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# Composite Turbine Blade Design Options for Claude (Open) Cycle OTEC Power Systems

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## ABSTRACT

Small-scale turbine rotors made from composites offer several technical advantages for a Claude (open) cycle ocean thermal energy conversion (OTEC) power system. Westinghouse Electric Corporation has designed a composite turbine rotor/disk using state-of-the-art analysis methods for large-scale (100-$MW_e$) open cycle OTEC applications. Near-term demonstrations using conventional low-pressure turbine blade shapes with composite material would achieve feasibility and modern credibility of the open cycle OTEC power system. Application of composite blades for low-pressure turbomachinery potentially improves the reliability of conventional metal blades affected by stress corrosion.

## INTRODUCTION AND BACKGROUND

The design of a cost-effective commercial application Claude (open) cycle OTEC power system requires the design, development, and demonstration of a relatively large, low-speed steam turbine. Because of the low absolute pressure of the open-cycle steam, the consideration of pressure losses is very important. Generally a single turbine is chosen to eliminate the pressure losses associated with the manifolding required for multiple turbines.

Westinghouse proposed a prototype turbine design in 1979 (1) that was nominally 43.6 m in diameter, rotated at 200 rpm, and generated 144-$MW_e$ gross (100-$MW_e$ net) power. Presently the largest diameter for a low-pressure stage in a conventional power plant is on the order of 4.5 m. Larger turbine blades are not currently manufactured because, at the present time, outputs in excess of 1300 MW are not required on a single shaft. Recent work (2) completed by Creare R&D, Inc. has shown that the cost of the state-of-the-art open cycle OTEC plant is not very sensitive to the number of turbines used. If the existing low-pressure turbines are adapted to OC-OTEC applications and used instead of a single specially designed turbine, the total cost of the plant increases by only 10%. Creare's results show that large turbine development should not be the critical issue for initial demonstration of OC-OTEC feasibility. Existing turbines of known cost and performance could be used at little extra expense, thus eliminating the uncertainty associated with Westinghouse's earlier turbine concept.

Although these smaller turbines are technically feasible, larger diameter rotors will eventually be needed for open cycle plant sizes greater than a few megawatts. The centrifugal stress imposed upon very long blades rules out conventional tapered solid steel rotor blades as a viable design for OTEC applications. Furthermore, even when conventional 4.5-m-diameter low-pressure rotors are adapted to OC-OTEC conditions, chloride carryover would be 100 times worse than is now marginally acceptable in conventional power plants, as reported in recent Electric Power Research Institute (EPRI) publications on low-pressure steam turbine blade failures caused by stress corrosion.

Fiber-reinforced plastics, however, are a feasible alternative to conventional steel blading. Their high specific strength and stiffness and low density make them ideal candidates for structures with mass-related stresses. The ability to mold these materials to very complex curvatures semi-independently of structural considerations allows greater freedom to optimize the blade geometry from a thermodynamic standpoint. The ability to vary the design within a given blade geometry also allows structural tuning to achieve the desired blade dynamic behavior. The structural design requirements can be met by varying material placement, fiber orientation, fiber selection, or other aspects of the structure.

The design of a rotor for open-cycle OTEC applications involves many interacting factors. The designer must analyze the blade stress levels to determine if they are low enough for safe operation. He must also perform a dynamic analysis to make sure that the blade has no self-destructive characteristics from an aeroelastic point of view. The designer must predict the geometry of the blade during operation so that one can compensate for the deflections that the blade experiences during operation. This allows the blade to have the proper geometry for maximum efficiency. Great care must be taken to design a blade that uses the least costly materials that will perform satisfactorily and to choose manufacturing processes that allow efficient use of labor. In addition, it is desirable to achieve an extended service life (up to 30 years has been suggested) in the interest of an economically competitive system cost.

In 1981, under subcontract to SERI, Westinghouse Electric Corporation working with Advance Ratio Design Co. and the University of Delaware summarized their findings (3) for an OC-OTEC turbine design. This paper expands upon the team effort by Advance Ratio Design Co. and the University of Delaware, which looked at the structural design and analysis of composite blades suitable for OC-OTEC application. Researchers derived a baseline design on which a composite prototype rotor could be based. The blades are unique in that they are long and slender (high aspect ratio) like the blades found on helicopters, yet they are highly twisted and cambered like most steam turbine blades. In addition, the supersonic tip speed creates further complications that were addressed in the analysis of this rotor system.

## LARGE-SCALE BLADE DESIGN EVOLUTION

Westinghouse first analyzed a scaled version of their existing low-pressure steam turbine blade. The blade was scaled by a factor of 3 in the chordwise direction and a factor of 11 in the spanwise direction. The scale factors were chosen based on the current understanding of the sizes required for the open-cycle OTEC turbine. This yielded a blade with a root chord of 0.6 m, a tip chord of 0.36 m, and an overall length of 12.1 m. The number of blades was set at 189.

Full-scale layouts were made at five equally spaced radial stations along the blade length. The first design approach, chosen for simplicity, had a constant thickness glass-epoxy shell around the entire airfoil perimeter with three webs and a polyurethane foam core, as shown in Fig. 1. An integrally bonded leading edge cup of stainless steel was included to provide erosion protection from any liquid drop carryover.

Under analytical scrutiny, the University of Delaware found that this design approach had an excessive twist-up of 14 deg, predominantly caused by aerodynamic loads. The twist-up was considered excessive because manufacturing differences would cause too much blade-to-blade variation. The blade would have to be manufactured with an opposite twist and designed to operate with that much deflection to achieve the thermodynamically required geometry.

Next researchers at the Advance Ratio Design Co. evaluated the effect of Kevlar and graphite reinforcements. The base design was constructed of glass-epoxy, so it was possible that some benefit could be obtained by using higher specific stiffness and strength reinforcement fibers such as Kevlar and graphite. We simply substituted the properties of the latter materials into the calculations for the glass-epoxy design without changing the geometric aspects of the design. Although it is valuable to know the effects of these changes in general, we believe that a least costly solution will most likely be achieved using the less costly glass-epoxy materials.

At this point Westinghouse, because of thermodynamic considerations, chose a new solidity. They also chose a new blade number (110) hoping that a lower aspect ratio would reduce the twist-up problem. This design had a root chord of 2.3 m and a tip chord of 1.5 m.

Westinghouse then made a set of half-scale drawings of this large chord blade using the original design concept. Since experience with the previous designs indicated that twist-up was the driving factor in the blade structural design, the University of Delaware examined this property first. The torsional frequencies seemed satisfactory but the twist-up was still excessive. Now, however, the twist-up was predominantly caused by the centrifugal twisting moment or the "Tennis Racquet Effect."

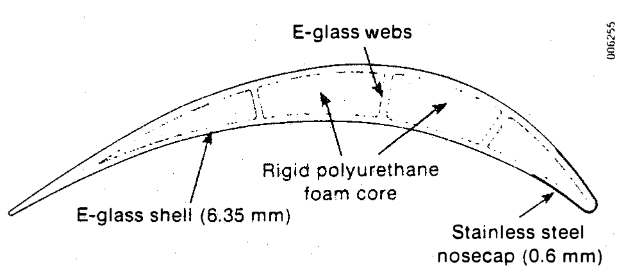


**Fig. 1. Preliminary conceptual blade design**

The most obvious solution seemed to be to increase the wall thickness; however, upon further examination of the parameters involved researchers saw that centrifugal twisting moment and torsional stiffness both increased in direct proportion to wall thickness, yielding no net improvement in twist-up for increasing wall thickness.

It was now quite obvious that a new design approach was necessary. The team chose a centrally located box spar running the length of the blade with the leading and trailing edge shapes formed by thin skins supported by a rigid polyurethane foam core. A stainless steel erosion cap was also included in this design.

This design yielded acceptable results in that it had a low twist-up because of its more centralized mass distribution. All bending and torsional natural frequencies seemed favorable for the first several modes in that no resonant or other such troublesome conditions occurred. Torsional deflections of only 1.6 deg were considered acceptable.

At this point, we found that all design concepts examined were susceptible to classical subsonic bending-twisting flutter because of the blade's relatively high aspect ratio and low twisting natural frequency.

Because of the low working-fluid-to-blade mass ratio, flutter suppression can be achieved by relatively small amounts of dynamic mass overbalance, which is possible because most of the deflection in the first mode occurs near the blade tip making a small added mass in this area very effective. Mass overbalance is achieved by adding a small fraction of the blade's internal mass in the outer 10% of the blade span near the leading edge. In this way, stable coupling between bending and twisting appeared achievable in all modal coupling patterns anticipated as flutter. Since the mass overbalance is small, it will not have excessive negative effects on natural frequency tuning, centrifugal twisting moment, and centrifugal force on the blades and disk.

At this point SERI requested that Westinghouse revisit the economic benefit of varying the blade number at constant rotor solidity. After examining several factors such as torsional deformation, leading and trailing edge stresses under static droop conditions, and total blade weight, no discrete blade number appeared clearly optimum. Westinghouse researchers thought that any "reasonable" number larger than 110 could be selected for further analysis and concluded that 220 blades would be used for the next portion of the design effort. The research team was then able to optimize the design from static, dynamic, thermodynamic, and manufacturing standpoints.

To decrease the amount of dead weight needed for dynamic mass overbalance the University of Delaware refined the box-spar design further. First the spar was swept forward as it progressed from the root to the tip, centrally located near the root and near the leading edge at the tip. Then the thickness of the leading edge cap was increased. This brought the chordwise center of gravity (CG) of the blade closer to the leading edge with the greatest change at the tip.

To obtain better performance and increase solidity, Westinghouse decided to decrease the blade taper ratio. The original tip-to-root ratio of 2/3 was changed to 9/10.

This design was completely analyzed from a dynamic standpoint with the following results:

- The blade torsional frequencies did not create difficulties with respect to forced torsional vibrations or flutter.
- The first mode of bending vibration (primarily in-plane) at one-half the normal rpm, has a natural frequency of 1/rev, which would have adverse effects on rotor mechanical stability. Stiffening of the inboard portions of the blade would be required.
- The second bending mode (mostly normal to the plane of rotation) was near resonance with 2/rev excitation.

Westinghouse used these results as guidance for the next design iteration, which used new airfoils derived from aerodynamic considerations. The calculated weight of the next generation blade was reduced by about 40% to 272 kg by reducing skin thicknesses and core density. This reduces centrifugal force and simplifies blade root-end and hub retention design.

The research team derived conceptual arrangements for a root-end restraint system using interblade spar-spacing blocks that bolt together to form a stiffening disk just inboard of the blade root-end airfoil. This makes it possible to tune blade fixities to alter blade dynamic characteristics.

## BLADE DESIGN ANALYSIS

Prof. M. Young, at the University of Delaware, carefully analyzed the aeroelasticity and structural dynamics of the evolved blade and disk design (see Fig. 2). He reviewed the aeroelasticity of composite blades with respect to statics, dynamics, transonic and stall flutter, buffeting, and other supersonic flow instabilities. Structural dynamics was divided into analysis of blade-coupled bending-bending vibrations, blade twisting vibrations, forced vibrations analysis, blade/disk interaction, and turbine-generator rotary torsional dynamics. Limited results of this final report (4) on the static and dynamic analysis follow.

### Statics: Torsional Deflection

The torsional deflection was calculated using graphic techniques. The information needed for this calculation is the net twisting moment (including aerodynamic and mass effects) and the torsional rigidity. We plotted the area under the twisting moment curve as a function of radial station starting at the tip and working toward the blade base. Next we divided the curve point by point by the GJ (torsional stiffness) value corresponding to the radial station and plotted those values. Finally, we plotted the area under the latter curve as a function of station working from the tip inboard, which yielded a plot of the elastic twist distribution for the rotor blade. This method is derived from the principle of mechanics stating that the rate of change of torsional shear strain is proportional to the rate of change of applied twisting moment. Figure 3 presents results of the torsional deflection calculations for the current configuration.

### Structural Dynamics: Bending and Twisting Vibration

The bending and twisting vibrations of slender turbine blades being considered for the OC-OTEC turbine rotor can be modeled analytically as a slender, highly twisted, rotating cantilever beam that is offset in its mounting from the axis of steady rotation of the turbine and is subject to a significant centrifugal force field.

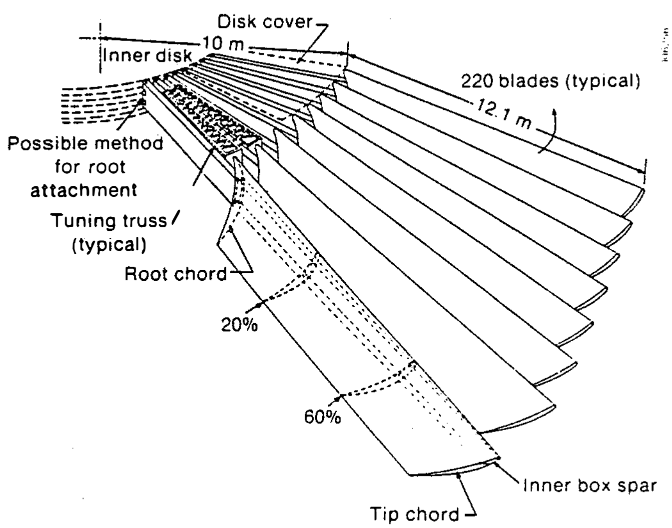


**Fig. 2.** **Artist's rendering** of partial rotor assembly

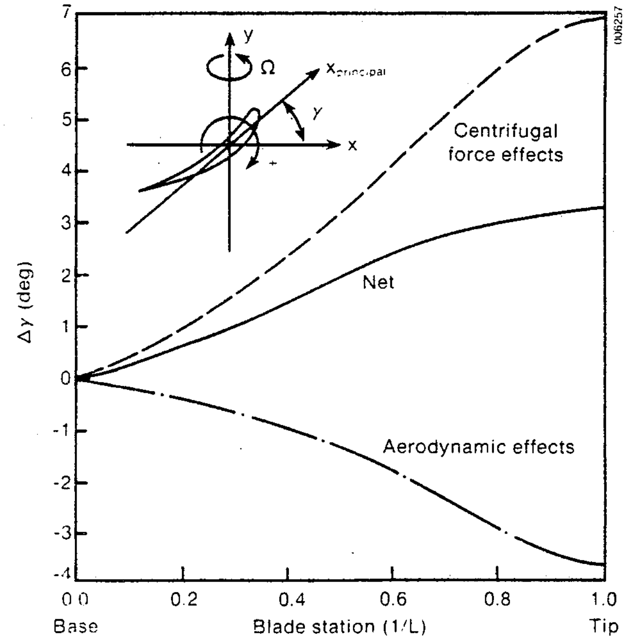


**Fig. 3.** **Torsional deformation vs. blade station (at 200 rpm)**

In the analysis, Prof. Young developed the equations governing these vibrations and then solved them analytically by a finite element-transfer matrix technique. He also developed a special-purpose digital computer program for the finite element calculations. The twisting vibrations were examined by a standard torsional influence coefficient/dynamic matrix-matrix iteration procedure. Figure 4 presents the results of these computations in the classical Campbell diagram.

The Campbell-type diagram summarizes the coupled, dual-bending modes of vibration of the swept-spar blade design. The natural frequencies of these modes are graphed as a function of the turbine rotational speed. The calculations are performed with a perfectly rigid, idealized disk that is motionless in inertial space. Accordingly, we must consider the influence of a realistic, somewhat flexible disk in interpreting these data. This is especially true for the first two blade modes, which have the smallest natural frequencies.

The second bending mode approaches the twice per revolution, straight line of excitation at normal operating speed in the diagram. This mode is mainly made up of bending motion out of the plane of steady rotation. A minor amount of inplane bending motion is caused by the built-in design twist and camber of the blade. This disk bending in the out-of-plane motion, especially the two-nodal diameter pattern, is not desired and should be avoided. This arises from the well-known frequency dispersion aspects of the coupled blade-disk interactive modes (5,6), which can result in a second harmonic resonant response of this backward wave, "wheel" mode (7). SERI believes that attention to this problem during the disk design phase and the relatively large internal damping levels (8) of a composited blade structure will eliminate or minimize this blade-disk response phenomenon.

Of equal importance, but quite different in its significance, is the fundamental mode of blade-bending vibration. This mode is mainly made up of in-plane motion and approaches the once-per-revolution excitation line near normal operating speed. Also, the dispersion effect of the coupling and interactive aspects of the disk design can, in this case, result in a natural frequency less than once per revolution at or approaching the normal operating speed of the turbine. Floating OC-OTEC systems could have additional design considerations beyond those for land-based options. The stability of the hori-

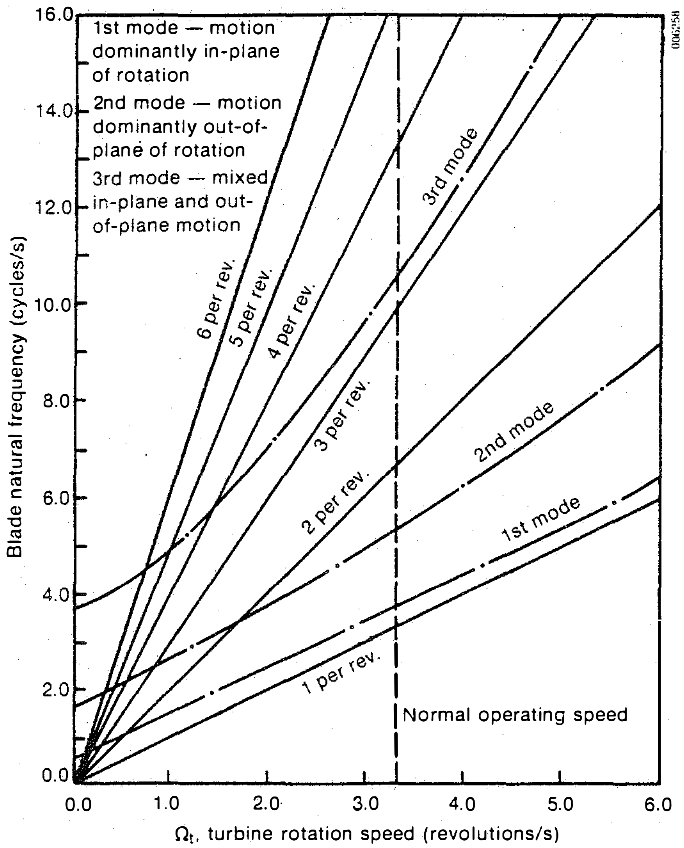


**Fig. 4. Campbell diagram**

zontal plane on which the turbine would be operating is dependent upon the damping characteristics of the vessel on which the turbine is placed. In this case the relatively low frequencies of the pitching-rolling-heaving modes of the moored hull that are less than the normal rotational frequency of the turbine can couple with the dominantly in-plane motion, fundamental mode of blade bending. The coupling of blade motion and ship's hull motion could be mechanically unstable and destructive (9,10) as in the so-called ground resonance mechanical instability of articulated rotor-type helicopters (11). In this instability, a retrograde whirling motion of the mass center of the turbine blades couples with the hull motion. The kinematic mechanism is made up of natural bending frequency of $\Omega$ turbine$^{(1-\varepsilon)}$, where $\varepsilon << 1$, becomes a frequency of $({}^{\Omega}\text{turbine}^{\varepsilon})$ in the nonrotating case. The blade motion couples in a divergent oscillation with hull modes of the same order of frequency, and much less than one cycle per revolution in the nonrotating reference system. This instability has been successfully combatted by damping the in-plane blade motion oscillations and the hull oscillations. In the case of soft-in-plane helicopter rotors (12) and in the OTEC turbine case, a significant level of equivalent viscous damping of approximately 10%-15% of critical in the blades is not achievable. We anticipate an order of magnitude less than this. Accordingly, a very significant level of damping is desirable in the hull-moored oscillating motions. Ocean engineers believe that this is achievable with a variety of hydrodynamic devices. As in the case of the disk design characteristics referred to earlier, one must plan and evolve the hull design characteristics for floating-type OTEC installations with a strong connection with the blade design. We believe that we can avoid both the potential wheel-type twice-per-revolution blade-disk interactive mode and the hull blade in-plane coupled mechanical instability. We can employ the design approach and methodologies developed thus far for land-based options eliminating this extra design concern.

### Static Droop Analysis

The team also evaluated the static droop using classical methods with zero rotational frequency and gravity loads only. Figure 5 shows that a maximum deflection of 0.38 m was projected at the tip.

### Forced Vibration Analysis

Prof. Young also carried out forced vibration response analysis using the augmented transfer matrices. He evaluated results of the bending moments for nonrotating, steady-state, and the first six harmonic cases. Figure 6 shows the steady-state bending moment solution, which appears to be within reasonable design limits for composite structures.

## THE "FINAL" CONCEPTUAL ROTOR DESIGN

The blade design that evolved from this effort is constructed of a main structural box spar with thin skins forming the leading and trading edge structures (see Fig. 2). The fiberglass reinforced plastic (FRP) box spar is constructed of 50% 0 deg fiber orientation and 50% ±45 deg orientation. The leading and trailing edge structures are formed from ±45 deg material orientation with a 112 kg/m$^3$ (7 lb/ft$^3$) polyurethane foam core. An erosion-resistant stainless steel strip, which is screwed to the blade structure for replaceability, provides leading-edge protection from droplet carryover from the OC-OTEC power system's evaporator.

The spar extends past the base of the blade to the attachment point, which is near the center of rotation. The spars form a "wagon wheel" configuration between the hub proper and the blades. The whole assembly is stabilized by inter-blade spacing blocks that connect the spars near the blade base to form the rim of the wagon wheel.

The spar is tubular in cross-section throughout its whole length. As it enters the airfoil portion of the blade, the spar begins to sweep forward until it becomes a D-spar at the tip of the blade. This helps create the proper chordwise balance needed for flutter suppression and other aeroelastic considerations mentioned earlier. After exiting the airfoil portion of the

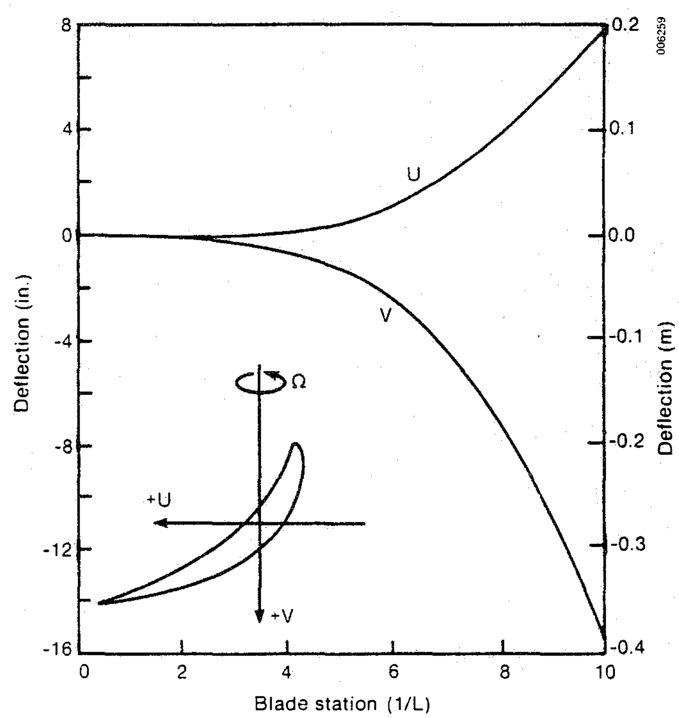


**Fig. 5. Static droop (nonrotating) vs. blade station**

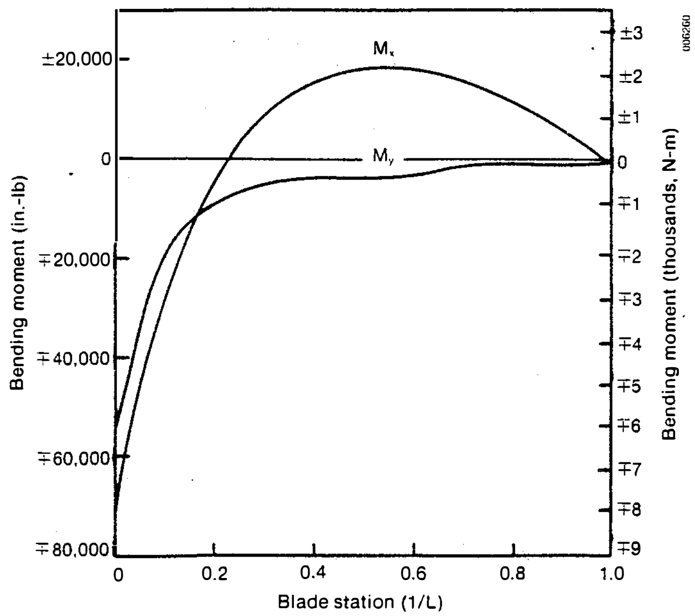


**Fig. 6. Steady-state bending moment vs. blade station** (normal operation at 200 rpm)

blade, the spar transforms into a rectangular cross-section. The spar fibers wrap completely around the retention fitting at the attachment point and continue to form both sides of the spar.

Much emphasis was put on reducing the size of the hub since the problems associated with designing, building, transporting, etc. a hub as large as 19.5 m seemed insurmountable. The minimum hub diameter was achieved using a staggered attachment bolt pattern that allowed just enough space for the spars to pass each other and places the attachment bolts on two separate circumferential rings. This yielded a hub diameter of approximately 3.25 m, still a large structure but much more manageable.

Although many details of this hub design have yet to be finalized, we feel the basic concepts outlined here provide a good basis for the development of an economic hub with high structural integrity. Industry would only pursue the subsequent development of this large a rotor if the economics of the smaller units were not cost-effective. Rotor units made out of composites for the existing size low-pressure turbines would be the next practical step toward proof of concept for OC-OTEC.

## CONCLUSIONS

The design of a large OC-OTEC turbine rotor presents many challenges to the design engineer. The large size and complex geometry coupled with the high centrifugal forces extend today's technology to its limit.

In this program we have tried to lay the foundation for the design of the OC-OTEC composite turbine rotor system. Methods for the dynamic analysis of this unique rotor system are within current state-of-the-art understanding of rotating turbo machinery, even at the 100-$MW_e$ size. These analyses allow one to predict the static and dynamic behavior of a given design so that resonances and other instabilities can be avoided. The research team analyzed candidate designs and exercised several design iterations. A baseline design was derived based on the results of these analyses and past experience gained from large rotor design and manufacture in other fields. In addition to dynamic analysis, researchers paid strict attention to the stress, material, and process aspects of the designs to assure adequate strength and manufacturability. Several special issues like vacuum environment effect on strength, moisture effects, creep, and fatigue showed positive results for the use of composites.

Since no problems arose that cause serious doubt as to the feasibility of turbine rotors of this size with this method of construction, we think that the construction of these rotors is well within current manufacturing technology. Composite blades of this size have already been manufactured for large military rotorcraft such as the Boeing Vertol Heavy Lift Helicopter.

Considering the scaling requirements (13) and reasonable engineering judgment, a small-scale turbine rotor fabricated out of composites would help demonstrate the technical feasibility of the Claude (open) cycle OTEC power system (14) and provide valuable data for the scale-up of larger composite blades.

Existing blade designs for low-pressure turbines in conventional power plants could be used as the first prototypes. The successful demonstration of the low-pressure turbine rotor prototypes with composite alternatives, having undergone a revisited structural and dynamic analysis as presented in this paper, may show that the existing turbine industry has a secondary use for their low-pressure turbomachinery. Furthermore, the use of composites may eliminate low-pressure steam turbine blade failures in conventional power plants, a problem that causes significant downtime and loss of revenue to the utilities.

## ACKNOWLEDGMENTS

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