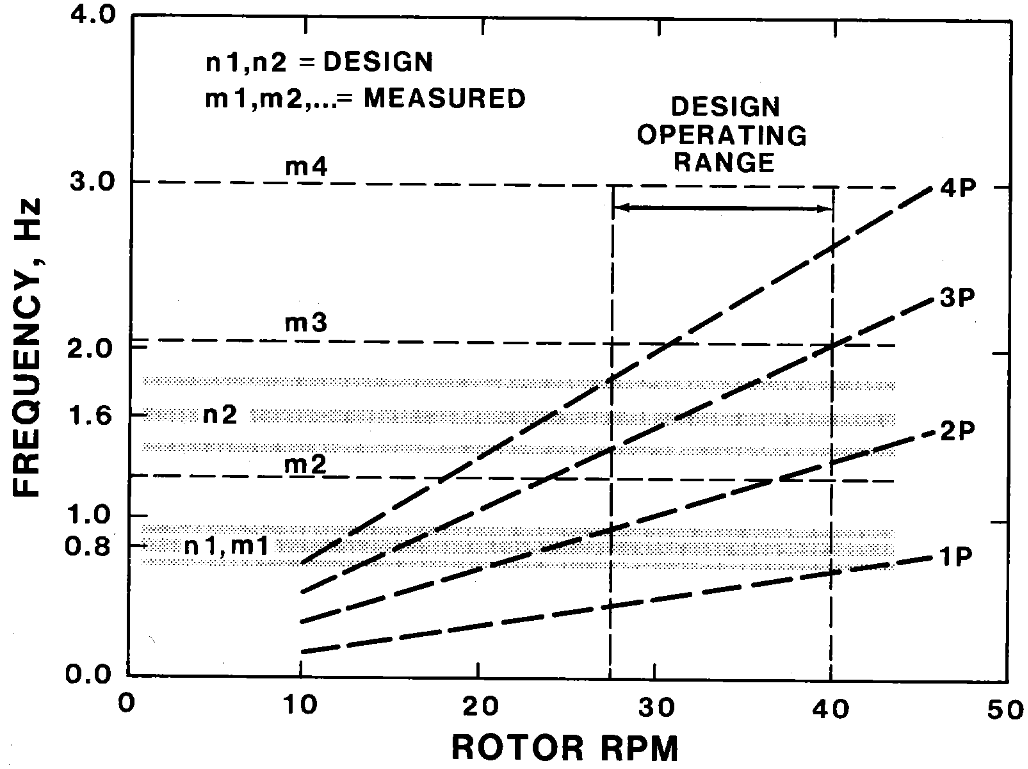

Fig. 6. Cable Fanplot

The predicted fanplot for the "as-built" finite element model with the brakes released is shown in Fig. 7. Because the brake boundary condition is eliminated, some of the stationary tower modal frequencies, as expected, are lower than the modal frequencies shown in Table I. The blade modal frequencies are not affected by this boundary condition. Because potential resonances are possible wherever a modal frequency crosses a per-rev line, both this rotor fanplot and the guy cable fanplot have been employed extensively during the resonance surveys to identify and avoid resonances.

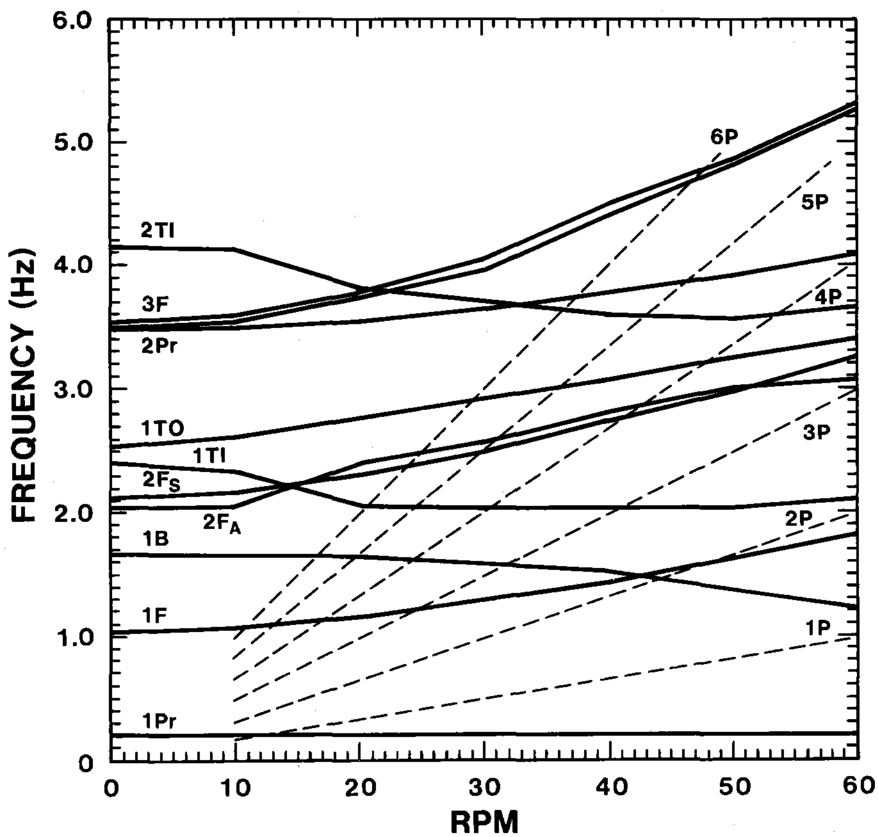

Fig. 7. Rotor Fanplot - Brakes Disengaged

The turbine has operated at rotation rates from 6 to 40 rpm in winds up to 17 mps. Time series have been analyzed with the aid of the data acquisition and analysis (DAAS) software (11). This software can plot any segment of a time series, compute the statistical properties of a segment, statistically reduce the data, and perform spectral analyses. A plot of a strain amplitude spectrum for a flatwise gauge at the upper root at 10 rpm is shown in Fig. 8. The first five per-rev peaks and several other peaks that indicate modal frequencies are evident. By plotting these measured modal frequencies at several rotation rates on the predicted fanplot, as shown in Fig. 9, one can track several rotor modes with rpm. The lower frequency modes (below 3 Hz) including the first and second flatwise (1F,2F), the first blade edgewise (1B), the first tower in-plane (1TI), and the second propeller (2Pr), all track along their predicted mode lines very well. The two measured first flatwise modal frequencies are either the antisymmetric and symmetric modes, which normally vibrate at the same frequency, or the two blades vibrating at slightly different frequencies. The first tower out-of-plane (1TO) does not show up in any of the many spectral plots examined thus far. The first blade edgewise mode (1B) was underpredicted by 5% at zero rpm, but above 25 rpm the observed and predicted frequencies coincide. The first blade edgewise crossing of the 3P line shows up as a larger spike in the spectra of lead-lag gauges at 32 rpm. The stresses are not high in this region in low winds, but significantly increase in winds above 13 mps. The first tower in-plane mode (1TI) tracks well except in the region where it crosses the second flatwise modes. An excitation of this tower in-plane mode begins around 39.5 rpm, and the response is still increasing at 40 rpm. This is due to the 3P crossing predicted at 40.5 rpm. The tower-in-plane excitation which includes blade edgewise motion causes significant lead-lag RMS stresses.

CENTRIFUGAL STRESSES

Figure 10 is a time series record that includes an upper root, flatwise-bending strain gauge and turbine rotation speed. Since the gauges are zeroed before testing, the mean stresses are due solely to centrifugal effects. As the rpm is increased from 0 to 40 rpm, there is an increase in blade bending stresses due to growing centrifugal loading. By averaging each flatwise strain gauge for 40 seconds at each rpm, centrifugal stresses are determined for comparison to analytical results. The increase of centrifugal stresses with higher rpm continues to offset the bending stresses due to gravity until the mean stresses are minimized at 37.5 rpm (the design troposkien rpm).

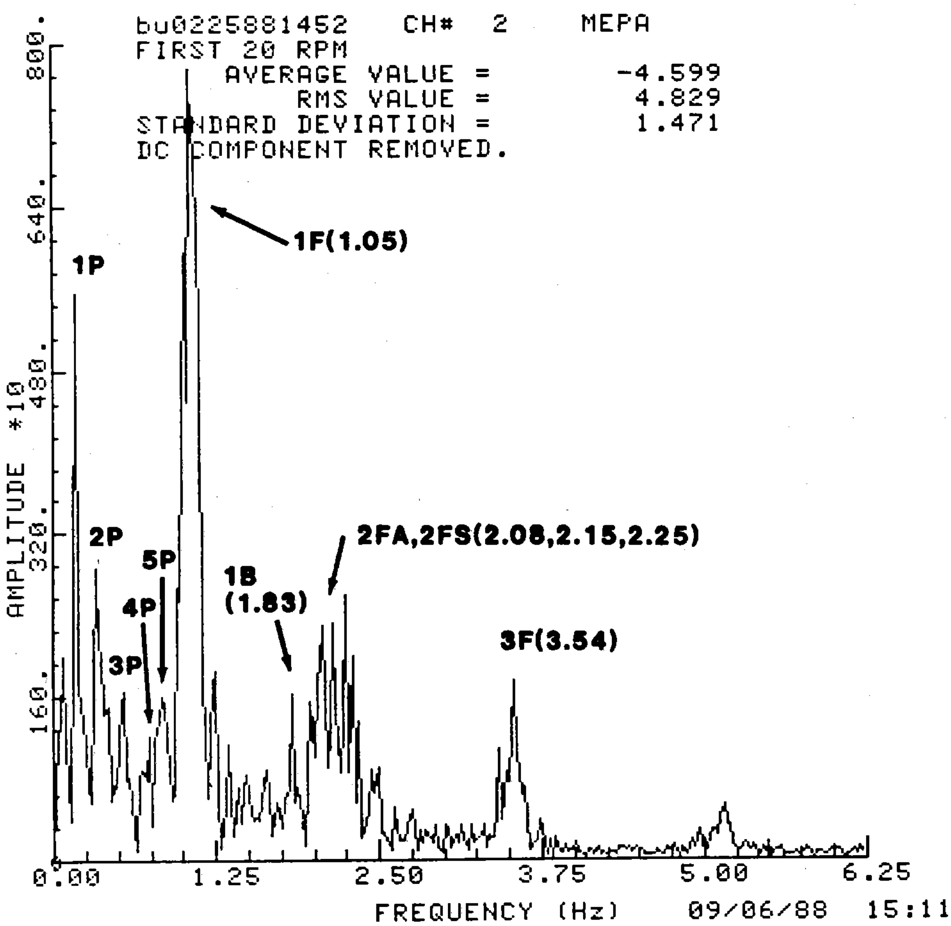

Fig. 8. Strain Amplitude Spectrum at 10 RPM

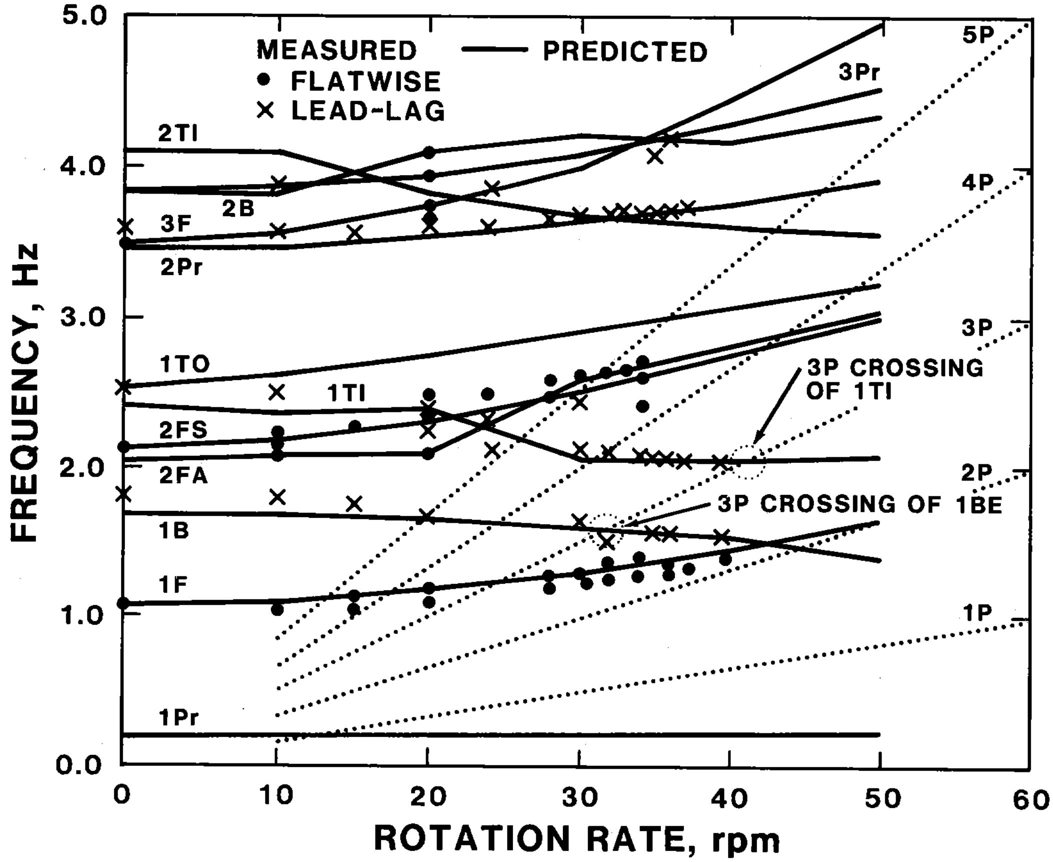

Fig. 9. Rotating Modal Frequencies - Measured vs. Predicted

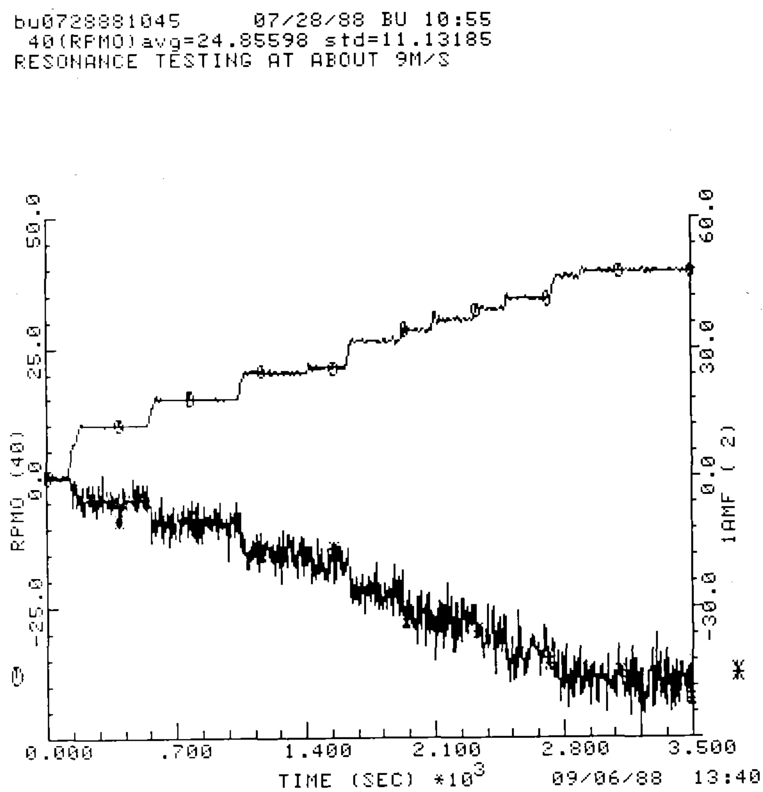

Fig. 10. Upper Root Flatwise Stress vs. RPM

Figures 11 and 12 show a comparison of the measured flatwise centrifugal bending stresses at 28 and 40 rpm to those determined analytically. The stresses along the blade from top to bottom are plotted left to right on the x-axis. There is excellent agreement between the measured and predicted values. Comparisons have also been made at 10,15,20,32,and 36 rpm, and similar agreement is observed.

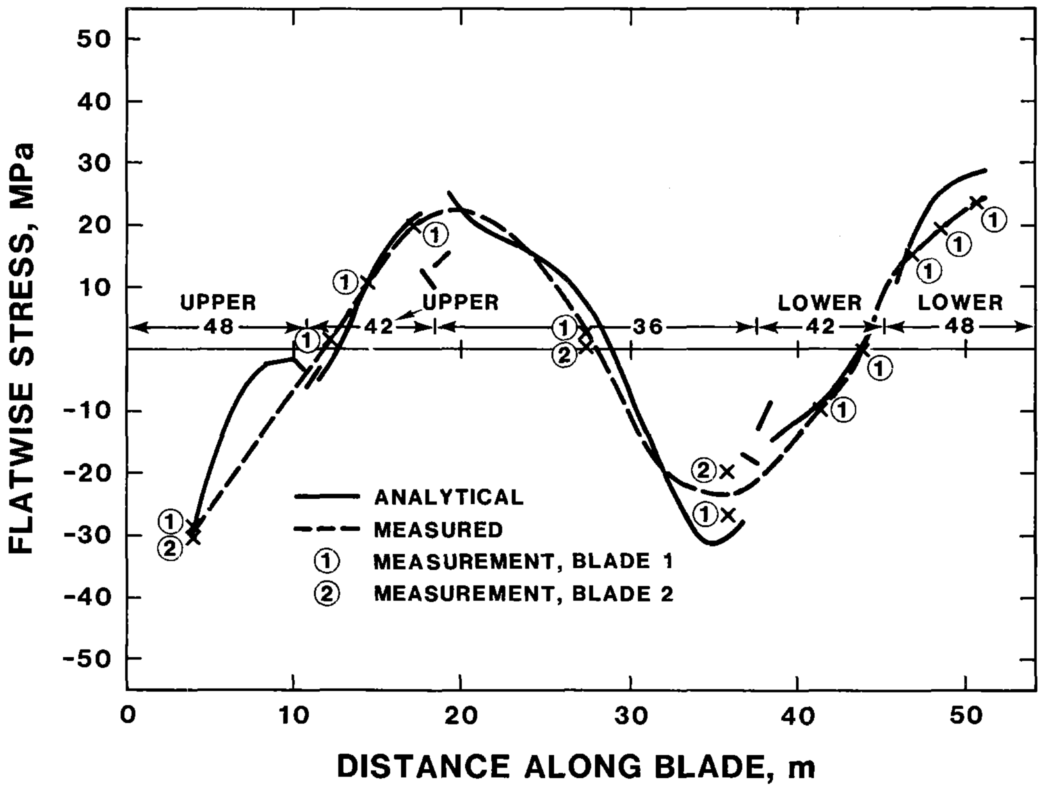

Fig. 11. Centrifugal Stress Distribution at 28 RPM

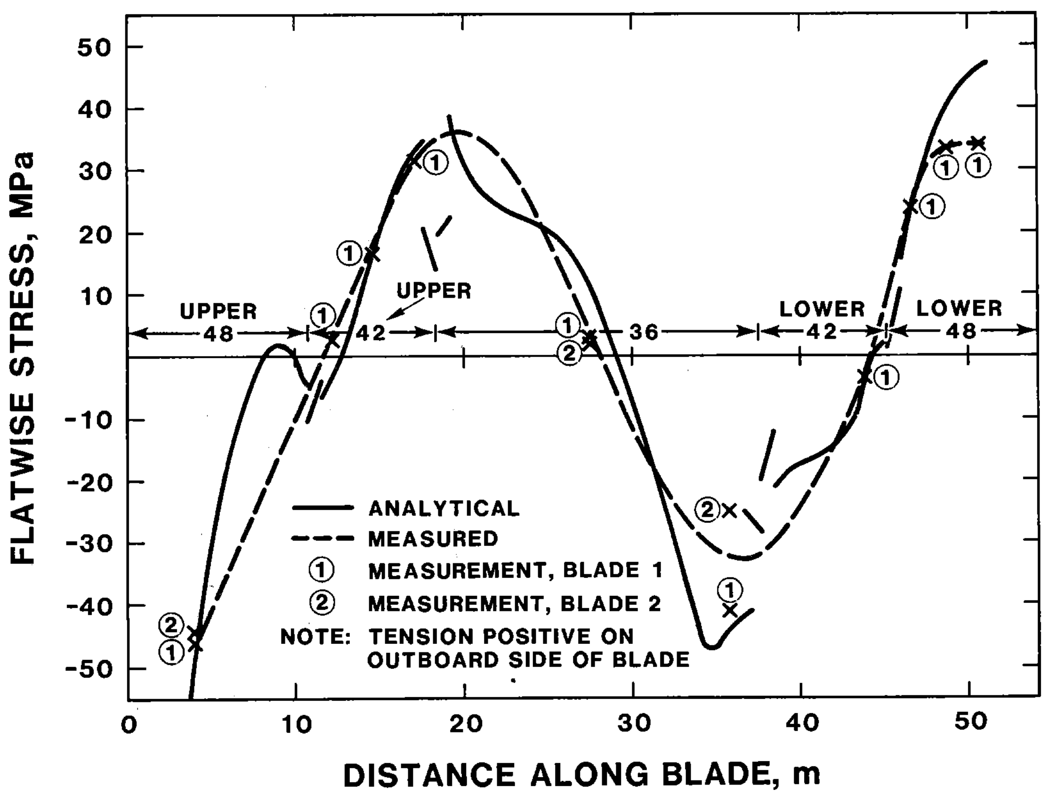

Fig. 12. Centrifugal Stress Distribution at 40 RPM

OPERATING STRESSES

Standard deviations of stress time histories for all recorded channels were computed for segments of data at various rotation rates and wind speeds. This information was used during the resonance surveys for diagnostic purposes (qualitative only) to determine the presence and severity of resonances. It requires hours of time series records at each rpm of interest when using the Method of Bins to obtain quantitative operating stress response and turbine performance as a function of wind speed. Sufficient averaged data are now becoming available for comparisons to predictions.

SUMMARY

At this early stage of testing, the turbine is responding structurally as expected. Predicted gravity stresses compare very well to measured values over most of the blade. Small adjustments to the model's blade attachment angles, if justified, may result in even better agreement.

The first guy cable mode was measured at its design frequency. Interaction between the heavy cable attachment hardware and the cables causes the second and higher modes to be lower than predicted because of the design assumption of a uniform cable mass distribution. A more exact cable model that includes the proper mass distribution of the attachment hardware has recently been developed. The second cable mode appears to become excited at two per-rev around 37 rpm.

A modal test was performed on the stationary rotor. The measured modal frequencies were compared to analytical predictions with excellent results. Rotating modal frequencies obtained from strain amplitude spectra also agree closely with predicted values.

Measured centrifugal stresses have been compared to analytical predictions at several rotation rates, and again, agreement has been excellent.

The close agreement between measured and predicted modal frequencies and mean stresses indicates that the finite element model accurately represents the 34-m Test Bed structure.

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