THE BOEING MOD-2 WIND TURBINE SYSTEM ROTOR

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INTRODUCTION

In a preceding discussion, Mr. Richard Douglas described the several very important trade studies which led to the final design stage of the MOD-2 Rotor. The following discussion includes the design details, significance of fatigue strength, design development test results, and conclusions of the preliminary design efforts.

SPECIFICATIONS AND REQUIREMENTS

The outstanding configuration requirement for the MOD-2 Rotor is the 300 foot diameter listed in Figure 1. It may well be the practical size limit of the future wind turbines even though no specific restraints were encountered in its design development. Rotor bending frequencies do, however, tend to reduce with increasing span and the Rotor, drive train, and tower frequencies, become more difficult to seperate from the two and four per rev forcing frequencies.

A second noteworthy feature of the MOD-2 is the Rotor's controllable tip. The 30% semi-span tip for each blade involves a bending moment which must be carried through the spindle. This bending moment is large enough to present interesting considerations in the detail design of the spindle assembly and its support.

The design load conditions are derived from both the environmental and functional requirements and may be categorized as being normal operating, operating fault, and non-operative. Both the limit design (static load) and the cyclic conditions of normal operation involve essentially the same environmental factors, i.e., steady winds, wind gradients, and gusts. The normal operation therefore governs the approach to fatigue restraint design while the non-operative conditions, such as the extreme winds, are the critical aspects of the buckling resistant design criteria.

Cost penalties associated with a minimum weight rotor were avoided by using the "soft" tower concept. Although the rotor speed passes through the rotor/nacelle/tower combined natural frequency during startup and shutdown, the dwell times are short and not detrimental. Weight restraints on the rotor are less severe than if the tower were "stiff" designed.

In order to assure the adequate fatigue strength for the 30 year lifetime of the Rotor, the number of joints designed into the skin panels is being held to a minimum. The number of joints is minimized by the use of maximum material sizes, consistent of course with forming capabilities. For general cost reduction, there is a requirement inherent in the MOD-2 Rotor for design simplicity, an inexpensive, easily welded steel alloy, and a minimum number of parts.

DESIGN

The final MOD-2 Rotor configuration, MOD-2-107, is composed of three sections as shown in Figure 2. The view is taken from the nacelle side and shows the down-going portion of the Rotor. The tip section is 45 feet, the mid section is 75 feet and the hub section is 60 feet long. The tip chord is 56.6 inches and the maximum chord, throughout the hub, is 136 inches. The maximum airfoil thickness varies from 6.78 inches at the tip to 57 inches at the hub.

When the controllable tip section is in the high drag, or feathered, position the tip section trailing edge is down-wind as shown in the small view labeled "High Wind Attitude". Station 360 is made into a bolted field joint to permit interchangeability of blade elements in case of damage and to facilitate shipping. The hole in the near side of the Rotor at Station 0 is the accommodation for the teeter assembly and stub of the low speed shaft.

The spindle installation is shown in Figure 3. The angle of attack is controlled by means of the tip actuator pushing against the stub fitting on the trailing edge of the tip. The spindle assembly is pressed into rib fittings in the tip and bolted to the inboard rib of the tip. The spindle rotates on two lubricated roller bearings located in the outboard end of the mid section. Access to the bearings for maintenance and to the control system is gained through small doors and hatches in the low stress areas of the compression skin panel. A non-structural fairing covers the entire cavity between the tip and mid sections.

The mid section is characterized by a very long, simple, formed skin construction terminating in a machined rib at Station 360. Considerable analysis has gone into the design of the back-to-back field joint ribs at Station 360 because of the obvious fatigue-critical nature of such a joint. The thickness of the joint flange is 1.25 inches.

The hub section completes the transition from the air foil shape of the blade to the non-lifting contour shown in Figure 4 in the central region of the hub. The hub section contains the teeter assembly which is crucial to the relief of high frequency cyclic loading to the low speed shaft. The Rotor is actually supported by two elastomeric bearings, which in turn transmit the dead weight, shear, bending and torsion into a teeter trunnion. The trunnion is welded to the stub of the low speed shaft. A teeter stop limits the travel to $\pm$ 5 degrees, and an associated teeter brake will hold the Rotor and prevent flapping motions when the Rotor is parked.

The field joint on the low speed shaft in Figure 4 is used for final assembly when the Rotor is hoisted horizontally 200 feet up to the nacelle. All of the electrical, hydraulic, and pneumatic subsystems enter the Rotor from the inside of this low speed shaft.

DESIGN PROCEDURE

The evolution of the MOD-2 Rotor to the status in preliminary design used load and material thickness iterations varying from rather simple analyses to sophisticated structural dynamics computer programs. For the Rotor geometry selected as a result of the trade studies early in the program, limit wind conditions corresponding to a hurricane acting uniformly across the Rotor disk and 99.99% cumulative probability of occurrence of gusts during operation were used to generate a bending moment distribution. The plate thicknesses, based on certain plate lengths (joint locations), were calculated from static buckling allowables and, with the resulting stiffness distribution, new iterations of loads, thicknesses, joints and allowables were performed. Since fatigue considerations were paramount, the following discussion of fatigue load spectra, fatigue allowables based on a fracture mechanics approach, and pre-flawed specimen testing summarizes the procedure to obtain a conservative fatigue resistant design.

Cyclic blade loads are caused by rotation in a gravity field, wind gradient with yawed flow, by wind gusts, and by startups and shutdowns. The magnitudes of these cyclic loads are dependent on Rotor characteristics, such as weight, hub restraints (teetering), lift and drag forces, and natural frequencies. They are also dependent on wind characteristics, such as steady operating wind speeds and associated turbulence. For the MOD-2, all wind turbulence up to and including gusts having a 99.9 percent cumulative probability of occurrence are included in the design environment. The number of cycles of loading are dependent on the design life of the MOD-2 (30 years), the rotating speed (17.5 RPM), the probabilities of occurrence of various wind conditions, and the ability of the MOD-2 design to attenuate to insignificant levels the response to low and high frequency turbulence. The attenuation is accomplished with an active control system for low frequency gusts, and the teetered hub for high frequency gusts. The resulting fatigue stress spectrum for one point on the blade is represented schematically in Figure 5. Three types of stress cycles are defined. Type I cycles are due to rotation in wind gusts at a given steady wind speed. Type II cycles are steady stress transition cycles due to gusts. Type III cycles are due to WTS startups and shutdowns.

The allowable fatigue stresses are calculated using the fracture mechanics pre-existing flaw approach. This assumes a crack-like defect exists in each critical area of the structure from the first day of operation. The behavior of the assumed pre-existing crack is characterized by a crack growth model which predicts the growth of the crack from initial size to failure. The crack growth model utilizes the stress intensity concept in predicting the crack growth behavior. The relationship between the characteristic initial stress intensity and the number of cycles to failure is shown in Figure 6. The initial flaw assumed in the analysis is larger by a factor of 2 or 3 than the minimum size which can be reliably detected during normal inspections. The conservatism in the initial flaw size assumption, therefore, translates to a significant margin in both allowable stress and life.

Verification of the crack growth model used in the determination of allowable stress levels was accomplished by spectrum load tests of pre-flawed (0.25 X 0.05 inches) specimens. A total of four different spectra were tested in order to provide a data base which would encompass variations in design load spectra.

Correlation between predicted and actual results is presented in Figure 7 where each bar represents a test point. The data points are evenly disbursed about 1.00, with an average correlation of 1.00, which means the actual and predicted stress are identical. The majority of the tests were conducted on other ASTM A Type steels. The excellent correlation between predicted and actual results verifies the applicability of the crack growth model for wind turbine type load spectra and A Type steels.

Although the same number of discrete load cycles are applied at each point along the Rotor, the magnitudes of the steady and alternating stresses vary from point to point. Since the allowable stresses are determined by establishing the crack growth behavior of an assumed initial flaw, the allowable stress changes as the imposed stress spectrum changes. There is a significant difference in stress spectrum at the hub relative to that at the tip of the blade. The allowable stress level by Rotor Station presented in Figure 8 reflects the changes in stress spectrum. The step decrease in allowable at Station 90 is an added conservatism to account for the redistribution of loads because of the change in sections at that station.

With respect to the maximum static load conditions, the Rotor is critical in buckling at several locations. The combination of curved and flat plates in a bending situation, such as exists in the MOD-2 blade preliminary design, presents an uncertainty in determining the end fixity of the curved panel and therefore the determination of the buckling strength of the section. The analysis for the MOD-2 blade design assumed a conservative approach to determine the buckling allowables. A blade bending test was desired to verify the analysis methods used for buckling under axial and bending compressive loading.

A full scale section representing the MOD-2 blade at preliminary design was prepared using the forming and welding procedures developed for fabricating the prototype Rotor. This section, 35 feet in length, represented the geometry from the Station 360 flanged field joint to Station 780, as shown in Figure 9. In the interest of economy, selected plates were standard thicknesses and therefore deviated slightly from preliminary design sizes. This had no effect on the objective of the buckling program.

An applied load transfer rib was added at one end and a skin thickness increase was built into the transition region at the root of the specimen. The test specimen was fabricated by the MOD-2 manufacturing shop and was available for the test on November 3, 1978. Test fixtures were designed and fabricated which would permit interfacing the specimen with the structural test strongback. Instrumentation of the specimen, installation on the strongback, and the completion of the test set-up is shown in Figure 10. The test consisted of applying a panel compression in the form of a couple and a shear load at the outboard end on the specimen. These loads were applied simultaneously and as percentages of the test load.

The specimen was subjected to 148% of the predicted ultimate strength without evidence of buckling. The primary objective the test was accomplished in that the buckling analysis method was shown to be conservative. The preliminary design of the blade was shown to have comfortable margins from the standpoint of buckling.

Another important aspect of the development program was the actual manufacturing experience. Considerable confidence and knowledge was gained in the forming, welding, and handling of the blade elements. Since the blade section was not tested to destruction, it was also possible to perform a post-weld stress relief heating cycle. Measurements made before and after the heating cycle showed no change in the specimen shape.

CRITICAL FACTORS

The concept of forming the skin panels of an all-steel, all-welded Rotor into an airfoil shape on large brake presses leads to great simplification in the overall manufacturing process. The feasibility of such operations was established on the MOD-1 blade. The technique has been improved on the development section for the MOD-2 Rotor.

The fatigue resistance of the MOD-2 Rotor joints is also of considerable importance. In addition to flaw growth development testing, and extensive fatigue load spectra analysis, a manufacturing plan has been developed which will permit fabrication of the critical tension skin joints in tooling with easy access for inspection and repair. At other places in the Rotor where ultrasonic and radiographic inspection are not feasible, Class C weld allowable stresses have been used in sizing structure.

COST DRIVERS

There are several factors that figure prominently in the cost of fabricating the MOD-2 Rotor. They are, not necessarily in order of importance; tooling, weld quality, weld length, and machined parts. There have been concerted efforts to reduce the cost of each of these in relation to the others, since all are interrelated.

The significant efforts on each were:

1. Tooling
   Adequate tooling to provide the necessary support for skin panel assembly and hard points at the ends of each section while minimizing final assembly tooling.

2. Weld Quality
   Reduction in the number of weld joints requiring ultrasonic and/or radiographic inspection, and maximizing the number of fillet versus groove welds.

3. Weld Length
   Finding suppliers who could furnish large plates to minimize the number of panel joints as well as finding the largest brake presses available for forming wide panels.

4. Machined Parts
   Minimizing the number of machined parts by use of welded assemblies and by simplifying the shape of the machined ribs to allow the use of two-axis milling machines.

MAJOR PROBLEMS

There are no major problems at the present time and none are foreseen. There is a need to proceed to prove the adequacy of design and cost analysis by actual fabrication and operational experience.

CONCLUSIONS AND RECOMMENDATIONS

It is concluded there are no specific restraints relative to maximum Rotor diameter. A controllable tip is the most significant complexity in the design of a large diameter Rotor. The tension skin joints are the most critical weldments in the Rotor and will be inspected for conservatively small flaws. The compression skin panels have a conservative buckling allowables as demonstrated by development test.

The bending frequency of the tower has been deliberately designed below that of the Rotor. This separation of frequencies is adequate enough to permit stiffness and weight changes in the Rotor which do not cause a deleterious coupling of the Rotor and tower at the two per rev and four per rev forcing functions.

It is strongly recommended that rotor, drive train, and tower designs proceed concurrently. Design decisions for each, which interact with other systems, may be made at least-cost and least-impact on any one component.

- External configuration requirements
  - Rotor diameter ≥300 ft
  - Airfoil contour = NACA 230XX
  - Twist about 50% chord = -2.5° to $+4^{\circ}$
  - Controllable tip = 30% semi-span
  - Pitch control = $+5^{\circ}$ to -97°
  - Teeter = $\pm 5^{\circ}$
- Environmental requirements
  - Design rotational speed = 17.5 rpm
  - Cut-off wind speed @ hub = 45 mph
  - Steady winds plus gusts
  - Lightning, temperature, precipitation, projectile
  - Non-operative: snow, ice, extreme winds
  - Handling and transportation
- Internal design requirements
  - Weldable, low cost steel construction
  - Commercial tolerances
  - Limit operating loads
  - Fatigue loads, 30 year life
  - Operating fault loads: overspeed, inadvertant feathering and braking

Figure 1. Specifications and Requirements

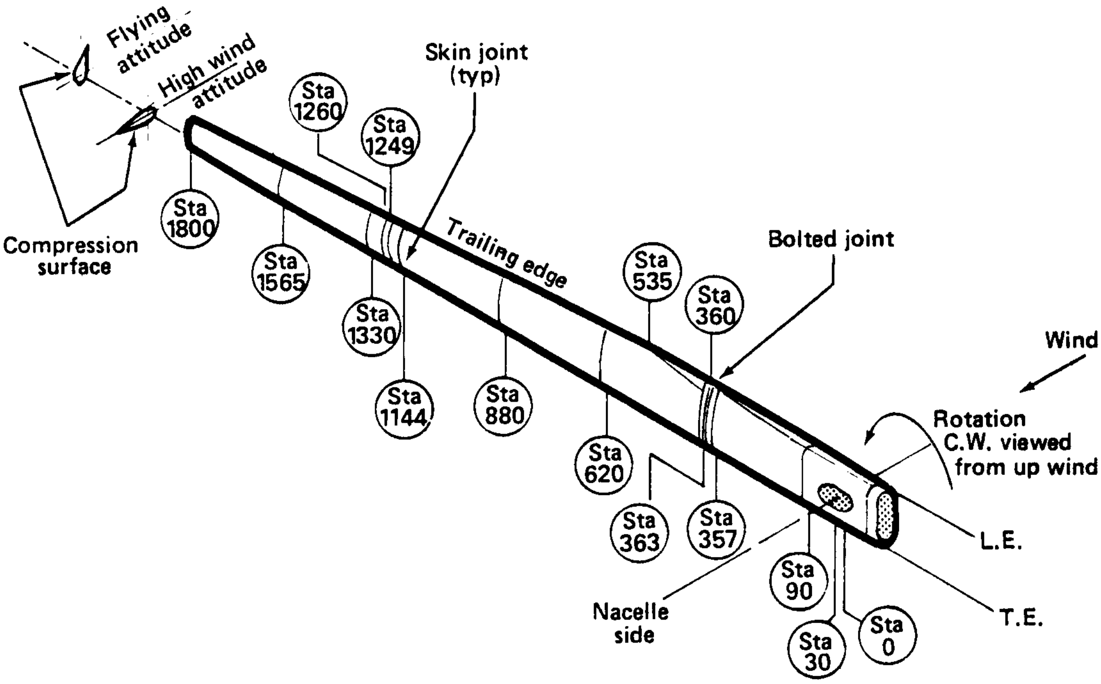

Figure 2. Rotor Configuration MOD-2-107

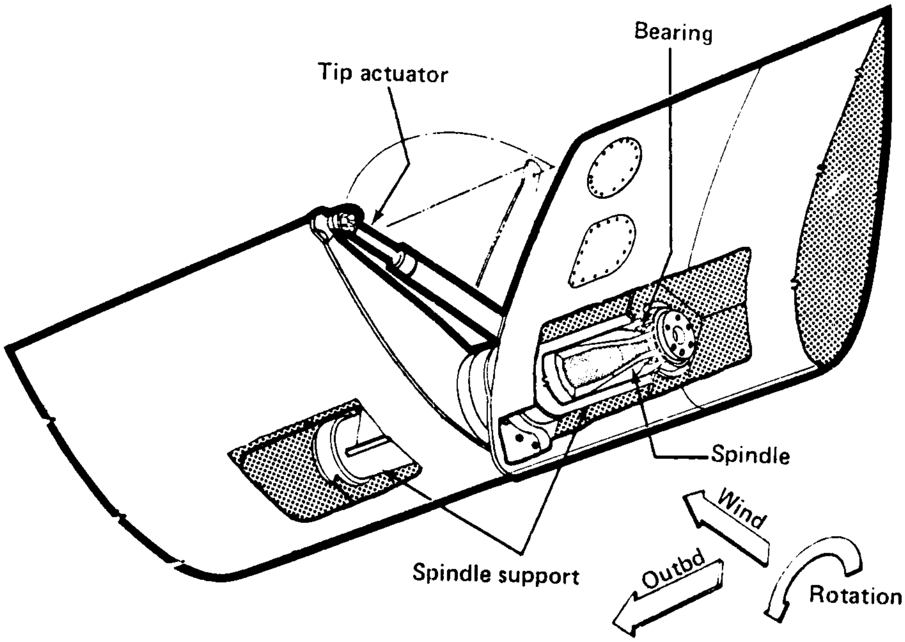

Figure 3. The Spindle

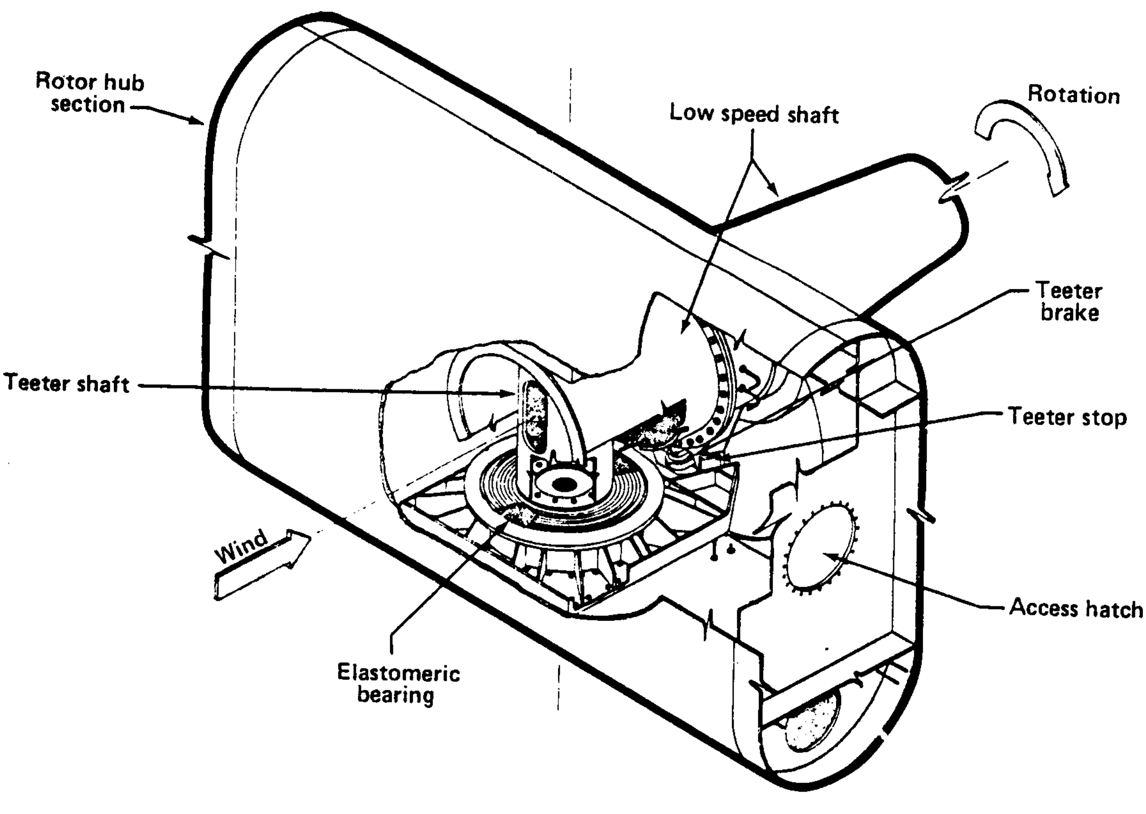

Figure 4. The Hub

For each point on the rotor:

Type I - one per rev alternating stress during gusty winds

Type II - steady stress transitions due to gusts

Type III - startup and shutdown stress cycles

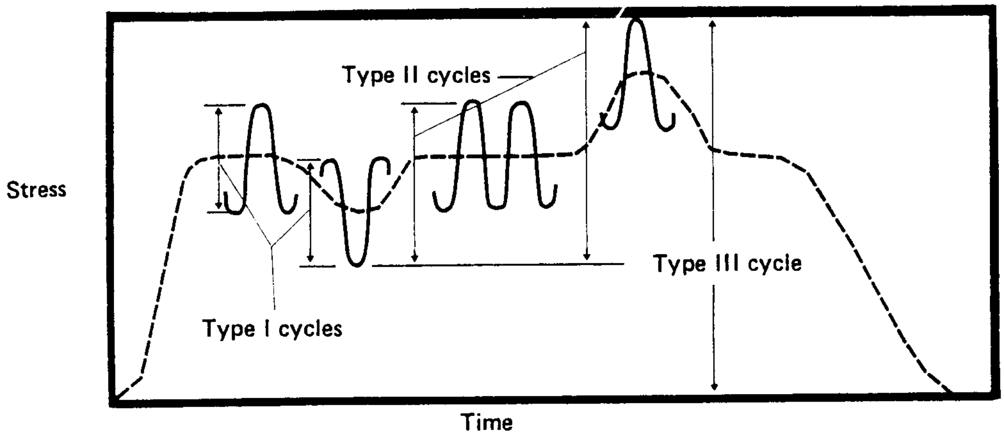

Figure 5. Rotor Blade Fatigue Spectra

- Model derived from spectrum load test results
- Model is good for all A type steels
- Model applicable to all wind turbine spectra

Crack growth model

- Accounts for threshold effects
- Accounts for retardation effects
- Predicts constant amplitude data

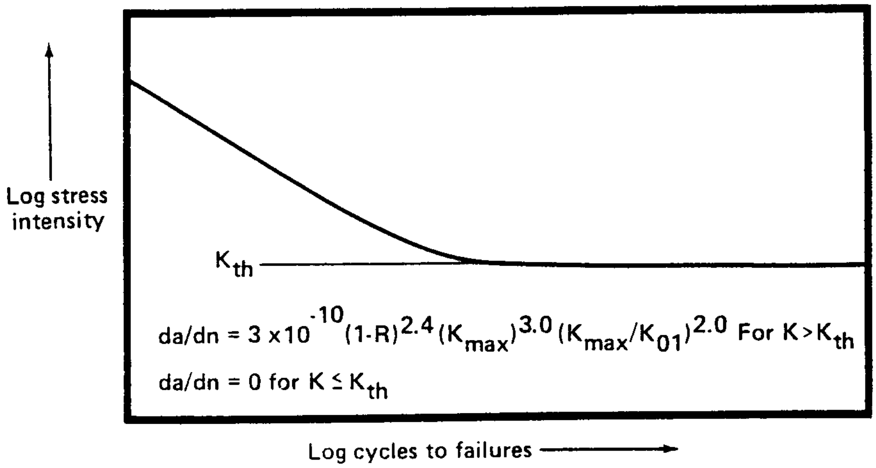

Figure 6. Fatigue Allowable Model

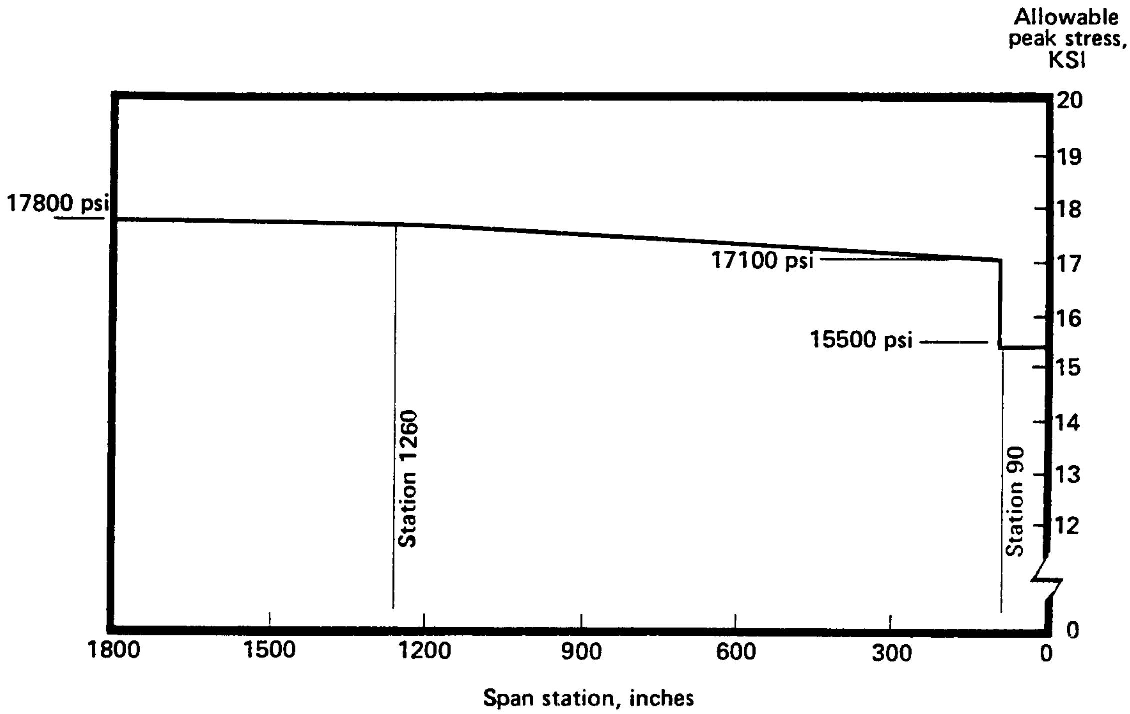

Figure 7. Rotor Fatigue Allowable Stresses

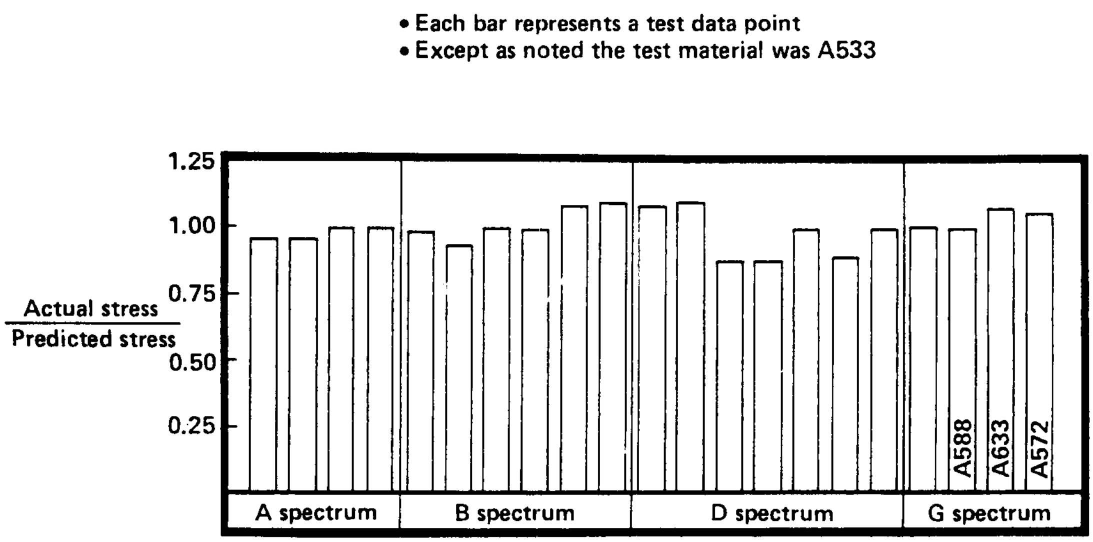

Figure 8. Correlation of Test and Predicted Results

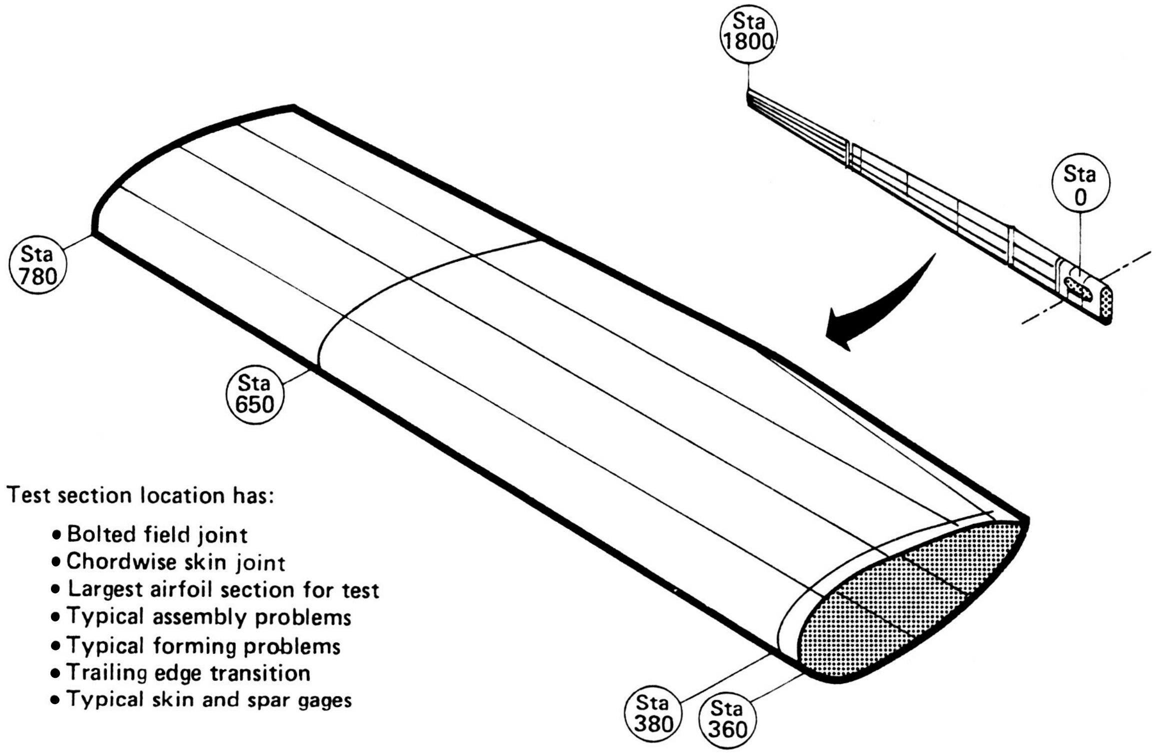

Figure 9. Rotor Blade Development Bending Test Specimen

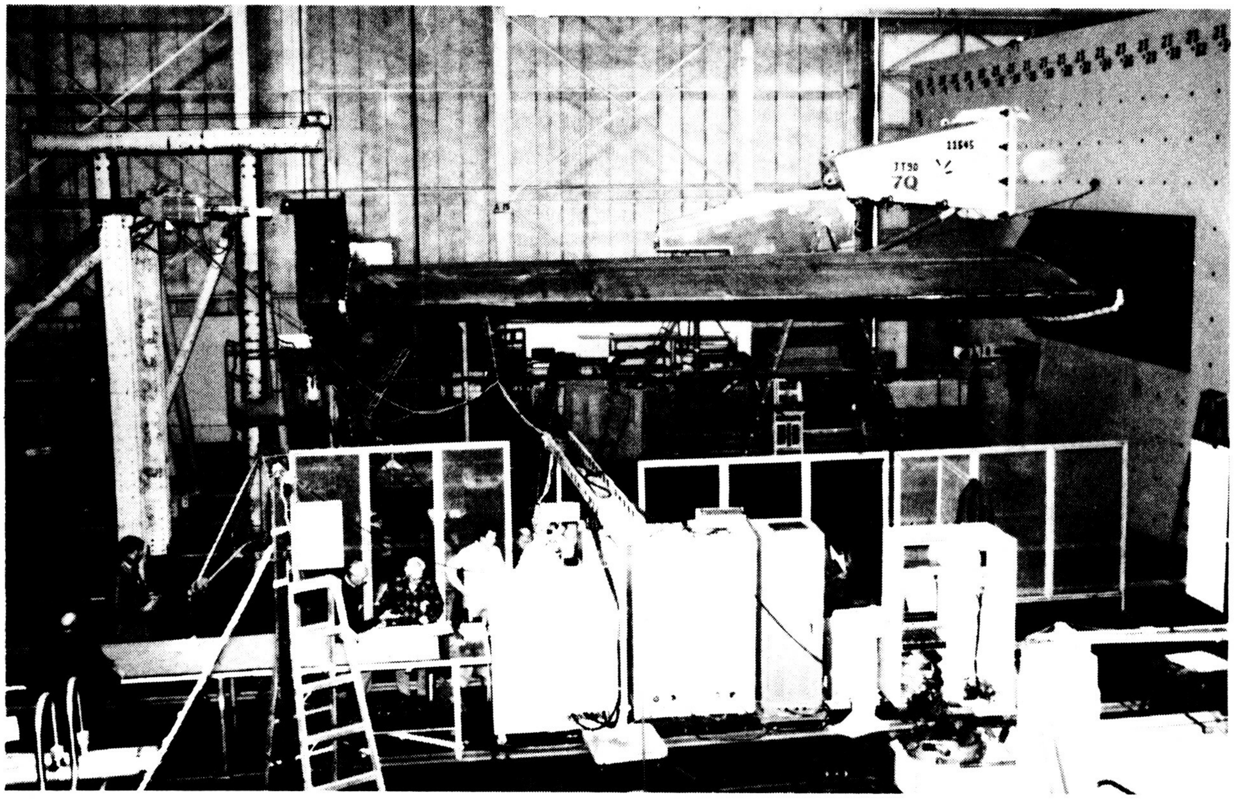

Figure 10. Rotor Blade Development Bending Test