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MOD-2 WIND TURBINE SYSTEM CONCEPT AND PRELIMINARY DESIGN REPORT

VOLUME II DETAILED REPORT

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Boeing Engineering and Construction
(A Division of The Boeing Company)
Seattle, Washington

July 1979

Prepared for
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
Cleveland, Ohio 44135
Under Contract DEN 3-2

for
U.S. DEPARTMENT OF ENERGY
Energy Technology
Distributed Solar Technology Division
Washington, D.C. 20545
Under Interagency Agreement DE-AI01-793T 20305

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1.0 SUMMARY

This report documents the work accomplished during the Concept Design and Preliminary Design phases of NASA Contract DEN 3-2. These phases were designated the Task II Analyses and Design phases of the MOD-2 Wind Turbine System (WTS) project.

The MOD-2 WTS project is a 36 month effort for the analyses, design, fabrication, assembly and installation of a wind turbine system configured for minimum cost of electricity in a region with a 14 mph average yearly wind at a 30 foot reference height. The wind turbine described in this report which meets these objectives has a 300 foot diameter rotor with the center of rotation 200 feet above the ground. The system generates electricity when the steady wind speed at hub height (200 feet) exceeds 14 mph. At 27.5 mph and higher (at hub height) the wind turbine produces the rated power of 2500 kW. Above 45 mph (at hub height) the system is shut down to avoid high operating load conditions. The annual energy output at a site with a 14 mph average wind speed referenced to 30 feet height (20 mph at the hub) is nearly 10 million kWh. This energy output combined with an estimated 100th production unit turnkey cost of $1,720,000 results in a predicted cost of electricity of 3.3¢/kWh.

The features and characteristics of the MOD-2 WTS evolved from a series of trade studies, inputs from NASA and several electric utility companies, and other analyses. Major features include an upwind rotor with a teetering hub and partial span pitch control; a planetary, light weight step-up gearbox driving a synchronous generator; and a low natural frequency (soft) steel shell tower.

Supporting analyses and test data used in the design studies are presented. Methodology to assure accuracy of such analyses and data are included. Production and construction methods and development tests are reported to show the method by which BEC will assure that the specified design will perform as predicted, and that it will meet the maintenance and reliability standards established.

2.0 INTRODUCTION

Wind energy systems have been used for centuries as a source of energy for man. At times considerable interest in developing large wind-driven electrical generating systems has existed. However, interest in these systems declined because they were not cost competitive with the fossil fuel systems in use. These efforts were also generally privately financed and suffered from the lack of a sustained research and development effort.

Recent shortages in our energy supplies coupled with the increasing costs of fossil fuels have forced the nation to reassess all forms of energy including wind power. The Department of Energy, the NASA Lewis Research Center, and Boeing Engineering and Construction (BEC) are engaged jointly in the development of a multi-megawatt wind turbine system (WTS) designated as MOD-2. The goal of the MOD-2 project is to produce a wind powered electrical generating system for utility application which is economically competitive with conventional power generating equipment. This goal specifically requires that the cost of electricity for the one-hundredth production unit does not exceed 4¢/kWh when the machine is working at a site where the average mean wind at a 30 foot height is 14 mph.

Task II of the contract consisted of concept design wherein many design configuration studies were performed to establish the configuration for least cost of electricity, and the preliminary design of the selected configuration wherein the subsystems were refined through design, cost, and failure mode and effect analyses.

To arrive at the configuration described in section 3.0, BEC evolved a baseline design and then performed many trade studies, sensitivity studies, failure mode and effect analyses, and consulted with several utility companies. These studies and how they were used to evolve the final configuration are reported in section 4.0. Supporting analyses such as structural, weights, cost, performance, safety, reliability, maintainability, and producibility are described in section 5.0. Manufacturing, assembly and test activities and plans are recorded in section 6.0. During these studies, emphasis was placed on utilizing design information and test data from other NASA/DOE Wind Turbine programs, such as MOD-0 and MOD-OA.

3.0 SYSTEM DESCRIPTION

This section describes the MOD-2 WTS(Wind Turbine System)as it stands at the completion of the Task II preliminary design phase of the project.

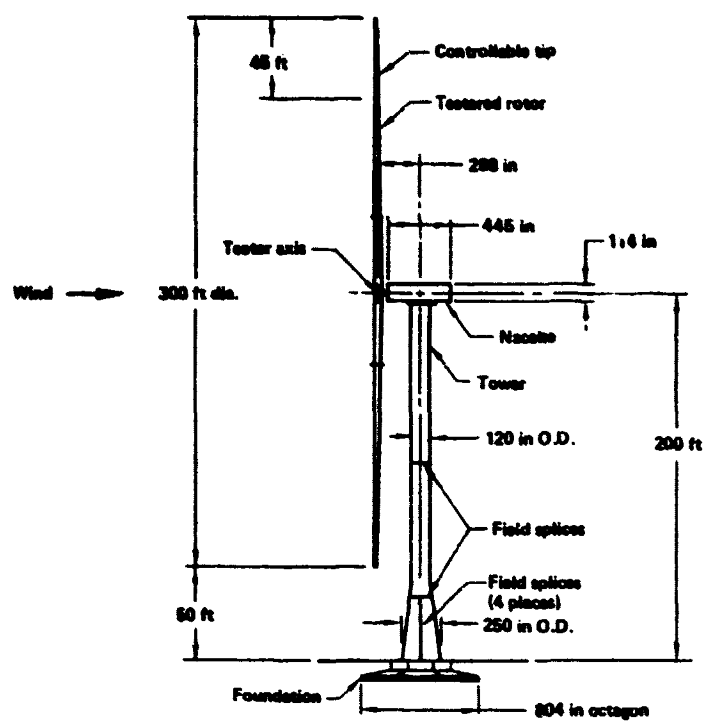
The current configuration evolved from a series of trade studies and sensitivity studies coupled with inputs from NASA, several utilities, and the results of a Failure Mode and Effects Analysis (FMEA). These studies resulted in a system optimized for a minimum cost of electricity while maintaining compatibility with current utility networks and meeting the safety, reliability and logistics requirements as described in section 5.4 of this report.

3.1 GENERAL ARRANGEMENT AND CHARACTERISTICS

The general arrangement and characteristics of the current WTS configuration are shown in Figure 3-1. It is designed for operation at sites where the annual average wind speed is 14 mph measured at 30 feet (20 mph @ hub height). The system generates electricity when the wind speed at hub height (200 feet) exceeds 14 mph. At 27.5 mph and higher (at hub height), the system produces the rated power of 2500 kW. Above 45 mph (at hub height), the system is shut down to avoid high operating load conditions. The annual energy output at a site with a 14 mph average wind speed is nearly 10 million kWh. This energy output combined with an estimated 100th production unit turnkey cost of $1,720,000 (in 1977 dollars) results in a predicted cost of electricity of 3.3¢/kWh at the bus bar. During operation, the wind turbine is tied to the utilities power grid through standard transmission lines.

The WTS is a horizontal axis machine utilizing a 300 foot diameter partial span control, upwind rotor. The rotor's center of rotation is 200 feet above ground level. It is coupled to the low speed shaft through an elastomeric teeter bearing. A 2500 kW synchronous generator is driven via a step-up planetary gear box and "soft" quill shaft. The generator, gearbox, hydraulic systems, electronic controls and other support equipment are enclosed in a nacelle mounted atop a cylindrical steel tower. The nacelle can be yawed (rotated) to keep the rotor oriented correctly into the wind as the wind direction changes. A hydraulic pitch control system is used to control the position of the movable rotor tips. The movable rotor tips are used to obtain the rated rotational speed of 17.5 rpm, and to maintain the proper power output at wind speeds above rated wind speed (27.5 mph @ hub).

The WTS is controlled by an electronic microprocessor. The microprocessor is designed to allow unattended operation of the WTS at a remote site. The microprocessor monitors wind conditions and the operational status of the wind turbine. Equipment failures result in automatic shutdown of the WTS. The systems status is monitored at the utility substation, from which maintenance crews are dispatched as needed.



Rated power	2,500 KW
Rotor diameter	300 ft
Rotor type	Teetered - tip control
Rotor orientation	Upwind
Rotor airfoil	NACA 230XX
Rated wind @ hub	27.5 mph
Cut-off wind speed @ hub	45 mph
Rotor tip speed	275 ft/sec
Rotor rpm	17.5
Generator rpm	1,800
Generator type	Synchronous
Gear box	Compact planetary gear
Hub height	200 ft
Tower	Soft-shell type
Pitch control	Hydraulic
Yaw control	Hydraulic
Electronic control	Microprocessor
System power coefficient	0.382

Figure 3-1. MOD-2-107 Configuration Features & Characteristics

3.2 SUBSYSTEM DESIGN

This section describes the basic subsystems of the wind turbine. It is divided into rotor, drive train, nacelle, tower/foundation, electronic control and electrical power system sections. The operation of each subsystem, its function, and the basis for selection of its particular characteristics are presented.

3.2.1 Rotor

The MOD-2 WTS has a steel, two bladed, teetering, tip control type rotor with continuous carry through structure at the hub. It utilizes a NACA 230XX series airfoil rotating at 17.5 rpm (275 ft/sec tip speed). The basic construction is a welded steel shell with steel spar members. These characteristics were chosen primarily on the basis of the trade studies that are presented in section 4.2.2 of this report.

As shown in Figure 3-2, the rotor is divided into three primary sections: the tip, the mid-section, and the hub section. The tip (outer 30%) is rotated with respect to the remainder of the blade to control rotor speed and power. The tip and mid-sections are the working portions of the blade. The hub section is attached to the mid-section with a field splice and is a transition from an airfoil cross-section to an oval cross-section.

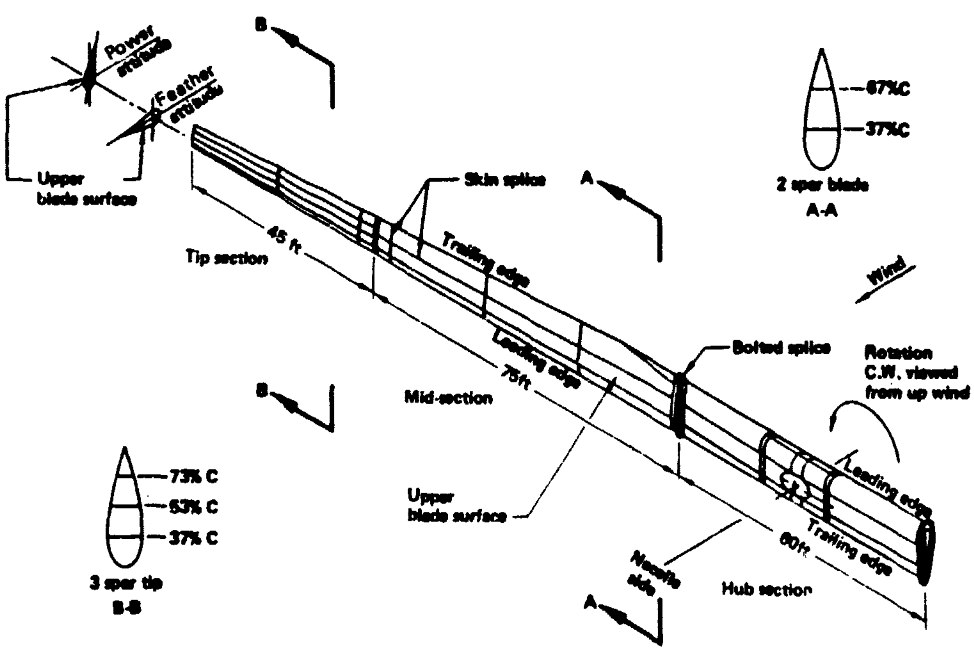


Figure 3-2. Rotor Blade Configuration

Each tip is controlled by means of a hydraulic actuator mounted on the blade mid section adjacent to the tip section as shown in Figure 3-3. The flow control to the actuators is governed by a signal from the automatic control system to servo valves. The position of the tips is monitored by position transducers and fed back to the control system. The normal rate of rotation of blade tips is from 0.1 to 1 degree per second.

In the event of a major system failure, the actuators drive the blade tips to the feathered position at rates of 4 to 8 degrees per second (depending on actuator position), using energy stored in separate hydraulic accumulators. The pitch rates are such that under any failure mode no overspeed will exceed 15% of normal rpm. Redundancy is provided by the ability of either one of the operating actuators to shut down the system should one actuator become inoperative. Locks are provided to hold the blade tips in the feathered position when the hydraulic system is depressurized.

The blade tip section with the spindle assembly and the hydraulic actuator are attached to the blade mid section as a unit (Figure 3-3). The attachment is made by 6 bolts for ease of assembly and removal. Either blade tip can be removed independently from the WTS. The spindle protrudes into the blade mid section in a way to provide a load path for centrifugal and bending moments. Tapered rings at the outboard rib and a close tolerence bushing at the inboard rib assure a tight fit between the spindle sleeve and the mid-section of the rotor. The bearings are lubricated by a long-life grease.

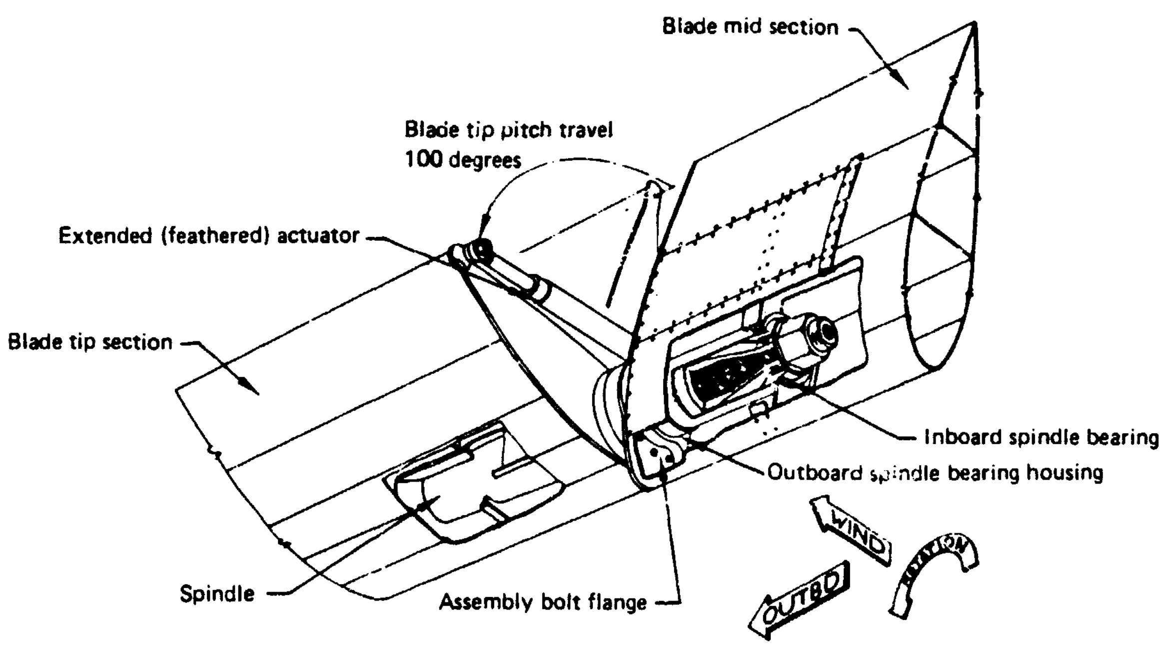


Figure 3-3. Tip Spindle Installation

The pitch control hydraulic system consists of an electrical motor driven pump, reservoir, accumulators, filters, heat exchangers, control valves, associated plumbing and actuators. Hydraulic components not located in the blade are installed on and rotate with, the low speed shaft (Figure 3-4). Electrical power and control signals are transferred from the nacelle electric power and control unit by brushes and a slip ring assembly.

Because the hydraulic system is located in a rotating environment, special attention has been extended to design for this environment. The components on the low speed shaft are exposed to approximately 1.3 g's and the reservoir has been tested in this environment with no adverse effects. The blade tip control actuator, rotating in an 11 g environment, has been selected with oversize rod and piston bearings, and is in the retracted position when exposed to this environment.

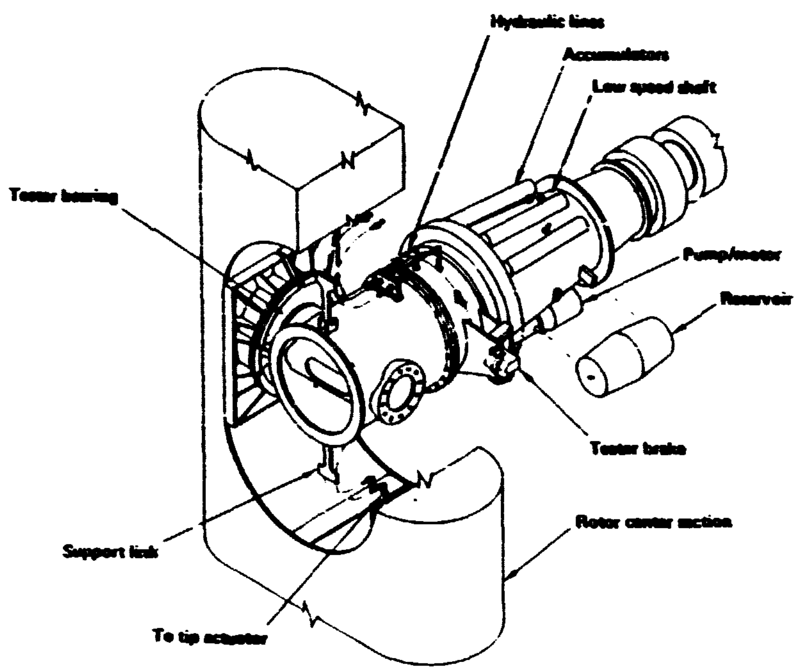


Figure 3-4. Pitch Control Hydraulics

The attachment of the rotor to the low speed shaft is by a teeter type bearing, which minimizes the one per revolution blade flapwise loads and the effects of small unsymmetric gusts, resulting in reduced fatigue loads. The teeter motion takes place in two elastomeric radial bearings which also transmit the rotor output torque into the rotor (low speed) shaft (Figure 3-5). The use of the elastomeric bearings eliminates lubrication requirements and prevents fretting that would be likely to occur if roller bearings were used. The elastomeric bearings consist of concentric alternate layers of sheet steel and rubber bonded together forming a package that is highly flexible in torsion and has sufficient radial load capability. Trade studies outlined in section 4.2.2.2 of the report showed that use of the teeter hub in place of rigidly mounted blades not only reduces the weight of the rotor assembly but also the tower and nacelle weights, thus reducing fabrication and material costs.

Stops are provided to limit teeter motion to $\pm$ 5 degrees. A teeter brake is provided to prevent rocking when the rotor is in the standby mode, and to dampen teeter excursions during starting and stopping. During erection, the rotor, with teeter bearings and upwind end of the low speed shaft installed, can be lifted in one piece and bolted in place at the low speed shaft bolt flange.

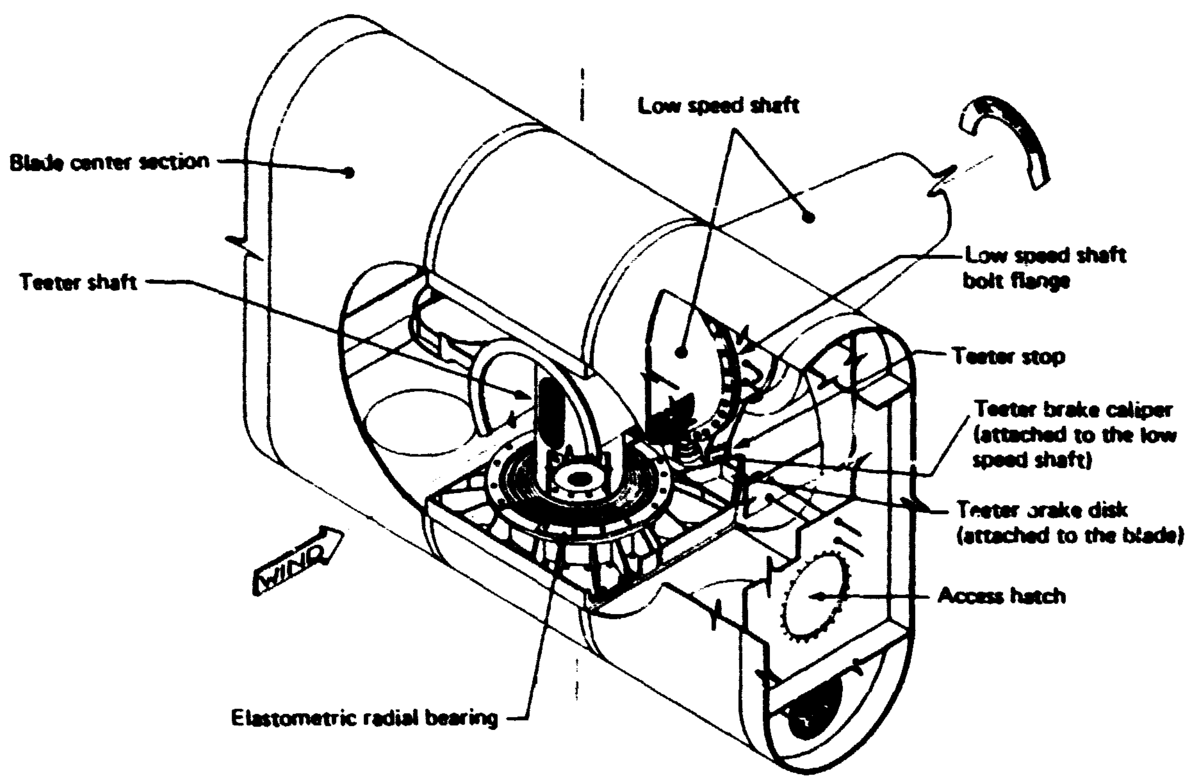


Figure 3-5. Teeter Bearing Installation

The entire rotor assembly is sealed, to allow use of a flow type crack detection system. The occurence of a significant crack in the rotor structure will result in WTS shutdown. This crack detection system is further described in section 4.2.2.8

3.2.2 Drive Train

The drive train subassembly consists of a low speed shaft, quill shaft, gearbox, high speed shaft, couplings, rotor parking brake, and generator. These major components are shown in Figure 3-6.

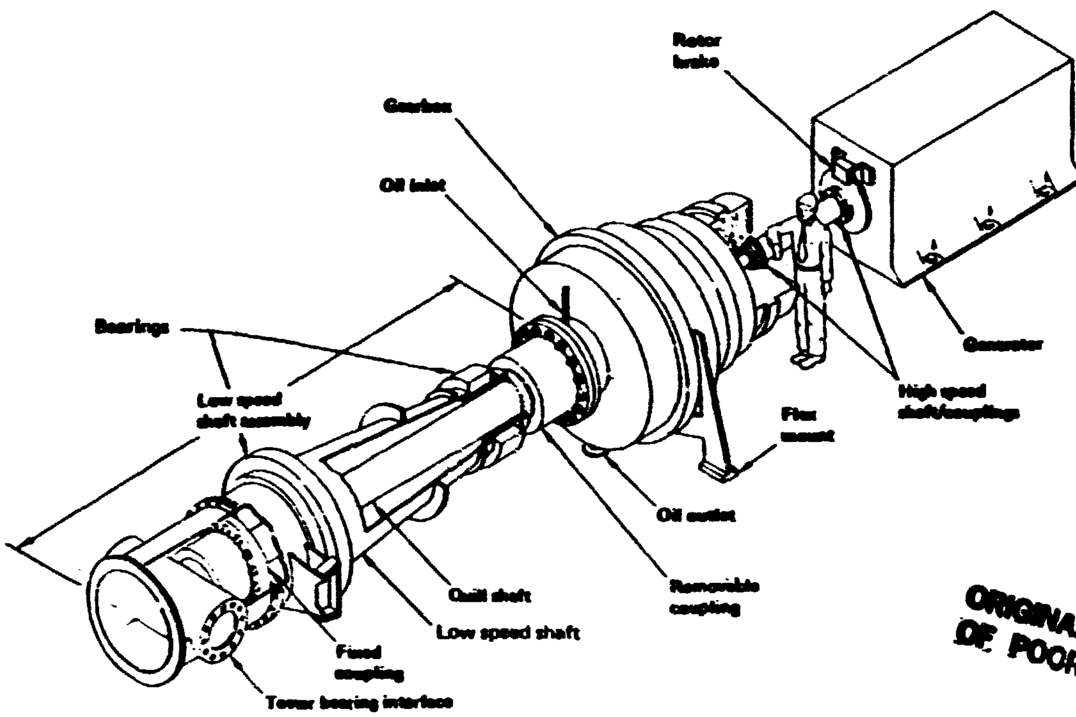


Figure 3-6. Drive Train

The large diameter low speed shaft transfers the rotor forces into the nacelle structure through the two shaft support bearings. The forward bearing supports the radial load while the aft bearing transfers both thrust and radial loads to the nacelle. The rotor torque is transmitted to the gearbox through the "soft" quill shaft. The purpose of the softness is to reduce the two per revolution rotor torque fatigue effects at the gearbox and to improve the quality of the generator output. The shaft designs were optimized through the trade studies described in section 4.2.3.2 of this report.

A 103:1 step-up of rpm from 17.5 to 1800 rpm is provided by a three stage epicyclic gearbox, which is smaller, lighter in weight, less expensive, more efficient and more tolerant to support deflections than a parallel shaft type gearbox with a similar rating. The trades involved between the epicyclic and parallel shaft gear boxes are fully described in section 4.2.3.1. The low weight of the gearbox (39,000 lbs) enhances the overall design of the supporting structure. It is flexibly mounted to the nacelle to reduce the effect of nacelle deflections on gear loads. Maintenance is enhanced by the small size of the gearbox; major repairs can be accomplished in the nacelle.

The generator is a synchronous electrical generator, rated at 2500 kW. The unit is an open frame, drip proof, four salient pole brushless machine that operates at 1800 rpm and has a shaft mounted exciter. This generator was chosen as a result of the trade studies described in section 4.2.4.1.

The rotor parking brake prevents rotor rotation when the WTS is not running. This prevents gearbox damage due to rotation without lubrication. The braking mechanism consists of a disc mounted on the high speed shaft and a spring actuated brake attached to the generator frame. The brake is disengaged by an electrically actuated hydraulic valve and engaged by spring force when the electrical circuit is open. The disc utilizes a replaceable element to minimize maintenance time in the event of brake disc wear.

As a result of the FMEA and maintainability studies a positive mechanical lock mechanism is incorporated in the high speed shaft. This prevents rotation of wind turbine during maintenance periods.

3.2.3. Nacelle

The nacelle houses the major subsystems of the MOD-2 WTS such as the drive train, generator with its accessories, yaw bearing and drive, and associated hydraulic systems for pitch and yaw control as is shown in Figure 3-7. Other equipment in the nacelle includes generator cooling air ducts, gearbox oil cooling radiators, maintenance lighting fixtures and wall plugs, electronics cooling and heating system, general nacelle air circulating system, fire protection equipment, and maintenance provisions equipment.

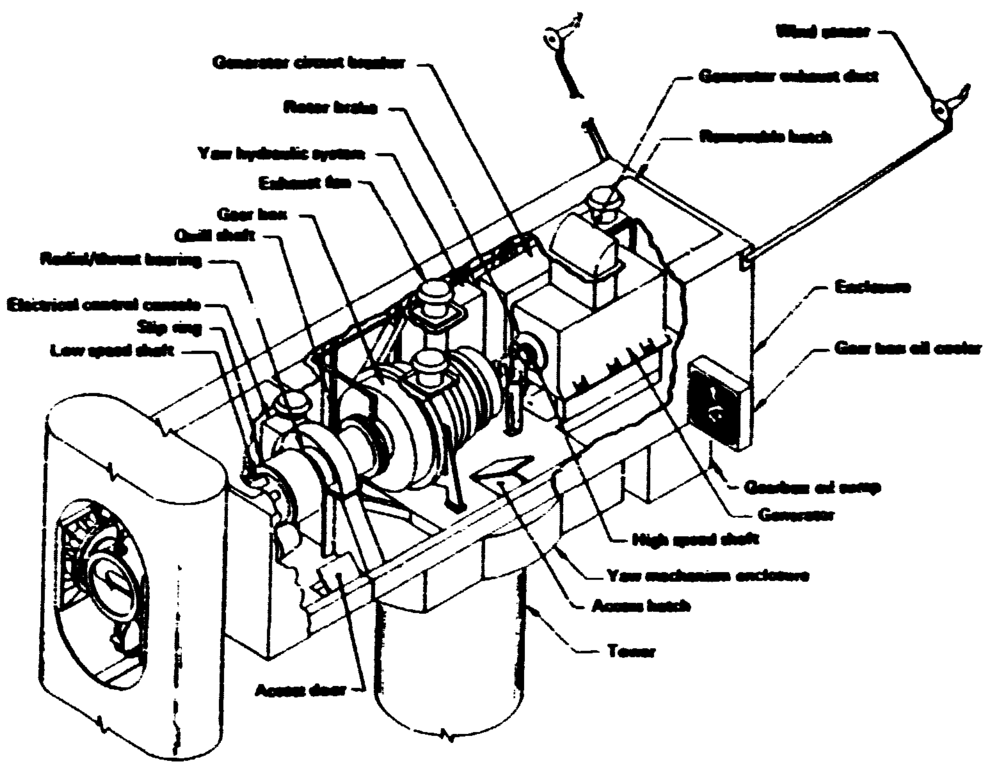


Figure 3-7. Nacelle Equipment Installation

Its primary functions are to provide a rigid mounting platform for the system components, react to rotor loads and provide environmental protection for the components.

The nacelle structure is of welded steel truss construction. Its primary dimensions are: 36.8 feet long, 9.3 feet high, and 11.3 feet wide. The top and sides are sheathed with corrugated steel sheets, and the floor is steel safety plate. A central floor hatch provides normal access from the tower, while hatches at either end provide emergency egress by means of a non-powered emergency man-lowering device. This general nacelle configuration was chosen as a result of the studies discussed in section 4.2.5.2 of this report.

Other primary considerations in the design of the nacelle were maintainability and safety. The MOD-2 design utilizes two overhead monorails for equipment handling. One of these extends through a large door in the downwind side of the nacelle. It can be used in conjunction with a portable hoist to raise equipment from the ground. There are also large overhead hatches for the installation or removal of large pieces of equipment. Care has been taken to allow sufficient room for maintenance procedures to occur within the nacelle.

Safety features of the nacelle are the integral fire extinguisher system, the emergency hatches and lowering devices, the ability to remove an injured person on a stretcher, and the positive mechanical shaft locking device. In addition, there is a nacelle intrusion alarm which will shut down the WTS should a person inadvertently enter the nacelle with the WTS operating.

The nacelle also houses the yaw drive system. The yaw system connects the nacelle to the tower as is shown in Figure 3-8. It rotates the rotor and nacelle into the wind at a rate of 1/4 degree per second, and holds them in position as commanded by the yaw control system. All rotor and nacelle loads are transferred to the tower through the yaw bearing which is of a crossed roller configuration with an internal ring gear. The raceway diameter is approximately 120 inches in order to handle the large overturning moments and to react to the rotor torque.

Proper nacelle orientation to the wind is maintained by the yaw control system. The control system utilizes a wind sensor to determine wind direction. To allow for the short period, wide directional variations common at low wind speeds, the yaw control system uses a 25.6 second average to determine wind direction. The control system then provides commands to the yaw drive system with the goal of maintaining orientation within $\pm$ 7 degrees of the wind direction to minimize energy loses. If the average deviation over an approximate 5 minute period exceeds $\pm$ 7 degrees, the control system will initiate yaw drive to align the nacelle with the wind vector. If the yaw error, averaged over a 2 minute period, exceeds 20 degrees, the machine is automatically shut down.

The yaw drive system operates at 2000 psi and consists of an electric motor, hydrualic pump, heat exchanger, reservoir, accumulators, filters, and the necessary valves and tubing. These drive a hydraulic motor which runs a pinion meshing with the gear on the inner race of the yaw bearing. The drive pinion and shaft are protected from overload and subsequent mechanical damage by a shear pin arrangement. The hydraulic drive system was chosen on the basis of the trade study discussed in section 4.2.5.1 of this report.

A hydraulic brake is applied to provide damping during yaw motion. The requirements for this brake were determined in the study described in section 4.2.5.3. An additional six brakes hold the nacelle from inadvertent yawing due to wind loads during "no yaw" operation. The yaw brake calipers are spring actuated and hydraulically released through the yaw drive hydraulic system. This is a failsafe feature assuring that the brakes are applied if there is a hydraulic failure. The brake disc has a replaceable element to minimize maintenance time if there should be excessive wear. A weather shield surrounding the yaw drive system provides both weather protection and a platform from which yaw brake and bearing maintenance can be performed.

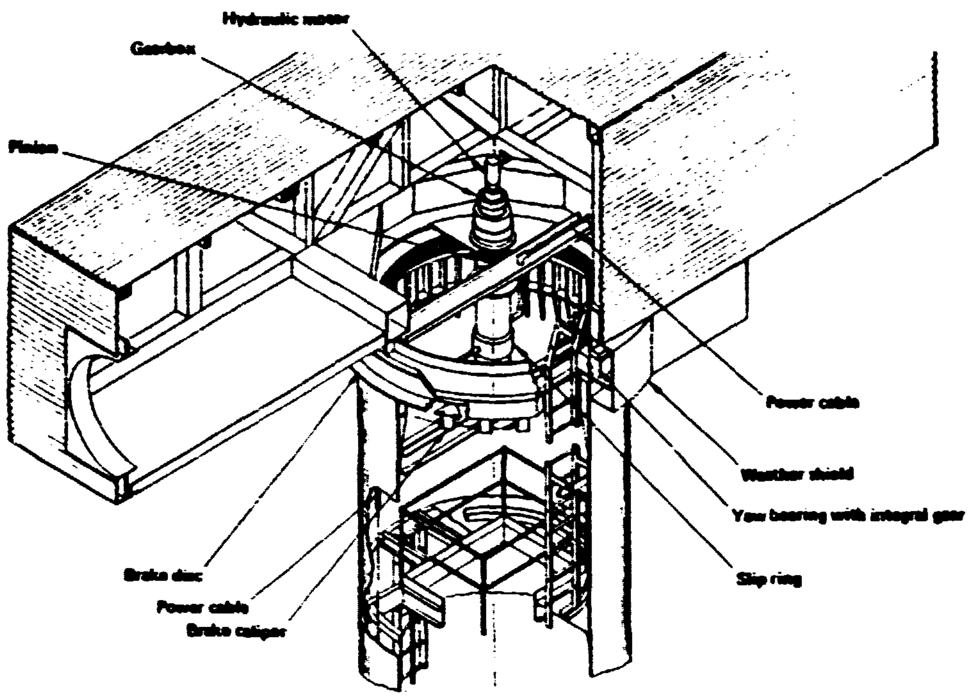


Figure 3-8. Yaw Drive Installation

3.2.4 Tower/Foundation and Facility Layout

The nacelle assembly is supported by a 193 foot tall cylindrical welded steel tower. The tower is 10 feet in diameter with a base section flaring to 21 feet in diameter at the ground. It is bolted to a foundation of reinforced concrete. In normal soil conditions, a buried octagonal stepped foundation configuration will be used as shown in Figure 3-9.

The tower is designed to have a low natural bending frequency (approximately 1.3 per rotor revolution) to reduce the alternating rotor loads transmitted to the tower. This tower concept was chosen after consideration of several other concepts as discussed in section 4.2.6 of this report, and as a result of vendor inputs concerning welded vs. bolted construction.

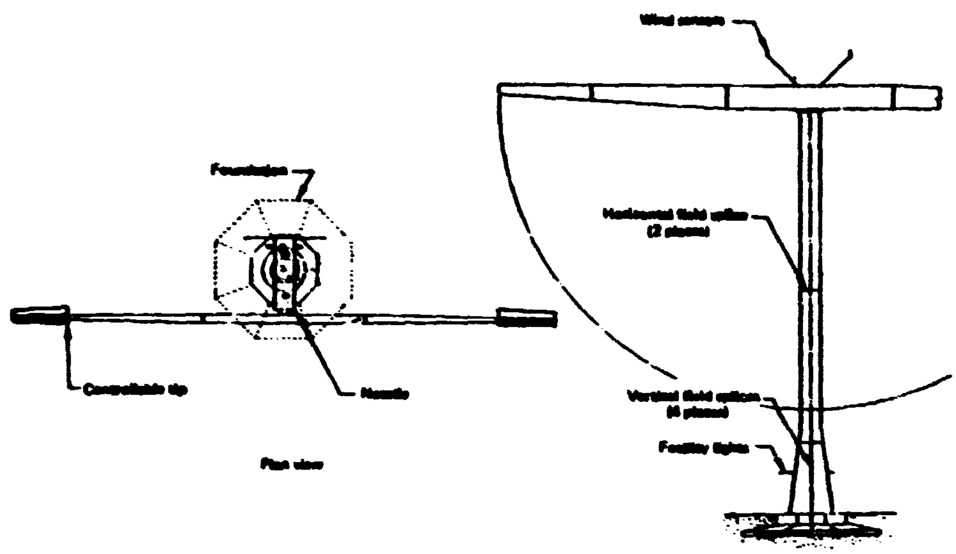


Figure 3-9. Tower/Foundation

The tower contains an internal lift to provide transportation from the ground to nacelle. The lift ends at a platform near the top of the tower, with final nacelle access by a ladder as shown in Figure 3-8. A ladder with safety rail runs the entire height of the tower to allow access in the event of a lift failure. The power cable runs from the electrical slip ring at the top of the tower, down the tower side to the bus tie contactor unit located on a separate concrete pad external to the tower (Figure 3-10). A step-up transformer is also located on this pad. Additional electrical and control equipment is located in the tower base. The power cable lines are buried from the tower to the utility connection at the transformer pad.

3.2.5 Electronic Control System

The control system provides the sensing, computation, and commands necessary for unattended operation of the WTS as shown in Figure 3-11.

The controller is a microprocessor which is located in the nacelle control unit and initiates start-up of the WTs when the wind speed is within prescribed limits. After start-up, it computes blade pitch and nacelle yaw commands to maximize the power output for varying wind conditions. Continuous monitoring of wind conditions, rpm, power and equipment status is also provided by the microprocessor which will shut down the WTS for out-of-tolerance conditions.

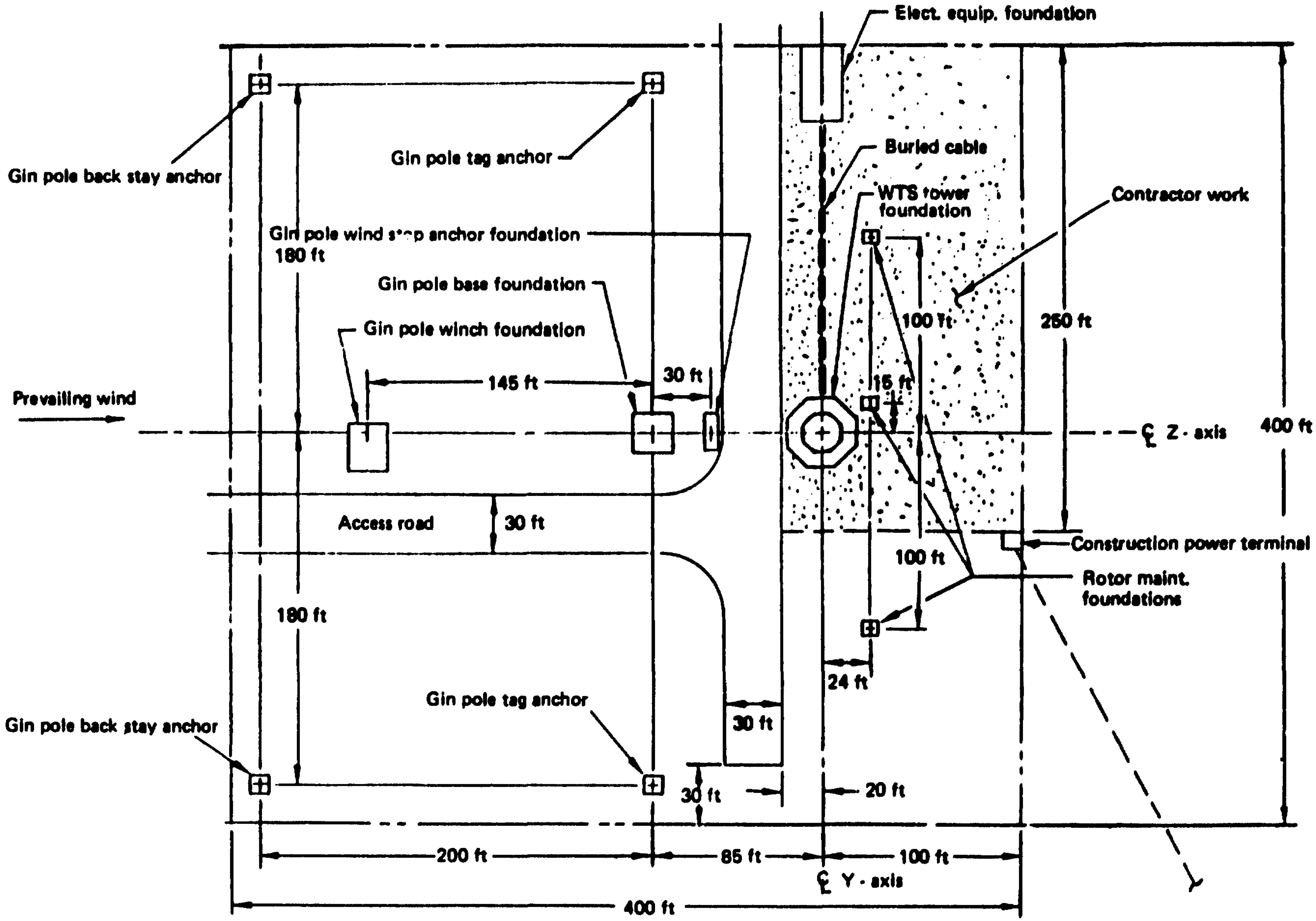


Figure 3-10. Facility Layout

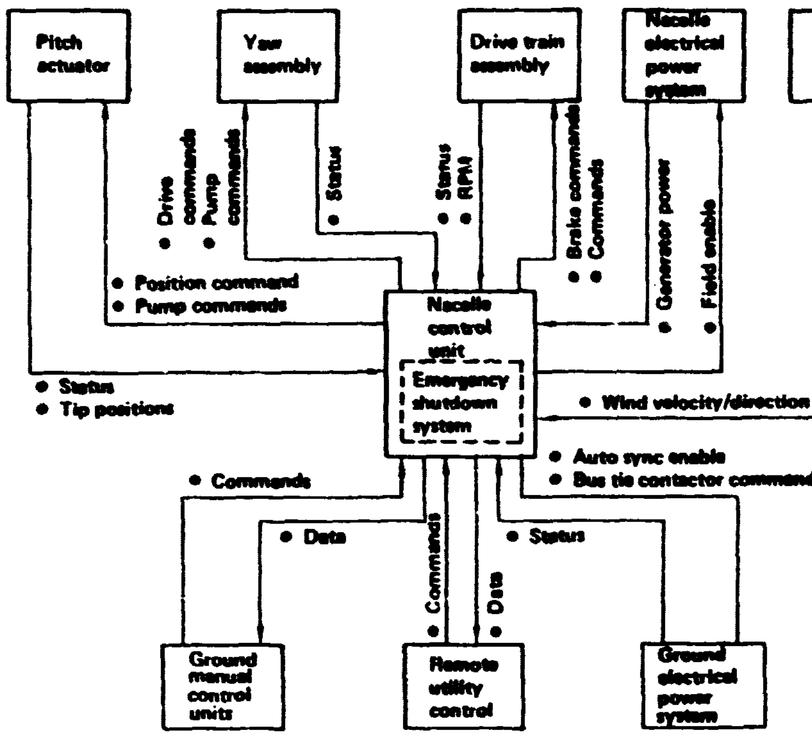


Figure 3-11. Control System Block Diagram

A control panel and CRT terminal are located in the tower base to provide operating and fault data displays and manual control for maintenance. A remote CRT terminal at the utility substation will provide display and limited WTS controls.

The WTS is protected from computer system failure by an independent failsafe shutdown system. The failsafe system also provides sensor redundancy on critical components, and initiates shutdown independent of the primary control system when necessary. The design of the failsafe system was governed by the results of the FMEA.

The control software for the microprocessor occupies approximately 12 thousand bytes of programmable read-only memory with an additional four thousand bytes of random access memory for operating and history data storage. The software control cycle is accomplished at a rate of 10 Hertz to provide a one Hertz response digital feedback control to the blade pitch system. In addition, each program cycle also samples all sensors, schedules the proper operating mode, and generates commands as required to control nacelle position with respect to the wind.

3.2.6 Electrical Power System

The WTS electrical power system is designed to deliver power to a utility transmission network. The system consists of the electrical equipment required for the generation, conditioning and distribution of electrical power to the utility and within the WTS as shown in Figure 3-12. In normal operation the generator receives its power in the form of torque at synchronous speed from the gearbox. Electrical power at appropriate voltage is delivered at a utility interface point which is the output side of a fused manual disconnect switch located at the foot of the tower. Once the WTS and the utility are electrically connected, the existence of the tie will automatically result in generator voltage and frequency control since the utility power grid is effectively an infinite bus to the WTS. Thus constant generator and rotor rpm will be maintained.

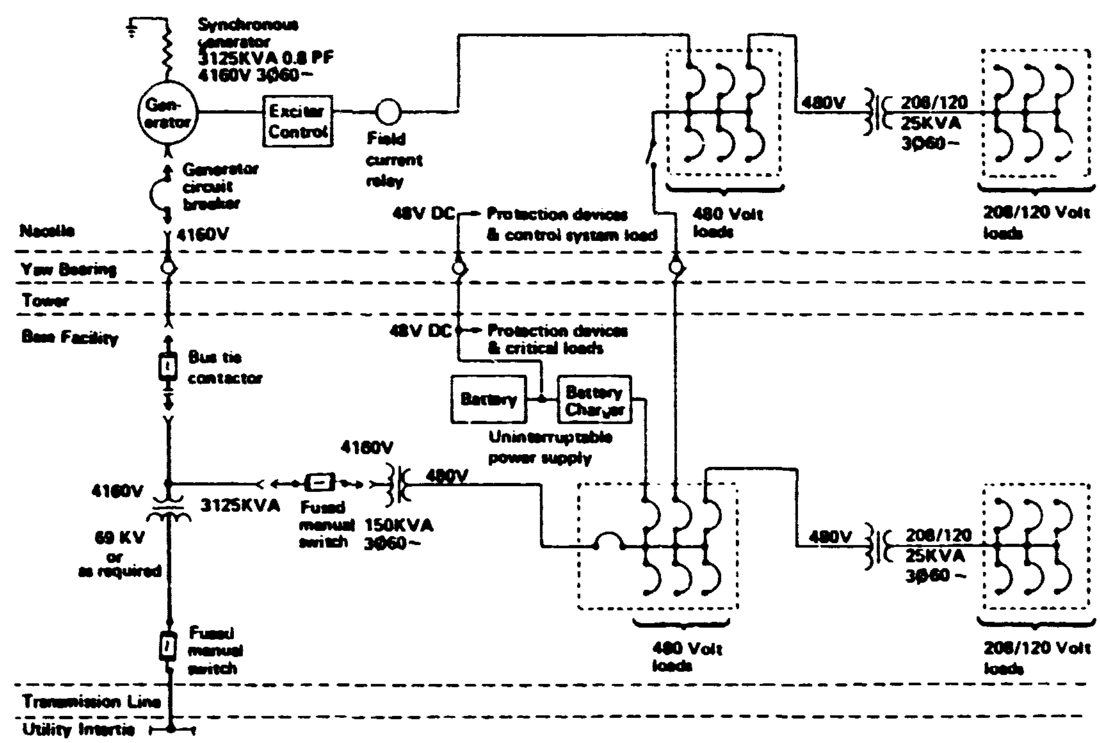


Figure 3-12. Electrical Power System Block Diagram

The MOD-2 electrical power system employs a four pole synchronous generator containing an integral brushless exciter. It is a 3 phase, 60 Hertz, 4160 v volt generator rated to provide 3125 KVA at 0.8 power factor, i.e. 2500 kW, at altitudes to 7,000 feet, or temperatures to 50 degrees C. This generator choice is based upon the results of the trade studies outlined in section 4.2.4 of this report, and the requirement for operation at altitudes to 7000 feet above sea level. Excitation control is provided to maintain proper voltage prior to synchronization with the utility and to provide a constant power factor output afterwards. Protective relays are provided to guard against potential electrical faults, out-of-tolerance performance, or equipment failures.

These relays will detect over-voltage, loss of excitation, underfrequency, overcurrent, reverse phase sequence, reverse power and differential current, and will protect the system by inhibiting synchronization, directing the control system to shut down or, if required, trip the generator circuit breaker. The operation of these protective relays was governed in part by the results of the FMEA.

Power is delivered to the utility transmission line through a bus tie contactor. Its operation is controlled by automatic synchronization equipment, located at the tower base. Accessory power for operation, control and maintenance is obtained from the utility or generator output depending on the operating mode, and is internally conditioned to appropriate voltage levels. A battery, floating across a charger, provides an uninterruptable power supply for operation of protective devices and critical loads. The use of a battery resulted from the trade studies in section 4.2.4. The results of the FMEA also contributed to the design of the uninterruptable power supply system.

3.3 SYSTEM PERFORMANCE

The performance of the MOD-2 WTS is specified by its system efficiency curve, its power and energy output distributions, and its annual energy output.

The efficiency of the MOD-2 WTS is described by a nondimensional number known as the power coefficient. Physically, the power coefficient is that fraction of the wind's kinetic energy passing through the rotor disk which is converted into electrical energy.

The system power coefficient for the MOD-2 WTS is shown in Figure 3-13. As indicated on the figure, the system power coefficient is derived from the rotor power coefficient and the efficiencies of the drive train and electrical subsystems. These subsystem efficiencies are discussed in more detail in section 5.3. Also indicated on Figure 3-13 is a rated power line for 2500 kW at sea level standard conditions.

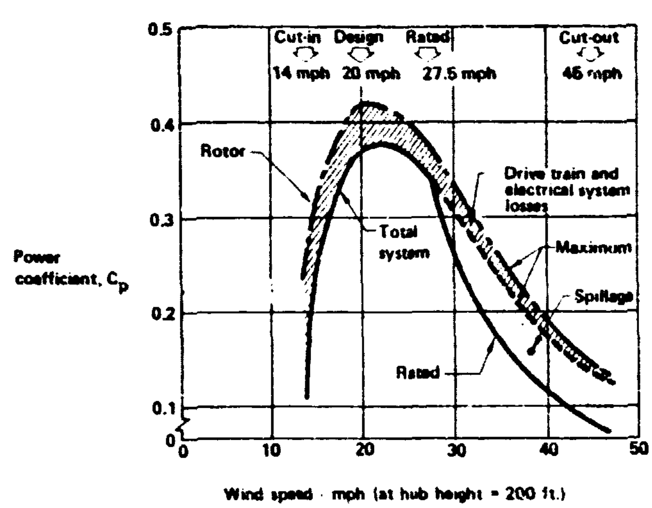


Figure 3 13. Cp Versus Wind Speed — Sea Level STD

The system efficiency curve can be translated into a power distribution curve when the atmospheric density is given. A derivation of this atmospheric density for specific site ambient conditions is discussed in section 4.4.2.

The MOD-2 system power output distribution is shown in Figure 3-14 for standard temperature at sea level and 7000 foot altitude. The cut-in, rated and cut-out wind speeds are also indicated on this figure. The rated wind speed is predicated upon use of a 2500 kW generator. The cut-in and cut-out wind speeds were selected in a trade study which is discussed in section 4.4.1.2 and 4.4.1.3.

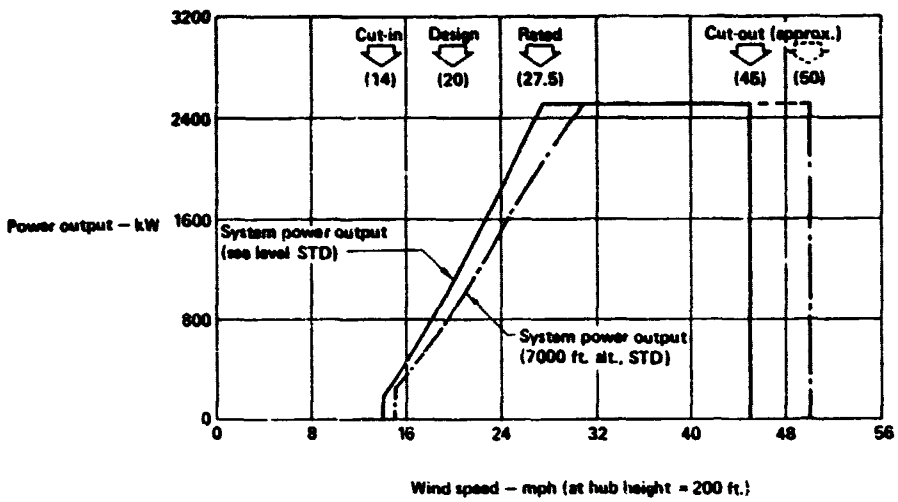


Figure 3-14. System Power Output Versus Wind Speed

The energy output frequency distribution for the MOD-2 WTS is derived by combining the output power distribution curve with the wind frequency distribution for a given site. The energy output frequency distribution for the MOD-2 WTS is shown in Figure 3-15 for sea level standard conditions. The associated wind frequency distribution is presented in Figure 3-16. The wind frequency distribution was specified by the NASA for MOD-2 design (see Appendix B, Specifications and Constraints). It represents a typical wind environment for potential wind turbine sites. The frequency distribution is typical of a site with a mean wind speed of 14 mph at an altitude of 30 feet and was used for optimization of the WTS characteristics. Following specific site selection, any new wind data for the site will be evaluated for possible impact on design or operating constraints.

The area under the curve is indicative of the time the WTS spends in different operating regimes. Approximately 59% of the time the WTS experiences winds between cut-in (14 mph) and rated (27.5 mph), while 18% of the time the wind speed is between rated and cut-out (45 mph). The remaining time (23%) the wind turbine experiences either winds too light or too strong for operation. Of this idle time, most occurs due to low wind speeds.

The annual energy output of the MOD-2 WTS is obtained by integrating the energy frequency distribution between the cut-in and cut-out wind speeds. For the MOD-2 WTS the total annual energy is 9,750,000 kWh, including the 0.967 system availability discussed in section 5.4.2. The MOD-2 WTS derives 61% of its energy when operating between the cut-in and rated wind speeds, while 39% of the annual energy is derived when operating above rated wind speed.

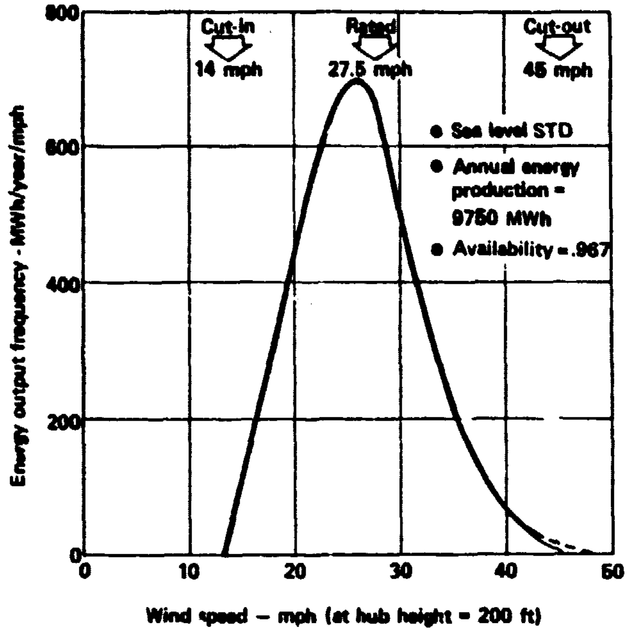


Figure 3-15. Energy Output Frequency Distribution

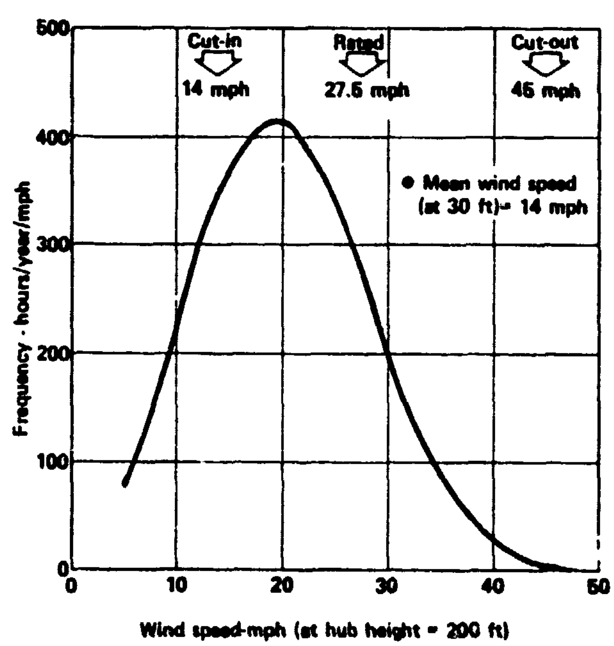


Figure 3-16. Wind Speed Frequency Distribution

3.4 WEIGHT

Table 3-1 presents the weight breakdown at the completion of the Task II period. The weights are estimates based on either preliminary design drawings, or vendor (catalog) weight estimates. Each drawing is broken into its separate components for which weights are estimated. This information is used as data for a computer program which determines the overall group weights as presented in the table. This weight information is used for both component design and cost estimation. As the cost of the WTS is closely related to its weight, any significant weight changes are carefully examined to insure they are required to meet the design requirements.

Table 3-1. Tier III Weight Report

Group	Element	Weight (lbs)
41	Blade	100,336
42	Hub	67,722
43	Pitch control	1,509
41-43	Rotor subassembly	169,567
51	Lo speed shaft + bearings	22,865
52	Quill shaft + coupling	9,483
53	Gearbox	39,000
54	Hi speed shaft + coupling	600
55	Rotor brake system	280
56	Lubrication system	6,664
57	Generator	17,000
51-57	Drive train	95,892
61	Nacelle structure	33,680
63	Yaw drive	17,742
64	Rotor support structure	7,152
65	Environmental Control + fire	870
66	Cabling + electrical facilities	645
67	Instruments + controls	690
68	Gen. accy. unit	2,500
61-68	Nacelle	63,279
71	Tower	246,536
72	Cable installation	4,130
73	Cable transition	500
74	Lighning protection	300
71-74	Tower subassembly	251,466
41-74	Total above foundation	580,204

3.5 DESIGN ENVIRONMENT

To insure the structural integrity of the MOD-2 WTS and its ability to operate in adverse environmental conditions, the design environment was defined as shown in Table 3-2. Most of the design environment was defined in Exhibit B (see Appendix B) of the contract. Some elements of the environment resulted from Boeing derived additions to Exhibit B. The environmental criteria considers both the geo-physical environment, and environments induced upon the WTS by each of its subsystems (i.e. vibration, electromagnetic, etc). The gust criteria contained within the wind criteria was developed by Boeing and NASA to allow calculation of the gusts effects on fatigue life. Previous gust models had been discrete which did not allow determining their effect on fatigue life.

Table 3-2. WTS Design Environment

	Transportation	Storage Installation	Operational
Duration	3 weeks	3 months	30 years (2104 hr/yr standby) (6662 hr/yr pwr opn)
Wind	Exhibit B	Exhibit B	Exhibit B
Shock & vibration	Exhibit B	Exhibit B	Exhibit B
Temperature	Same as operational	Same as operational	-40°F (-40°C) to 120°F (48.9°C) ambient air
Solar radiation	Same as operational	Same as operational	363 BTU/ft^2/hr, 4 hrs daily 6 months annually
Lightning	Negligible	Negligible	Exhibit B
Rain	Same as operational	Same as operational	4 inches/hour
Hail	1.0 inch dia. 50 lb/cu. ft. 132 ft/sec. vel. (applicable to shipping containers)	1.0 inch dia., 50 lb/cu. ft. 66.6 ft/sec. vel. (applicable to storage containers)	1.0 inch dia., 50 lb/cu. ft. 66.6 ft/sec. vel. (horiz. & vertical surfaces)
Ice (glaze)	2.0 inches thickness 60 lbs/cu. ft. (applicable to shipping containers)	2.0 inches thickness 60 lbs/cu. ft. (applicable to storage containers)	2.0 inches thickness 60 lbs/cu. ft. (on all external surfaces)
Snow	41 lbs/sq. ft. (shipping containers)	41 lbs/sq.ft. (storage containers)	Blade: 21 lbs/sq. ft. Nacelle: 41 lbs/sq. ft.
Humidity Sand/Dust Salt Spray Fungus	Same as operational	Same as operational	Exposure equivalent to MIL-STD-210B for exposed or sheltered ground equipment, as applicable
Fauna	Exposure to insects	Same as transportation	Same as transportation plus 4 lb. birds at 35 mph for stationary surfaces above 150 ft.
Noise	Negligible	Negligible	Negligible
Seismic	Exhibit B	Exhibit B	Exhibit B
Altitude	Same as operational	Same as operational	Sea level to 7000 ft.

3.6 COST

Estimated 100th production unit costs for the MOD-2 WTS are summarized in Figure 3-17. The turnkey estimates include all costs associated with the manufacture, assembly and installation of the WTS. The manufacturing costs are based upon a dedicated high rate production facility as discussed in section 5.5.3. The spares and operations and maintenance cost ground rules are presented in section 5.5.4.

Turnkey account		Cost
1.0	Site preparation	$162,000
2.0	Transportation	29,000
3.0	Erection	137,000
4.0	Rotor	329,000
5.0	Drive train	379,000
6.0	Nacelle	184,000
7.0	Tower	271,000
8.0	Initial spares	35,000
8.A	Non-recurring	35,000
9.0	Total initial cost	$1,561,000
	Fee (10%)	156,000
	Total turnkey	$1,717,000
10.0	Annual operations and maintenance	$15,000

The cost estimating ground rules are as follows:

- All costs are in mid 1977 dollars
- Costs are fully burdened and a 10% fee is included
- Costs of installation and operation are based on a 25 unit farm
- Rate of installation is one WTS per month per farm
- Sites are generally flat with few natural obstacles and soil is easily prepared for foundation (land cost not included)
- Transportation costs are based on rail and truck transport over a distance of 1,000 miles

Figure 3-17. 100th Production Unit Costs

The 25 unit farm has a total rated power of 62.5 MW. Each unit contains a step-up transformer that increases the generator output voltage to 69 kV for transmission. Costs beyond the step-up transformer are not included in the turnkey costs.

3.6.1 Cost of Electricity

The primary objective in the development of MOD-2 WTS is to provide a WTS that produces energy at a cost of electricity under 4¢/kWh based on 1977 cost forecasts. The total cost of the MOD-2 WTS of $1,720,000 together with the O & M costs of $15,000 and an annual energy output of 9.75×10^6 kWh results in a projected cost of electricity of 3.3¢/kWh. This is based upon a levelized fixed charge rate approach as presented in section 5.5.5.

4.0 SYSTEM DEVELOPMENT

This section documents: (1) the initial MOD-2 requirements, (2) the approach used to optimize the WTS design, and (3) the work accomplished to evolve the MOD-2 configuration described in section 3.0.

The general approach employed to optimize and define the configuration is shown in Figure 4-1. The process started with a set of requirements specified by NASA (Exhibit B of the contract). A baseline configuration was defined together with its cost, performance and weight. A production scenario was established and a farm of 25 MOD-2 units was defined as a basis for the cost analyses. Site ground rules and a maintenance and operations scenario was established.

Design trade studies were identified based upon their criticality to the system's performance and cost. Applicable subsystem or systems designs were prepared for these selected studies and appropriate structural analyses were conducted. Design iterations were performed as a result of the structural analyses and the final configurations were then analyzed for weight, performance and cost and then compared to the baseline or other trade configurations. The final configuration was selected on the basis of the lowest cost of electricity, personnel safety and program risk.

Other related efforts that had a major influence on the final design described in section 3.0 were the sensitivity analyses applied to the specifications (4.4), the failure mode and effects analyses (4.5), the inputs from utility company consultants (4.6), and the various analytic, test and operational data obtained from NASA/DOE.

4.1 OBJECTIVES AND REQUIREMENTS

All design and trade studies have been based on the specifications and constraints as given in Exhibit B of the contract (see Appendix B). The majority of these specifications together with additional major criteria developed coincident with the studies are summarized in Table 4-1. Examples of the additional or refined criteria developed are the new wind profile expressions and the gust criteria discussed further in sections 5.1.5. All criteria developed was thoroughly explored for the effect on cost, safety and programmatic risk.

4.2 TRADE STUDIES

Engineering effort during the concept design phase of the program was directed primarily toward conducting and evaluating numerous trade studies. The objective of this effort was to optimize system performance to obtain least cost of electricity prior to proceeding with the detail development and design phases of the program. As a result of these studies, several basic concept changes were made to the original proposal configuration, such as: change to an all steel welded rotor; change to an upwind rotor; change to a teetered rotor; and change to partial span pitch control system.

In the presentation of these studies, the objective has been to adopt a standard format and to compare to an initial baseline configuration whenever possible. As the concept design phase progressed, the trade studies incorporporated changes to the baseline. Some of the studies have been carried to the level which

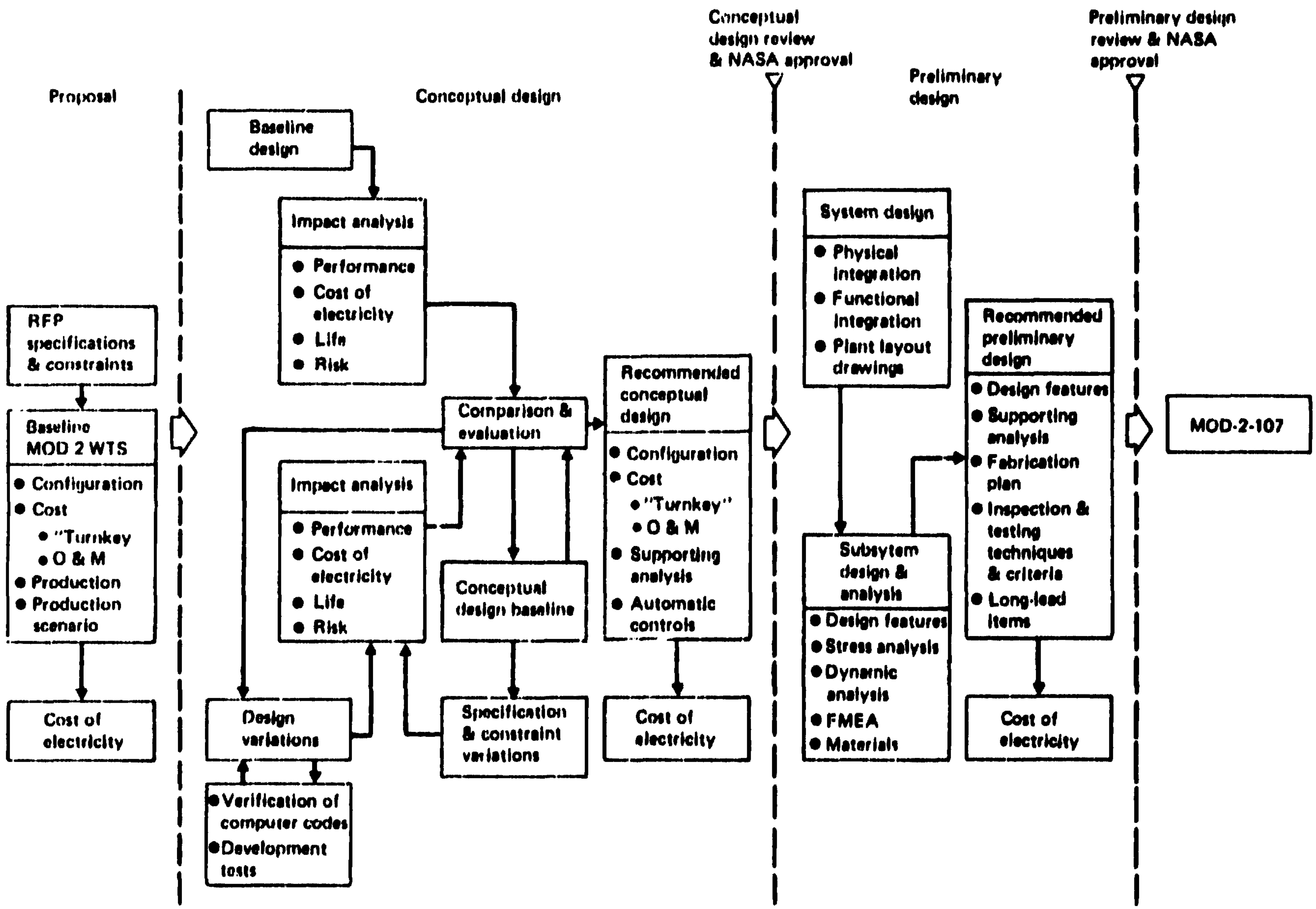


Figure 4-1. MOD-2 Wind Turbine Design Approach

Table 4-1. MOD-2 Objectives and Requirements

Requirement/Objective	Value
Design Requirements	
Service Life	30 years
Power Output (3 phase,60 HZ)	Mega-watt range
Rotor Orientation	Horizontal Axis
Weight and Dimensional (Transport)	Yes
Finish Duration	Yes
Cut-in/Rated/Cut-out Wind Speed	14/27.5/45 mph
Color and Identification Markings	Yes
Rotor Diameter	≥ 300 feet
Environmental	
Mean Yearly Wind Speed at Site	14 mph at 30 ft
Wind Gradient	Variable Power Law
Wind Speed Duration	Weibull
Gust Criteria	Yes
Altitude	0-7000 ft
Lightning Model	Yes
Seismic	Yes
Temperature range	-40° F. to 120° F.
Other (rain, hail, snow, etc)	Yes
Max. Design Wind	120 mph at 30 ft.
Safety	
Fail Safe (Unattended)	Yes
Fire Detection	Yes
Site Security System	Yes
Hazard Protection	Yes
Network and Turbine Protection	Yes
Self Protection in Emergency	Yes
Obstruction Marking and Lighting	Yes
Operations & Maintenance	
Tools, Vehicles	Commercial
Automatic/Manual Operation	Yes
Availability	.90 Minimum
Remote/unattended Control	Yes
Data Systems Channels	≤100
Maintenance Concept	Yes
Cost	
Cost of Electricity	Under 4¢/kWh
Units in Farm	25
Production Rate	Yes
Fixed Charge Rate	.18
Cost of Elect. Equation Specified	Yes
Site Definition	Yes
Transport Distance	1000 miles

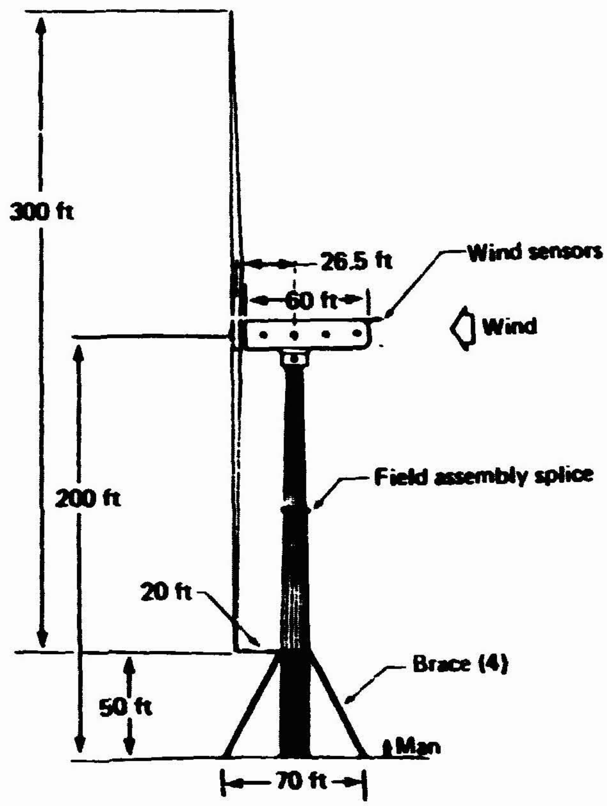
presents the effect of a subsystem change on the total WTS configuration and on the resulting C.O.E. Some studies, where the results showed a conclusive trend of optimization, were evaluated only to the subsystem level. For all studies, the results are presented in terms that are most appropriate to the particular study. Each study, if not compared to the initial baseline described below, had a consistent set of basic criteria and therefore the results and conclusions are valid. The trade studies described in this section (4.2) are listed in Table 4-2.

Table 4-2. Trade Study Summary

SUBJECT	REFERENCE PARAGRAPH
Two vs Three Blade Rotor	4.2.2.1
Teetered vs Rigid Rotor	4.2.2.2
Optimum Rotor Speed	4.2.2.3
Partial vs Full Span Rotor Control	4.2.2.4
Aluminum vs Steel Tip	4.2.2.5
Airfoil Geometry	4.2.2.6
Blade Material and Configuration Studies	4.2.2.7
Crack Detection System	4.2.2.8
Crack Stopping Design	4.2.2.9
Metal vs Composite Rotor	4.2.2.10
Upwind vs Downwind Rotor	4.2.2.11
Tip to Ground Clearance	4.2.2.12
Tilted vs Non-tilted Rotor	4.2.2.13
Epicyclic vs Parallel Shaft Gearbox	4.2.3.1
Shaft Configuration Studies	4.2.3.2
Generator Selection	4.2.4.1
Selection of the Diesel Driven Generator	4.2.4.2
Location of the Generator Circuit Breaker	4.2.4.3
Wiring Transfer Across the Yaw Bearing	4.2.4.4
Electric vs Hydraulic Yaw Drive	4.2.5.1
Nacelle Configuration Studies	4.2.5.2
Yaw Stiffness Requirements	4.2.5.3
Soft vs Stiff Tower	4.2.6.1
Soft vs Soft-soft Tower	4.2.6.2
Braced vs Conical Base	4.2.6.3
Transition Section	4.2.6.4
Machine Size Optimization	4.2.7
Analog vs Microprocessor	4.2.8.1
Multiplexer, Ground Computer vs Nacelle Microprocessor	4.2.8.2

4.2.1 Trade Study Baseline

In general, the initial baseline for the trade studies was the original proposal configuration, updated to reflect more accurate cost and weight data. Figures 4-2 and Tables 4-3, 4-4 and 4-5 illustrate this baseline configuration, features, weights, and costs.



Rated power	2500 kW
Rated wind at hub	28 mph
Mean wind at 30 ft	14 mph
Rotor tip speed	275 FPS
Rotor rpm	17.5
Generator rpm	1800

Figure 4-2. Trade Study Baseline Configuration

Table 4-3. Trade Study Baseline Features and Characteristics

RATED POWER	2500 KW
ROTOR ORIENTATION	DOWNWIND
ROTOR DIAMETER	300 FEET
ROTOR BLADE MATERIAL	STEEL/PAPER HONEYCOMB
RATED WIND @ HUB	28 MPH
MEAN WIND @ 30 FEET	14 MPH
DESIGN WIND @ HUB	20 MPH
ROTOR TIP SPEED	275 FT/SEC
ROTOR RPM	17.5
GENERATOR RPM	1800
GENERATOR TYPE	SYNCHRONOUS
GEAR BOX STEPUP RATIO	102.8
GEAR BOX TYPE	3-STAGE PLANETARY
HUB HEIGHT	200 FEET
TOWER	STEEL (SHELL) TYPE
CAPACITY FACTOR	0.37
DESIGN C_L	1.0
DESIGN TIP SPEED RATIO	9.4
MAX. SYSTEM POWER COEF.	.38

Table 4-4. Trade Study Baseline Weights

ROTOR	177,000	
BLADE (2)		104,000
HUB		53,000
HINGE (4)		20,000
DRIVE TRAIN	111,000	
GEAR BOX & SUPPORT		70,000
GENERATOR		16,000
MECHANISM & SHAFTS		25,000
TURN TABLE	33,000	
NACELLE STRUCTURE & ENCLOSURE	35,000	
TOWER	300,000	
	656,000#	

Table 4-5. Trade Study Baseline Production Costs (100th Unit)

	ELEMENT	(10^3\$)
1.	SITE PREPARATION	153
2.	TRANSPORTATION (1000 mi)	40
3.	ERECTION & CHECKOUT	197
4.	ROTOR SUBASSEMBLY	268
5.	DRIVE TRAIN	462
6.	NACELLE SUBASSEMBLY	216
7.	TOWER SUBASSEMBLY	201
8.	INITIAL SPARE PARTS	20
9.	TOTAL INITIAL COST	1,557
10.	ANNUAL OPERATION & MAINTENANCE COST	20

Where other than the above configuration was used as a comparison baseline, the actual baseline used is appropriately described as part of the following individual study descriptions.

4.2.2 Rotor

This section reports the results of trade studies conducted to evaluate alternative rotor configurations and characteristics.

4.2.2.1 Two V.S. Three Blade Rotor

The MOD-2 two blade rotor configuration is described in Section 3.2.1. This trade study provided the basis for selection of the number of blades.

OBJECTIVE: The objective of this study was to determine whether the increased energy captured by a 3-blade rotor due to its higher rotor efficiency was offset by the added cost of the third blade and a more complex hub.

The blades were assumed to be of identical planform for both 2-blade and 3-blade rotors, with plate gages and weights adjusted to agree with the respective loads. The hub was reconfigured for the 3-blade rotor and a splice added for shipment considerations. Both hubs were of the fixed or rigid type.

The yaw drive, braking systems, and drive train were resized in accordance with the loads and rotor rpm. Other WTS components were assumed to be unaffected.

RESULTS: It was determined that the increased energy output of the 3-blade rotor is more than offset by the increased system cost, such that the cost of energy is higher for the 3-blade system.

The net system cost increase resulted primarily from a 3-blade rotor being more complex and the requirement for a higher ratio gearbox.

The essential parameters effecting the 2 or 3 blade rotor trade study are shown in Table 4-6.

Table 4-6. Essential Parameters Affecting Rotor Weight

	2 Blade	3 Blade
Rotor Diameter, ft.	300	300
Generator Rating, kW	2500	2500
Solidity	.030	.045
Rotor rpm	17.6	14.4
Tip speed, f.p.s.	275	225
Bending Moment, ft. lb. X 10^{-5} (ϕ = -20^o, R/r = .156)		
Flapwise Steady	-13.78	-9.66
Alternating	6.77	5.6
Chordwise Steady	-.997	-.408
Alternating	17.29	16.6
Resulting Weight, lb.		
Single Blade	52,000	48,000
Total Rotor	177,000	262,000

The increased performance of the 3-blade configuration as shown in Figure 4-3 is 2.5%.

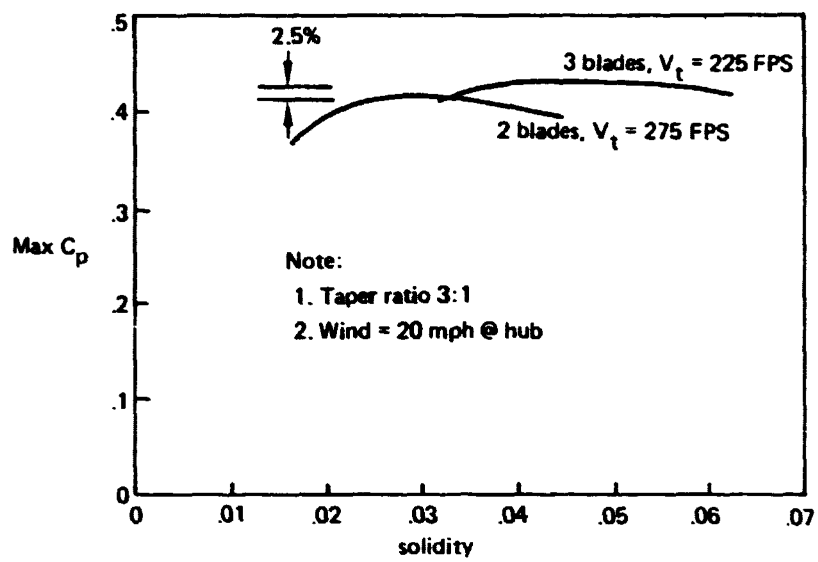


Figure 4-3. Blade Number Performance Effect

SUMMARY: Table 4-7 summarizes this study.

Table 4-7. Summary - Effect of Number of Blades

Criteria	Configuration		Remarks
	2 blade	3 blade	
Rotor diameter, ft	300	300	50 ft ground clearance used for both cases
System C_p max	.380	.396	
Yaw system cost, $	85,000	72,000	3 blade system requires smaller drive & brakes
Rotor cost, $	268,000	371,000	3 blade hub requires splices for shipping
Drive train cost, $	462,000	538,000	3 blade configuration requires larger gearbox
System cost - 100th unit, $	1,557,000	1,723,000	
Operations & maintenance cost per year, $	20,000	22,000	
Energy out, kwh per year	8,686,000	8,890,000	Factored to account for downtime
Cost of electricity, ¢/kwh	–	Adds .28	

CONCLUSIONS: The increased energy output of a 3-blade rotor is more than offset by increased costs for a wind turbine of size and capacity similar to the MOD-2.

4.2.2.2 Teetered VS Rigid Rotor

The term "teetering" refers to the addition of one degree of freedom to rotor restraint, such that the rotor is free to teeter in and out of the disk plane. Flapwise alternating bending moments from gust and other transient loads are reduced by this method. Three studies were undertaken which dealt with the teetering concept.

OBJECTIVE: The objective of the first study was to compare the cost of electricity of a WTS which incorporated the teetering method of rotor restraint to one rigidly restrained. The second study compared various methods of stopping the rotor at the limits of the teeter angle. The third study compared roller, plain, and elastomeric bearings for the teeter hinge.

RESULTS: Loads-Significant percentage reductions in system loads, particularly alternating loads, are experienced with teetering as shown in Figure 4-4.

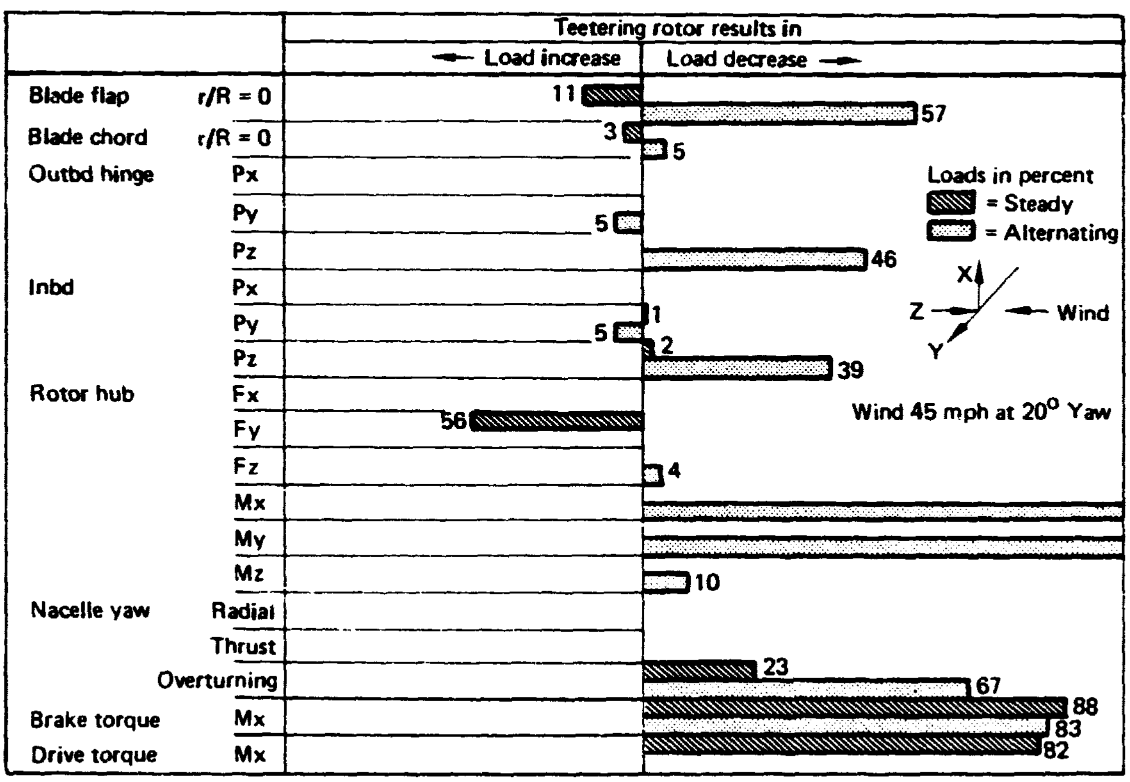


Figure 4-4. Load Change Summary

Tower Stiffness - Tower wall thickness reductions are possible because the stiffness designed tower reacts the teetered rotor as though the rotor weight were lumped at the center of rotation. Reductions are shown in Figure 4-5.

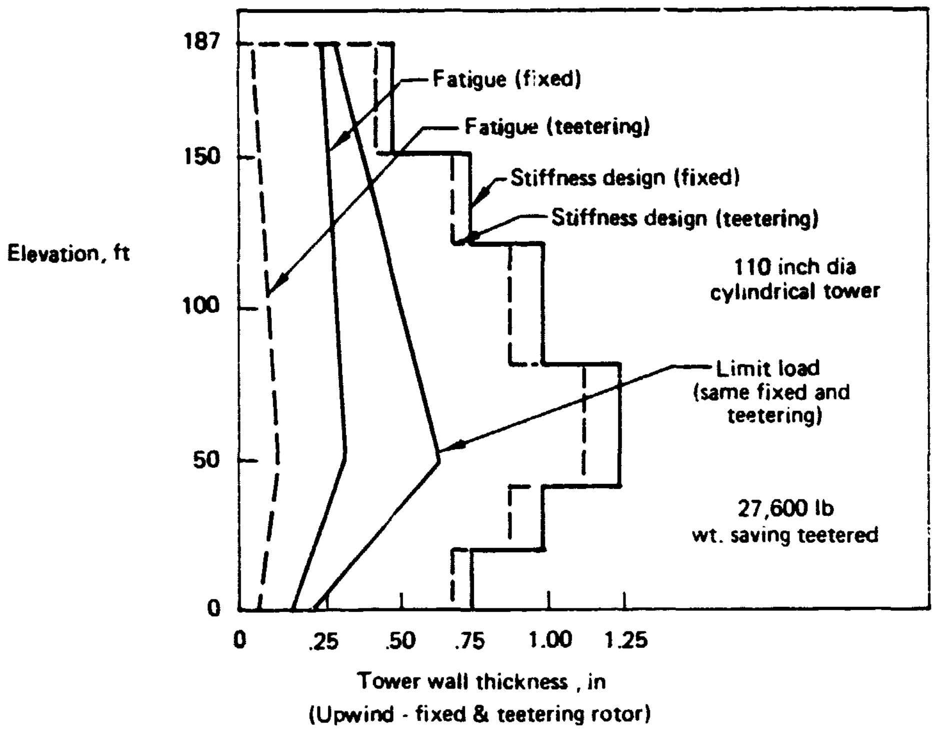


Figure 4-5 . Tower Design

Weight - Significant reductions are possible in the fatigue designed areas of the rotor, the stiffness designed tower, and the yaw drive as shown in Table 4-8.

Table 4-8. Weight Comparison (Teetering Versus Fixed)

ITEM	Teetering Results Increased Weight, lb.	Decreased Weight, lb.
Blades (2)		10,000
Hub		9,068
Rotor Shaft	2,926	
Control System	1,513	
Nacelle		7,960
Yaw System		11,027
Tower		27,600
Total	+4,439	-65,655

Total weight reduction = 61,216 lb.

Table 4-9 summarizes teetering vs fixed hub rotor study.

Table 4-9. Trade Study Summary - Teetering Versus Fixed Hub Rotor

	Fixed rotor	Teetering rotor	Remarks
Design complexity	Baseline	More complex	Well within state-of-the-art
Erection	Similar		
Transportation	Baseline	$1900 reduction	Per thousand miles
Weight	Baseline	61,000 lb lighter	Due to reduced loads
Annual energy - kwh	Same		
Hardware cost	Baseline	$66,800 reduction	
Reliability	Baseline	10% reduction MTBF	For blades & hub only
Maintainability	$20,000	$20,200	Per year
Safety	Similar		
Technical risk	Baseline	Lower	Gusts & alternating loads have less effect on structure & yaw system
Cost of electricity	Baseline	.14/kw-hr reduction	

CONCLUSION: Based on the reduced cost of energy the teetered rotor concept offers the best design for a large wind turbine system.

TEETER STOP STUDY RESULTS: Various methods were investigated to minimize the loads induced by teeter stop contact. The methods included hard or high spring rate bumper stops, 4 configurations of springs, hydraulic dampers, and friction brakes. Friction brakes combined with bumper stops at the travel limits displayed the least technical and design impact on the rotor. Table 4-10 summarizes this study.

Table 4-10. Teeter Stop Study Summary

	Rigid	Spring	Damper	Brake	Remarks
Design complexity	Least complex	Average	Average	Average	
Weight (lbs)	400	600 - 3,600	750	900	
Damping	None	Little	Viscous	Friction	
Cost $	3,200	4,500-8,800	4,700	4,600	
Teeter lock for maintenance	Tool required			Provided by brake	
Maintainability	Least maintenance	Low maintenance	Some maint	Some maintenance	
Technical risk	Highest shock load	Causes rebound	Potential leakage	Lowest risk	Brake provides design flexibility
Rotor design impact	Highest shock load	Some impact	Some impact	Low impact	
Rotor excursions (low rpm or parked)	Travels stop-to-stop			Resists teetering	

CONCLUSION: The friction brake is the best concept for minimizing teeter stopping loads due to lower loads and comparable cost.

TEETER BEARING STUDY RESULTS

Roller, plain, and elastomeric bearings were compared for use in connecting the teetering rotor to the drive shaft. Twenty four bearing characteristics were compared as shown in Table 4-11.

Table 4-11. Teeter Bearings Comparison

Characteristic	Roller	Plain	Elastomeric
1. Requires lubrication	Yes	Yes	No
2. Lube distribution problem	Yes	Yes	No
3. Requires lube seal	Yes	Yes	No
4. Requires environmental seal	Yes	Yes	Sun & oil shield
5. Ages in sunshine	Seal ages	Seal ages	Yes
6. Temperature extremes	Lube affected	Lube affected	Stiff when cold
7. Compression set	No	No	Yes
8. Fretting corrosion	Yes	Dep on lube dist	No
9. Brinnelling	Yes	No	No
10. Failure detection	Noise	Measure clearance	Visual crumbs
11. Shape and finish of mating parts critical for inst'l	Yes	Yes	No
12. Alignment critical for even loading	Yes	No	No
13. Inner face rotation	No	Yes	No
14. Outer race rotation	No	No	No
15. Interface fabrication risks (rework or scrap mating parts)	High	Medium	Low
16. Requires preloading	Yes	No	No
17. Axial motions detrimental to bearing	Yes	No	No
18. Breakout torque	No	Yes	No
19. Centering spring rate	No	Yes	Yes
20. Field replacement difficulty	Most	Medium	Least
21. Operational experience	Yes	Yes	Yes
22. Distribution of bearing material	360°	360°	Where required
23. Pricing per set (bearings only)	\$8,000	\$17,600	\$8,000 - 11,000
24. Cleanliness	Good	Good	Bad

The teeter bearing study is summarized in Table 4-12.

Figure 4-12. Teeter Bearings Study Summary

	Roller	Plain	Elastomeric	Remarks
Design complexity	Critical fits & alignment	Most parts	Least complex	Roller & plain bearings require lube systems and seals
Erection (installation)	Most difficult	Most parts	Least difficult	Roller brng requires interference fit & axial preload
Cost $ (brngs only)	8000	17,600	8000 - 11,000	
Life	Limited by lube problems	Limited by lube problems	Best life capacity	
Maintainability	Most difficult	Difficult	Least difficult	Elastomeric requires no lube, easiest to inspect, easiest to replace
Technical risk	Known problems	Known problems	Heliocopter experience	

CONCLUSION: The elastomeric bearing is the best bearing for the teeter hinge due to less complexity and maintenance costs, ease of erection, and long life.

4.2.2.3 Optimum Rotor Speed

The MOD-2 rotor speed is 17.5 rpm. This trade study provided the basis for this speed selection.

OBJECTIVE: The objective and purpose of this study was to determine whether the cost of electricity could be reduced if the WTS operated at higher speeds. Although the available annual energy was less for operating at higher speeds (see Figure 4-6) it was felt that a possible reduced cost of electricity could result because of the lower drive torque. The three rotor speeds considered were 17.5, 19, and 20.5 rpm.

RESULTS: The results of this study are described below:

Rotor - The weight of the rotor was reduced (-2,100 lbs. and -4,400 lbs. for the 19 and 20.5 rpm speeds respectively). This was caused by the beneficial effects of the blade centrifugal forces reducing the compression buckling loads.

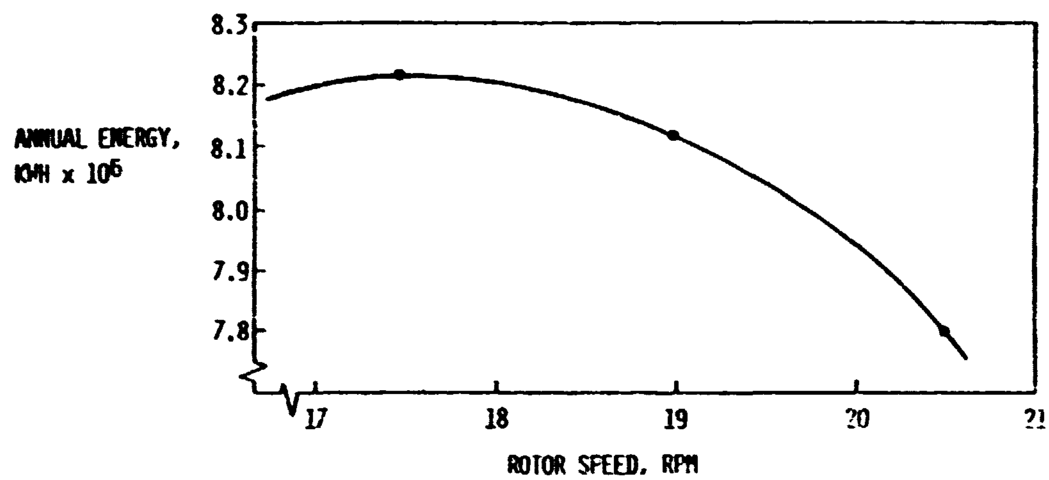


Figure 4-6 Annual Energy Versus Rotor Speed

Drive Train - The low speed shaft, quill shaft and gear box were lighter (8,400 lbs. for 19 rpm and 14,000 lbs. for 20.5 rpm). The lighter weights were due primarily to the reduced torque at the higher speeds and a stiffer quill shaft. No weight change was identified for the generator.

Nacelle - The lighter rotor, low speed shaft, quill shaft and gear box produced smaller loads into the nacelle structure. Additionally, the shorter quill shaft allowed the length of the nacelle to be reduced. These caused the weight of the nacelle to be reduced by 6,000 pounds for the 19 rpm speed and by 10,000 pounds for the 20.5 rpm speed.

Tower - The higher speeds required a stiffer tower to be designed. The tower weight increased by 24,500 and 44,900 pounds for the 19 and 20.5 rpm rotor speeds respectively.

Foundation - The foundation volume was reduced by 51 cubic yards for each of the higher speeds.

Energy output - The available annual energy was 1.1% less for the 19 rpm and 5.0% less for the 20.5 rpm rotor speeds. (See Figure 4-6)

Costs - The costs of the wind turbine systems were reduced by $20,400 and $34,100 for the 19 rpm and 20.5 rpm speeds respectively. This cost reduction was not enough to offset the reduced wind energy available for operating at the higher speeds. Combining the energy available and the cost reductions, no change in COE resulted for the 19 rpm speed and a COE increase of .12¢/kWh for the 20.5 rpm.

STUDY SUMMARY:

	19 RPM	20.5 RPM
WEIGHT, LBS.	+ 8,000	+16,400
COST, $	-20,400	-34,100
ANNUAL ENERGY, %	- 1.1	- 5.0
COST OF ELECTRICITY, ¢/kWH	± 0	+.12

CONCLUSION: No change in rotor rpm was recommended for the MOD-2 because there was no cost advantage offered by higher rpm.

4.2.2.4 Partial vs Full Span Rotor Control

The tip control trade study was conducted to evaluate a rotor which is fixed in pitch over most of its length. The outer tip only is capable of pitching for startup, shutdown, speed control, and gust attenuation.

OBJECTIVE: The objective of this trade study was to compare the cost of energy of a system with full blade span pitch control with one incorporating partial span pitch control. Two data points, 20% of span and 40% of span were selected for study.

RESULTS: Design Requirements - The ability to startup in low wind speed using partial span blades were compared to full span startup as shown in Figure 4-7.

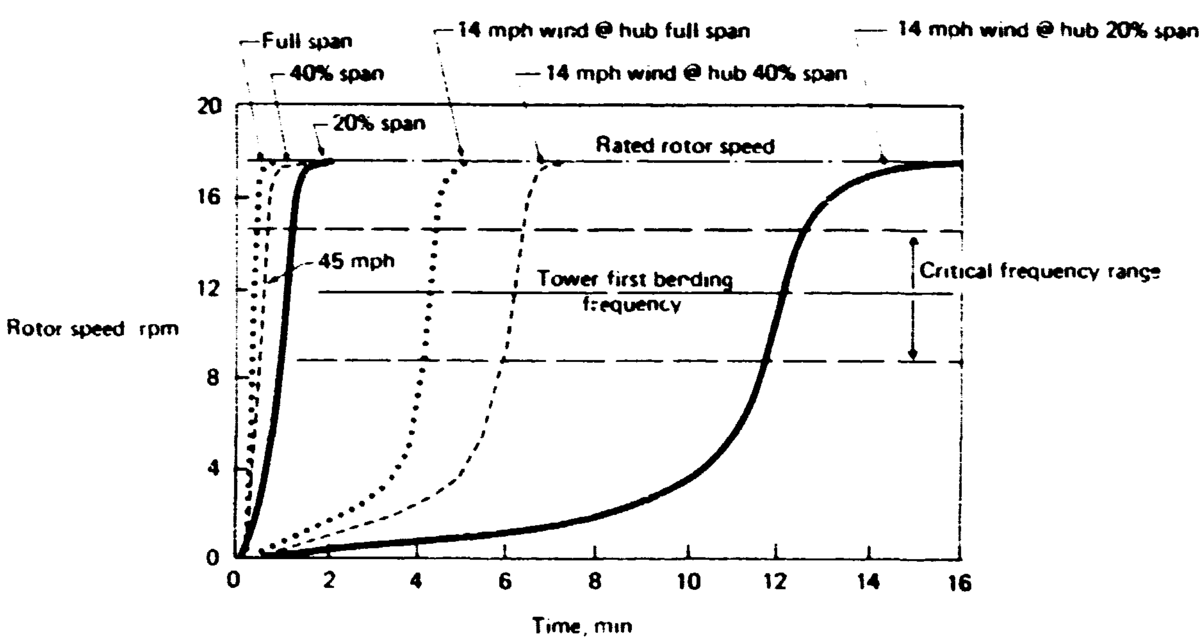


Figure 4-7. Startup Characteristics

It was established that partial span startup could be accomplished with minimum dwell time in the critical frequency range of the tower. Similarly, capability was established for normal and emergency shutdown without detrimental overspeeding of the rotor, and for ability to maintain synchronization. Relative annual energy was determined to be .995 for 40% and .985 for 20% compared to full span control.

Rotor - Considerable design simplification is possible with partial span by using carry-through structure in the hub area (See Figure 4-8).

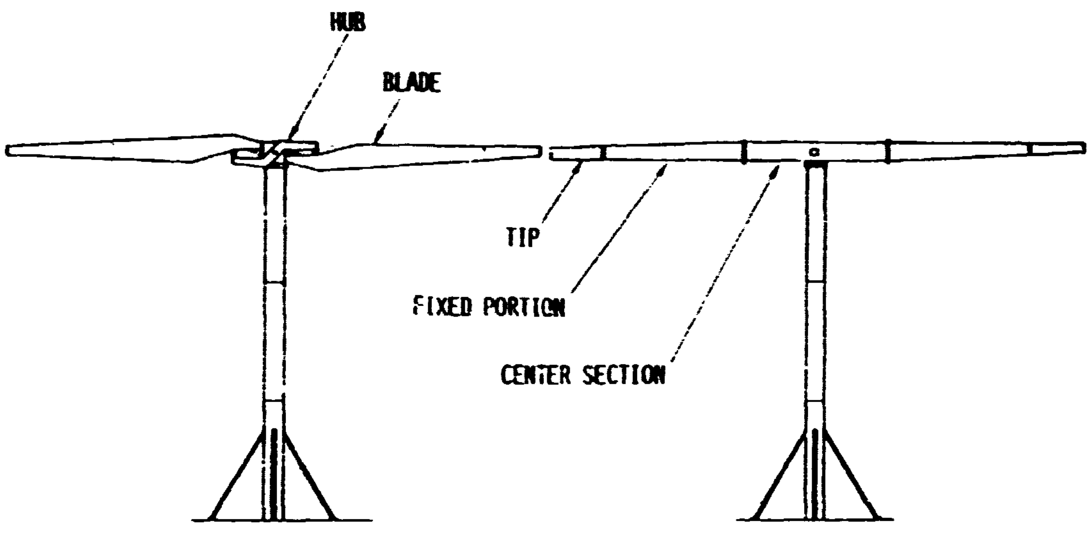


Figure 4-8. Configuration Comparison

The pitch control system is also simplified since the loads and resulting mechanism envelope are considerably reduced and the location is moved to the rotational interface. (See Figure 4-9).

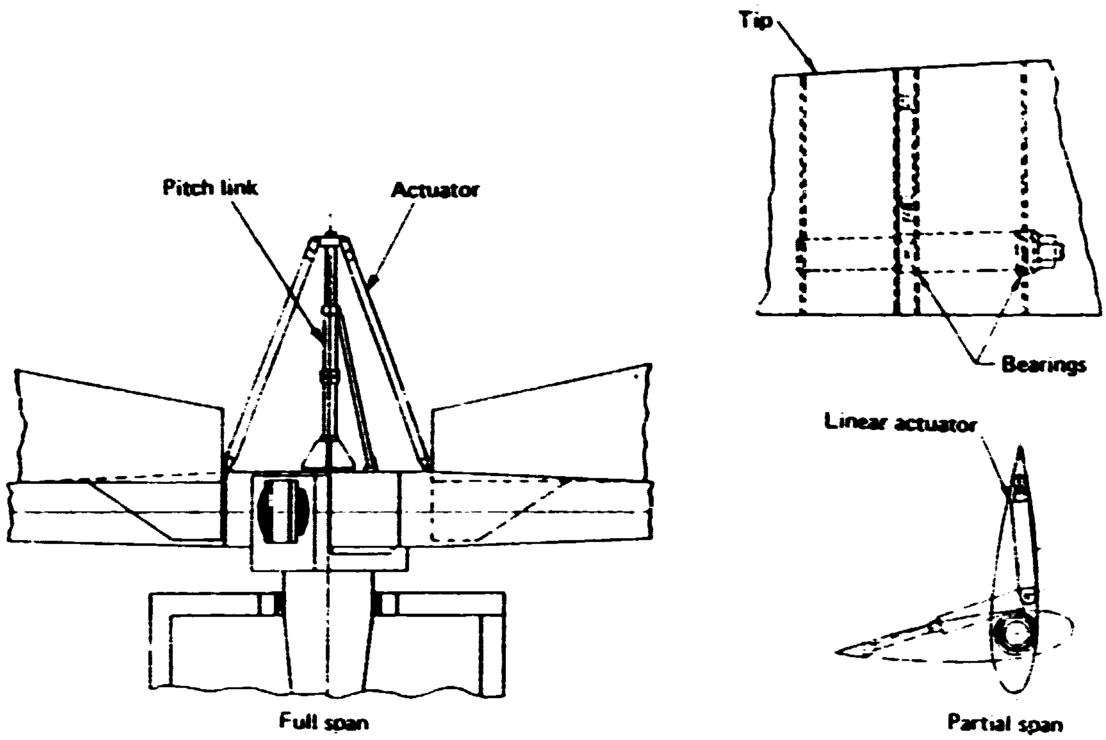


Figure 4-9. Pitch Control Comparison

Significant rotor weight reduction resulted from the reduction in high wind bending moments, elimination of the hinge overlap areas near the root, and a more efficient chordwise bending section in the root area. (See Figure 4-10).

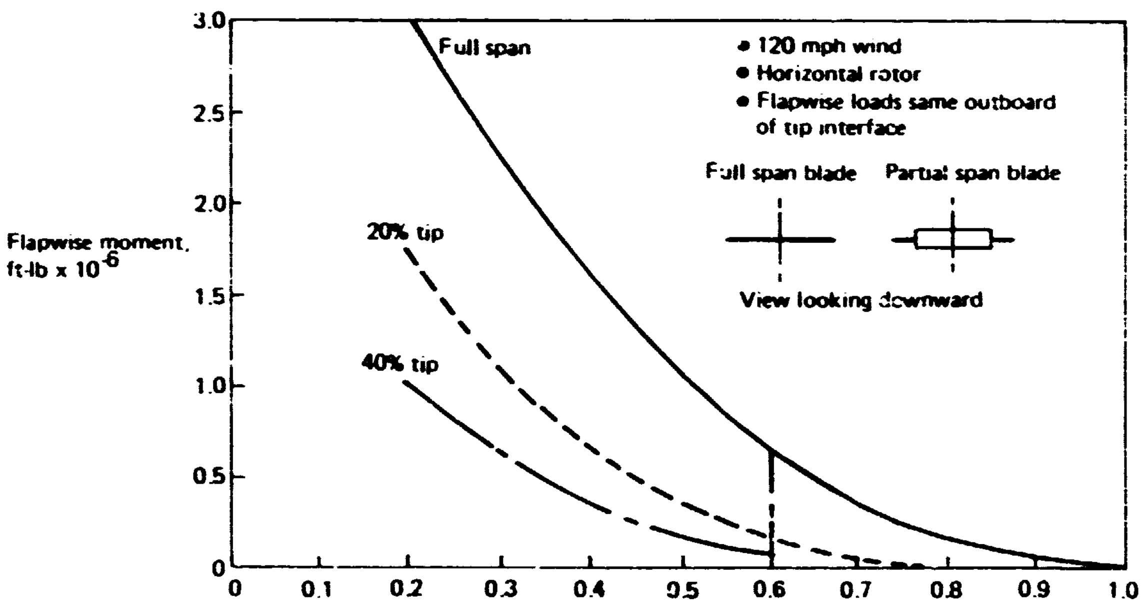


Figure 4-10. Parked Blade Loads

Drive Train and Nacelle - Design simplification of the teetering mechanism is possible with partial span control since straight-through hub structure permits easier attachments. However, a more flexible quill shaft is necessary to attenuate coriolis loads because of the increase in teeter angle excursion of the partial span blade. A longer quill shaft was required for the partial span to satisfy stiffness requirements which resulted in a longer nacelle. A longer nacelle to house a longer quill shaft was a necessary item for the partial span concept.

Tower - A lighter tower is possible with a partial span rotor since weight of structure aloft is reduced.

The trade study is summarized in Table 4-13.

Table 4-13. Partial Versus Full Span Control Summary

CONCEPT	FULL SPAN	PARTIAL SPAN		REMARKS
		20% TIP	40% TIP	
DESIGN COMPLEXITY	BASELINE	LESS COMPLEX HUB		
ERECTION	BASELINE	SAFER INSTALLATION ROTOR TO SHAFT		
TRANSPORTATION	BASELINE	$6,600 LESS	$6,700 LESS	LESS SYSTEM WEIGHT SMALLER ROTOR PARTS
WEIGHT	BASELINE	81,500 LBS LESS	83,900 LBS LESS	
COST	BASELINE	$193,500 LESS	$152,300 LESS	TIP LENGTHS IN EXCESS OF 30% REQUIRE AN EXTRA WELD SPLICE
ANNUAL ENERGY - kWh	BASELINE	1.5% LESS	.5% LESS	
RELIABILITY	BASELINE	NEGLIGIBLE DIFF.	NEGLIGIBLE DIFF.	
MAINTAINABILITY	BASELINE	$500/YR INCR. (3%)	$500/YR INCR. (3%)	
SAFETY	BASELINE	SAME	SAME	
TECHNICAL RISK	BASELINE	MARGINAL STARTUP		
COST OF ELECTRICITY, ¢/kWh	BASELINE	.41 ¢/kWh REDUCTION	.32¢/kWh REDUCTION	

CONCLUSION: The partial span control concept provides a lower cost of electricity for a large wind turbine system. A 30% span tip control is best because of the marginal startup capability of smaller control spans.

4.2.2.5 Aluminum vs Steel Tip

OBJECTIVE: The objective of the study was to determine if weight and cost could be saved by making the MOD-2 wind turbine blade tip out of aluminum.

RESULTS: Initial loads indicated that most of the tip was buckling critical, and that an aluminum tip would save both weight and cost. However, a re-evaluation of the wind environment indicated a large part of the blade tip to be fatigue critical. A weight and cost study using the new loads was performed giving the following results:

- Weight Saved Over Steel - 600 pounds
- Cost Difference

Aluminum Material Added Cost	$23,000
Aluminum Fabrication-Saving	$-7,000
Additional Cost for Aluminum...	$16,000

CONCLUSION:

For a typical tip blade section, the allowable stress of a buckling critical aluminum structure is about 2/3 that of a steel structure. Therefore, the weight of the aluminum buckling critical structure is about 1/2 that of steel. For the same fatigue loading spectrum, the allowable stress for aluminum is about 1/3 that of a steel structure resulting in structures of approximately equal weight.

In addition, the use of the aluminum tip would require development work on the aluminum structure in addition to the development on the steel structure. Therefore, the steel tip was considered best at this time for the MOD-2 WTS.

4.2.2.6 Airfoil Geometry

OBJECTIVE: A trade study was conducted to optimize the airfoil shape for the MOD-2 rotor.

RESULTS: As noted in Section 3.2.1, the NACA 230XX airfoils were selected for the baseline trade study rotor. The initial portion of this study was directed toward the selection of a second airfoil shape to be used in rotor design.

The airfoil families considered for the second airfoil shape were the standard NACA 44XX airfoils, modified NACA 44XX airfoils, and modified NACA 430XX airfoils. Profiles for the twenty-one percent thick members of these airfoil families are illustrated in Figure 4-11. The profile of the NACA 23021 airfoil is also shown for comparison. Note that the modified airfoil families were designed with blunt trailing edges. In addition, the NACA 430XX airfoils were modified near the leading edge to remove surface concavities.

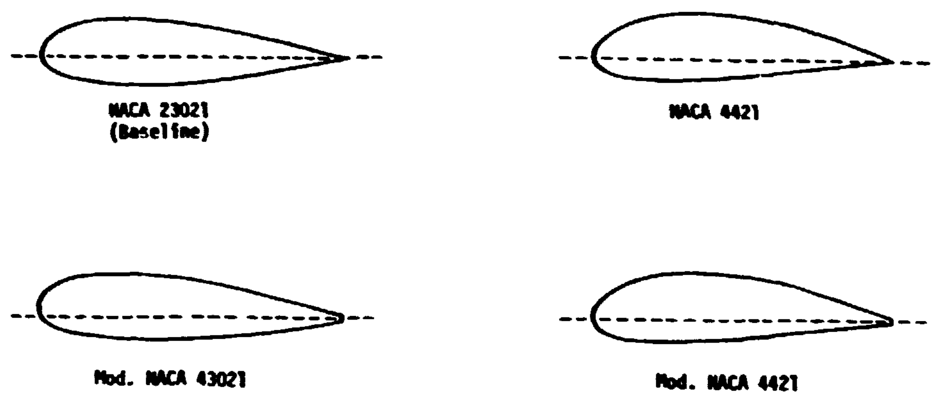


Figure 4-11. Profiles of Candidate Airfoil Shapes

A parametric comparison of the four airfoil families is presented in Table 4-14. The indicated parameters are considered relevant to airfoil selection for rotors designed to operate in the windmill state. As a result of this comparison, the modified NACA 430XX airfoils were selected as the second airfoil shape.

Table 4-14. Parametric Airfoil Comparison

PARAMETER	NACA 430XX (L.E. & T.E. Mod.)	NACA 230XX	NACA 44XX (T.E. Mod.)	NACA 44XX
C_d @ C_l = 1.0	.0138	.0147	.0140	.0162
Stall Margin (@ C_l= 1.0)	8 Deg.	6 Deg.	8 Deg.	8 Deg.
Stall Characteristic	T.E.	L.E.	T.E.	T.E.
Blunt T.E.	Yes	No	Yes	No
Potential Buckling Problems	No	No	Yes	Yes
Effect of Roughness ▷	2	1	3	4
Premature T.E. Separation	Min.	Moderate	Minor	Major
C_m (Aero Center)	-0.035	-0.01	-0.08	-0.08

▷ Smallest numbers have the least effect on aerodynamic performance

The Modified NACA 430XX airfoils were then used to design an alternate rotor. The initial design approach retained the solidity (σ = 0.030) of the baseline rotor. The resulting alternate rotor was found to produce two percent more power at the design point than the baseline rotor. However, this rotor optimized at a rotor speed of 15.9 rpm, while the baseline rotor speed was 17.5 rpm. Because the rpm decrease impacted other system components, the alternate rotor was redesigned by trading solidity for rotor speed. The final alternate rotor, which optimized at a solidity of 0.025, still produced two percent more power at the design point than the baseline rotor. The design parameters for the baseline and alternate rotors are compared in Table 4-15.

Table 4-15. Comparison of Rotor Designs

Design Parameters	Rotor Configuration	
	Baseline	Alternate
Airfoil Shape	NACA 230xx	Mod. NACA 430xx
Diameter	300 ft.	300 ft.
Design Velocity	20 mph	20 mph
Rotor Speed	17.5 rpm	17.5 rpm
Solidity (σ)	0.030	0.025
Taper Ratio	3.0	4.0
Twist	8.0 Deg.	8.0 Deg.

The two rotor designs were subjected to further analysis. For example, the alternate rotor using the modified NACA 430XX airfoils was found to produce only 0.1% more energy than the baseline rotor using the NACA 230XX airfoils. Also, for identical operating conditions, the alternate rotor was found to operate at higher load levels over the baseline. As a result, the blade weight would increase to obtain the same fatigue and buckling margins.

CONCLUSION: Based on the results of this study, the NACA 230XX airfoils were selected as most suitable for the MOD-2 rotor.

4.2.2.7 Blade Material and Configuration Studies

The current all-steel blade design configuration is described in Section 3.2.1. The rotor weight affects the weight and cost of many other items in the wind turbine system; therefore, the minimum rotor weight should correspond to the lowest wind turbine system cost.

OBJECTIVE: The objective of the study was to determine a rotor construction such that the cost of electricity is minimized.

RESULTS: Airfoil characteristics such as airfoil type, thickness, taper, twist, planform area, and tip speed determine the blade configuration. Some of these are discussed in section 4.2.2.6 (Airfoil Geometry). Within the constraints of the blade geometry, additional trade studies were made to select the spar and trailing edge materials and designs.

To select the optimum blade design, a steel "D" spar was examined in combination with 24 different concepts for the trailing edge in sufficient detail to evaluate the cost, weight, and technical risk. Table 4-16 shows relative merits of the eight most prominent design candidates.

Table 4-16. Preliminary Evaluation of Trailing Edge Concepts

BASIC CONFIGURATION	PRIMARY PARAMETER	SECONDARY PARAMETER	NUMBER OF ALTERNATE CONFIGURATIONS INVESTIGATED	RESULTS OF STUDY			
				COST (BEST)	WEIGHT (BEST)	TECHNICAL RISK	FURTHER INVESTIGATION
STEEL FACED SANDWICH	STRUCTURAL	FRAME	5	$70,000	8500#	HIGH	NO
ALUM FACED SANDWICH	STRUCTURAL	SHELL	9	$75 000	4400#	HIGH	NO
CORRUGATED PANELS	NON-STRUCTURAL	TWO AUX. SPARS	3	$57 000	5300#	HIGH	NO
STEEL/PLYWOOD LAMINATE	STRUCTURAL	FRAME	1	$59 000	7000#	MODERATE	NO
ALUM/PLYWOOD LAMINATE	STRUCTURAL	FRAME	1	$77 000	3000#	MODERATE	NO
FOAM FILLED BLOCKS	NON-STRUCTURAL	FRAME	1	$35 000	3200#	MODERATE	NO
FOAM FILLED F/G FAIRING	NON-STRUCTURAL	SHELL	1	$36 000	2800#	LOW	YES
STEEL PLATE WELDMENT	STRUCTURAL	SHELL	3	$47 000	16250#	LOW	YES
			TOTAL 24				

Two of these, a foam-filled fiberglass fairing and a steel plate weldment, were selected for a more detailed comparison with the baseline, a steel skinned honeycomb or foam filled panel construction. The three configuration candidates are shown in Figure 4-12.

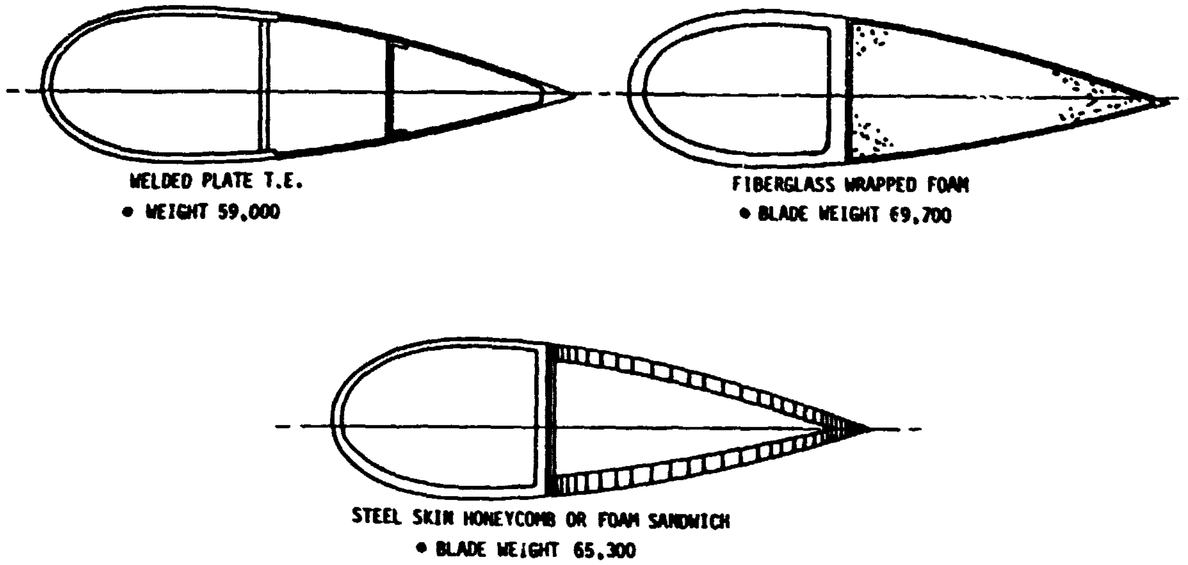


Figure 4-12. Blade Configuration Candidates

Table 4-17 shows a cost comparison for the three blade configurations, and Table 4-18 gives the final trade study results.

Table 4-17. Cost Data Comparison (Two Blades)

		MFG	MATL	2 BLADES TOTAL
STEEL	L.E.	55,520	161,812	217,332
	T.E.	28,619	13,786	42,405
		$84,139	$175,598	$259,737
STEEL SKIN Honeycomb or Foam Sandwich	L.E.	79,200	173,500	252,700
	T.E.	28,101	21,196	49,297
		$107,301	$194,696	$301,997
WRAPPED FOAM	L.E.	89,750	175,648	265,398
	T.E.	18,525	14,502	33,027
		$108,275	$190,150	$298,425

Note: L.E. designates the "Leading Edge" or steel "D" spar and T.E. the "Trailing Edge" or blade aft spar.

Table 4-18. Trade Study Summary - Blade Design

	ALL STEEL T.E. (STRUCTURAL)	STEEL SKIN HONEYCOMB OR FOAM CORE	PLASTIC T.E. (NON-STRUCTURAL)
BLADE WEIGHT(PER BLADE)	59,000 #	65,300#	69,700#
MOST SIGNIFICANT STRUCTURAL FAILURE	FATIGUE OF D-SPAR OR T.E.	FATIGUE OF D-SPAR	FATIGUE OF D-SPAR
MOST SIGNIFICANT MAINTENANCE ITEM	REPAINTING	BOND FAILURE & REPAINTING	TOUCH UP OF PLASTIC FINISH
TECHNICAL RISK	LOW	HIGH	MEDIUM
DESIGN COMPLEXITY	MEDIUM	HIGH	LOW
COST COMPARISON	$260,000	$302,000	$298,000

CONCLUSIONS: The results of this study show the all steel blade to be the optimum design for large wind turbines similar to the MOD-2 WTS.

4.2.2.8 Crack Detection System

A major wind turbine system problem is rotor fatigue. The MOD-2 rotor is subjected to approximately 200,000,000 alternating stress cycles during its 30 year design life. Historically, structures of this type occasionally develop fatigue cracks, initiating from either internal flaws or external damage. Since rotor failures are both expensive and dangerous, some means of detecting cracks prior to ultimate failure is desirable. Fail safe structure, combined with frequent inspections, serves this purpose on aircraft. However, this type structure is excessively expensive for wind turbine application and the large number of fatigue cycles requires very frequent and costly inspections. Therefore, development of an automatic crack detection system was initiated.

OBJECTIVE: The purpose of this study was to examine several potential fatigue crack detection systems and, if possible, select one for further development.

RESULTS: It was determined that a crack could be detected in a pressurized blade by measuring either the airflow through the crack or the resulting pressure drop in the blade. The essential element in the system is the comparison of the pressure or flow in the two blades to minimize the effects of ambient temperature changes. A flow system was selected for further study as it is less impacted by leaks and environment than a pressurized system.

In the proposed system, that was developed (see Figure 4-13) air is compressed to 100 pounds per square inch (psi), dried, and stored in a tank.

Air is bled out of the tank through a pressure regulator which maintains a one psi pressure downstream of the tank. The low pressure air is divided and fed through flowmeters into each of the two blades. The air exits the blades through fixed orifices. Relief valves protect the structure in case of malfunction of the pressure regulator or plugging of the exit orifices.

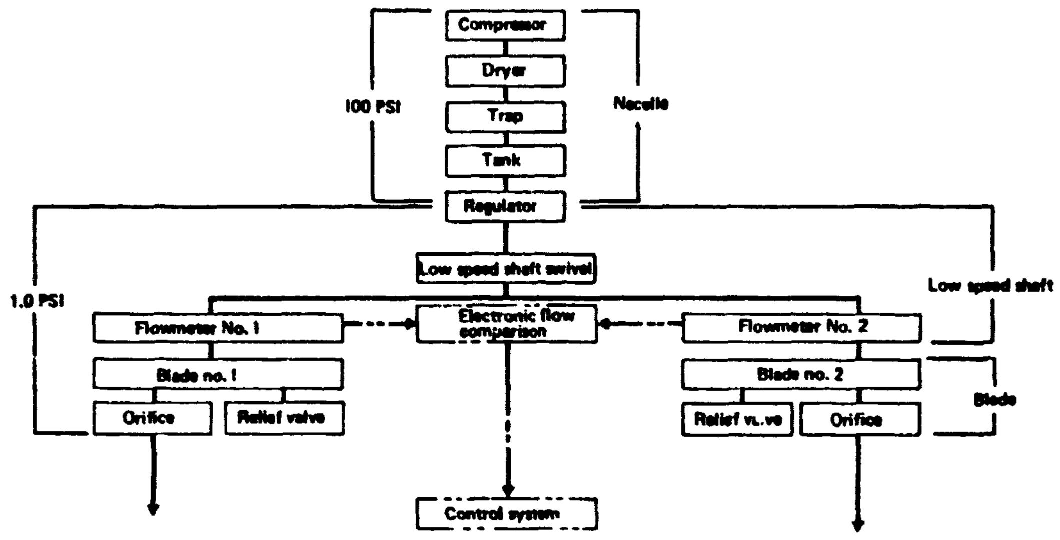


Figure 4-13. Blade Crack Detection System Block Diagram

The outputs of the flowmeters are compared. When the difference in flow exceeds a predetermined amount, a signal is sent to the control system which then shuts down the wind turbine.

The system has a threshold capability of detecting a 12 inch long crack in the MOD-2 blade. There are approximately 28 days from the threshold time that a crack can first be detected until the crack reaches a critical length of 30 inches (Figure 4-14).

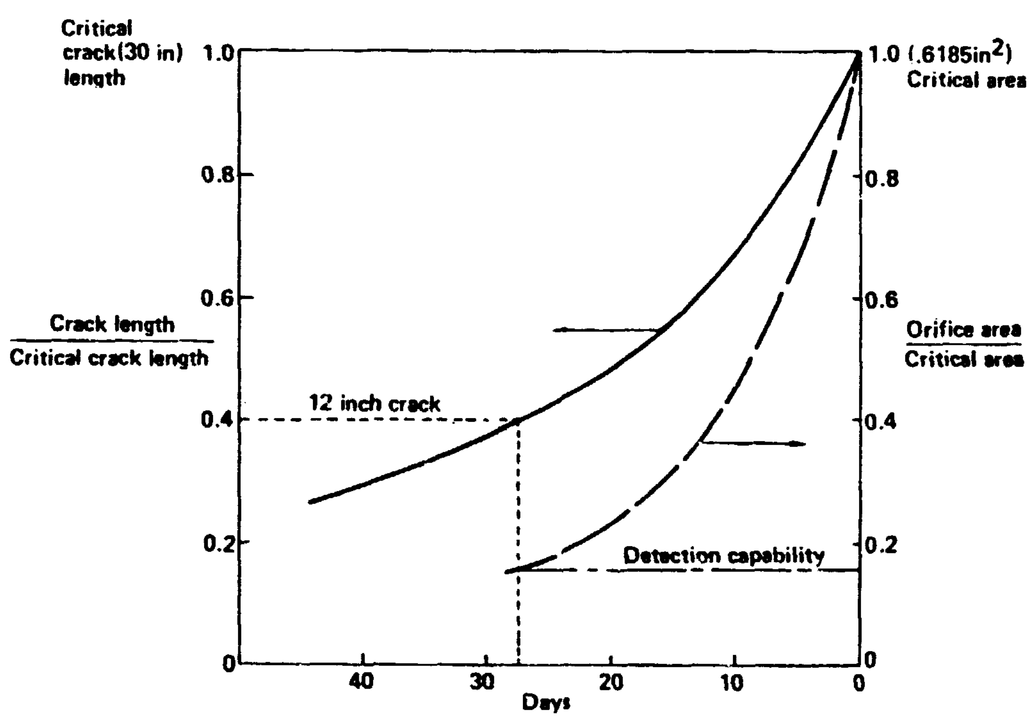


Figure 4-14. Margin Between Detectable and Critical Crack Size

The arrangement of the system for the MOD-2 wind turbine is shown in Figure 4-15.

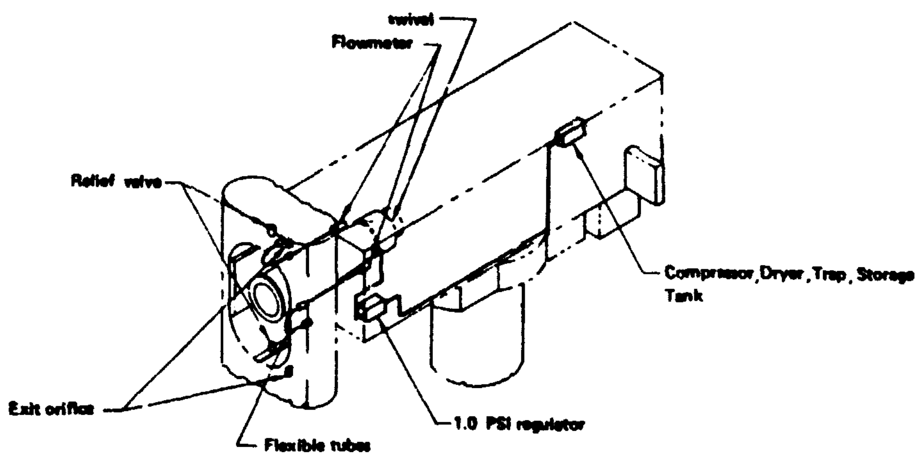


Figure 4-15. Crack Detection System

CONCLUSION: The flow rate comparison system is best because it can detect a crack 28 days in advance of complete failure, and it is less costly and has a lower risk than a pressurized system or crack stopping design (section 4.2.2.9).

4.2.2.9 Crack Stopping Design

One way of reducing the probability of a blade failure is to incorporate crack stoppers in the blade design.

OBJECTIVE: The purpose of this study was to examine techniques used for stopping cracks and evaluating their use on the MOD-2 wind turbine blade.

RESULTS: The technique used to stop cracks is to divide the skin structure into panels and use bolted joints at the edge of the panel where it is desired to stop the crack. Some of the configurations studied are shown in Figure 4-16. Crack stoppers are used on the edges of panels subject to high fatigue stresses.

The combination of welded and bolted structure is more costly to build than all-welded structure. The bolted joints give rise to stress concentrations since the welded structure tends to pick up more than its share of load. The combination of structure may be more susceptible to fatigue cracks than all-welded structures. The cracks should stop at the bolted joints, but will result in additional stress at these joints. This, in turn can result in a new crack starting on the other side of the joint. The result is an extremely short period required between inspections to ensure that a failure does not occur.

CONCLUSION: For the above reasons, a crack stopping system is not considered practical for wind turbine blades.

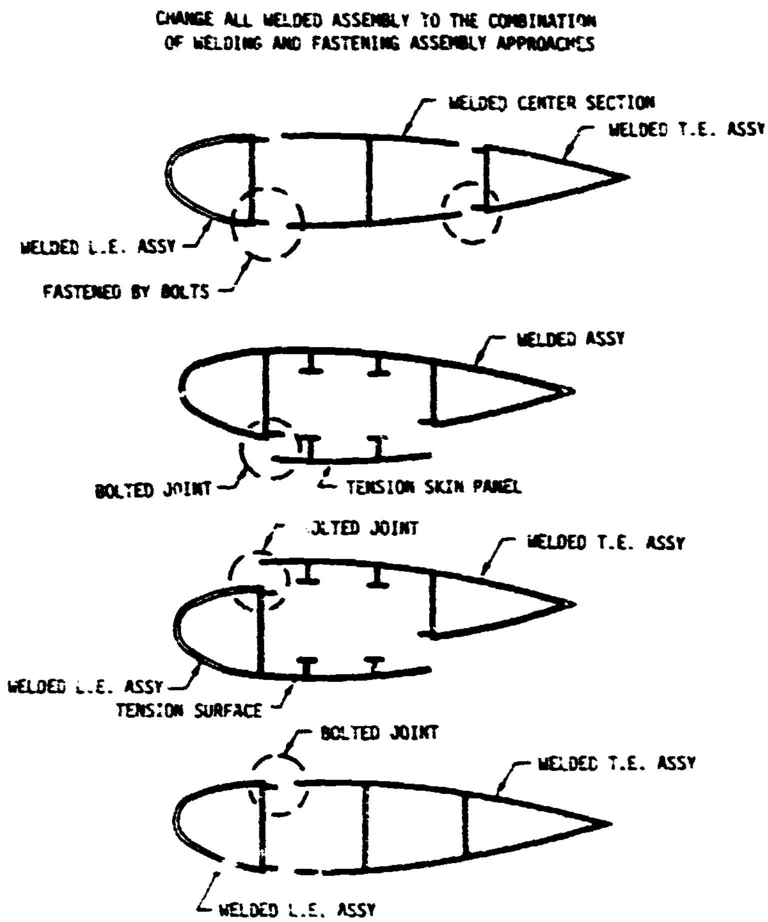


Figure 4-16. Crack Stopping Approaches

4.2.2.10 Metal vs. Composite Rotor

The baseline rotor for MOD-2 is of welded steel construction. This trade study evaluated the technical, programmatic and economic comparison between the baseline steel and a composite (fiberglass) rotor. The design and analysis leading to the development of the composite rotor is provided in section 4.3. That data is the basis for this comparison.

OBJECTIVE: The purpose of this study was to provide a technical, programmatic and economic comparison of the two rotors with the objective of recommending the appropriate blade concept for MOD-2.

RESULTS: The results of this trade study are provided in Tables 4-19 through 4-24.

CONCLUSION: As indicated in Tables 4-19 through 4-24, the steel rotor is evaluated as the more mature system with some additional development required for the composite rotor. The composite rotor configuration (sec. 4.3) is believed to be a good design for the MOD-2 system and when fully developed may offer advantages in the scenario of quantity production.

The primary reasons for recommending the continuation of the steel rotor as baseline for MOD-2 are:

1) Less technical risk.

2) A more firm understanding of material properties and fabrication processes.

3) Design maturity - no impact on prototype program schedules.

Table 4-19. Rotor Comparison Technical

	Steel	Composite
Materials	• Steel – 100%	• Steel 33 % • Filament wound fiberglass 65.5% • Layup fiberglass 1.5%
Process	• Weld	• Weld • Adhesive bond
Joint design	• Tension bolts	• Shear bolts and adhesive
Allowables	• Compression, fatigue, and joint allowables verified by test • Negligible temperature affect on allowables and modulus	• Allowables not fully verified by test • Appreciable temperature affect on allowables and modulus
Erosion/UV	• Resistant	• Requires paint and L.E. boot
Corrosion	• Requires primer and paint	• Fiberglass is resistant • Metal parts require primer and paint
Lightning	• Inherently resistant	• Requires development of protection system
Maintenance	• Weld repair of cracks • Paint	• Repair minor damage • Repair lightning damage • Paint • Replace blade for major damage

Table 4-20. Rotor Weight Comparison - Pounds

	Steel rotor	Composite rotor
Hub (excluding teeter brg.)	67,730	35,820
Inboard blade	71,810	86,280
Outboard blade	20,760	12,560
Hydraulics	740	740
Electrical	100	100
Pivot installation	7,880	6,100
Actuators and locks	880	880
Total rotor	169,570	141,480

Table 4-21. Rotor Comparison System Weight - Pounds

	Steel rotor	Composite rotor
Rotor	180,570	141,400
Drive train	65,800	97,100
Nacelle	63,200	63,200
Tower	251,470	251,470
Total system	580,210	553,400

Table 4-22. Rotor Cost Comparison (1UUth Production Unit) - Dollars

	Steel rotor	Composite rotor
Hub (excluding teeter brg.)	167,500	98,500
Inboard blade	83,300	127,400
Outboard blade	25,700	29,200
Hydraulics	5,000	5,000
Electrical	4,700	4,700
Pivot installation	33,400	33,400
Actuators and locks	2,300	2,300
System cost impact (Δ $)	–	13,000
Total	321,900	313,600

Table 4-23. Rotor Maintenance Comparison

ITEM	STEEL BLADE	FIBERGLASS	COMMENTS
Repair resulting from lightning strikes	Local repaint to prevent corrosion - estimated frequency: 6 months	Screen repair and local repaint to prevent U.V. damage and moisture penetration - estimated frequency: 6 months	No difference (Testing of fiberglass lightning protection required to verify extent of damage)
Repairs resulting from rifle fire	Rifle fire will penetrate outboard sections and chip paint on inboard sections	Rifle fire will penetrate inboard sections and imbed in surface. Will go through outboard sections.	No difference
Preventative maintenance painting	Inboard section - 8 year interval Control tips - 4½ years Average annual levelized O & M cost = $830	Entire blade once every 16 years Average annual levelized O & M cost = $460	Δ = $370 per year less for fiberglass blade
Trailing edge repairs	———	Local repair - estimated at once per 5 years. Average annual cost = $345 (includes lost power and levelized O & M)	Δ = $345 per year more for fiberglass blade

Table 4-24. Rotor Trade Study Summary

Parameter	Steel rotor	Composite rotor	Remarks
Rotor weight	Baseline	Less 20,000lb	
System weight	Baseline	Less 27,000lb	
Technical risk	Welded steel	• Welded steel • Fiberglass allowables • Joint design	Fiberglass risk difficult to assess since there is little applicable long term service experience
Maintenance	Weld repair of thru cracks	• Repair lightning damage • Replace T.E. segments • Repair minor damage	Fiberglass would require rotor replacement for major wind blade damage
System impact	Baseline	Minor	System can be designed to accept composite rotor with minor impact
Schedule (1st prototype)	Baseline	7½ month slide	Development testing not yet complete for composite rotor
Program cost (1st prototype)	Baseline	Significant increase	• Development testing • Equipment and tooling • Program delay
Rotor recurring cost (100th unit)	Baseline	Possible small decrease	• Composite is highly automated production process
System turnkey cost (100th unit)	Baseline	Negligible difference	
Cost of electricity	Baseline	Negligible difference	

Recommendation: The welded steel rotor should be retained as baseline for the MOD-2 WTS.

FIGURE 4-17 DELETED

4.2.2.11 Upwind Vs. Downwind Rotor

The current upwind rotor configuration is described in section 3.1. This trade study provided the basis for the configuration selection.

OBJECTIVE: The objective and purpose of this study was to compare the cost of electricity as generated by an upwind rotor with that produced by a downwind arrangement. The power and loads generated by MOSTAB, under identical wind conditions, served as the basis of the comparison.

RESULTS:

Rotor - It was determined that the blade flapwise and chordwise bending moments were slightly lower with an upwind rotor as shown in Figure 4-18 and 4-19. The blade loads were smaller for the upwind rotor because tower wake effects were eliminated, allowing a total blade weight saving of 3000 lb.

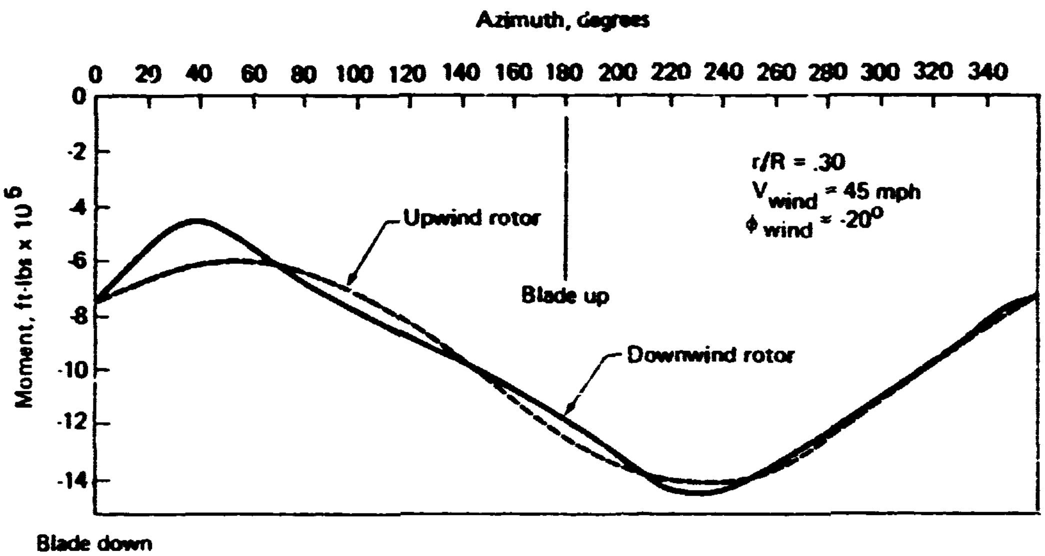


Figure 4-18. Blade Flapwise Bending Moment

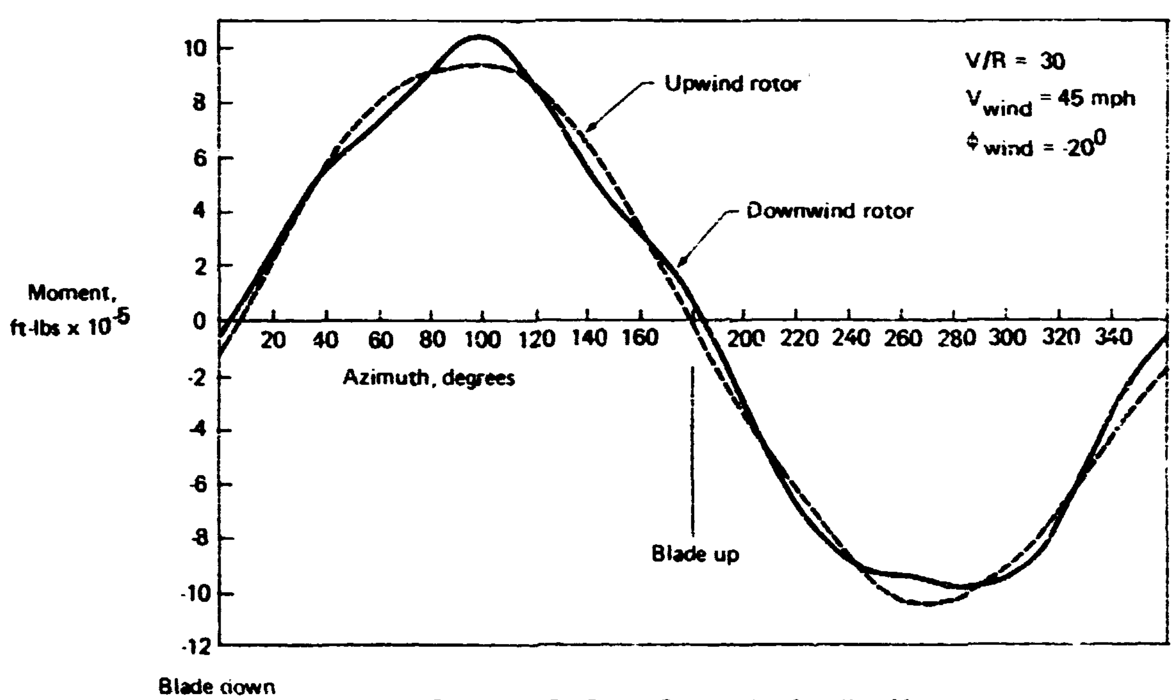


Figure 4-19. Blade Chordwise Bending Moment

Blade weight savings are illustrated in Figure 4-20. Nearly all the weight saving occured between blade stations 528 and 948 as this was the area designed by fatigue.

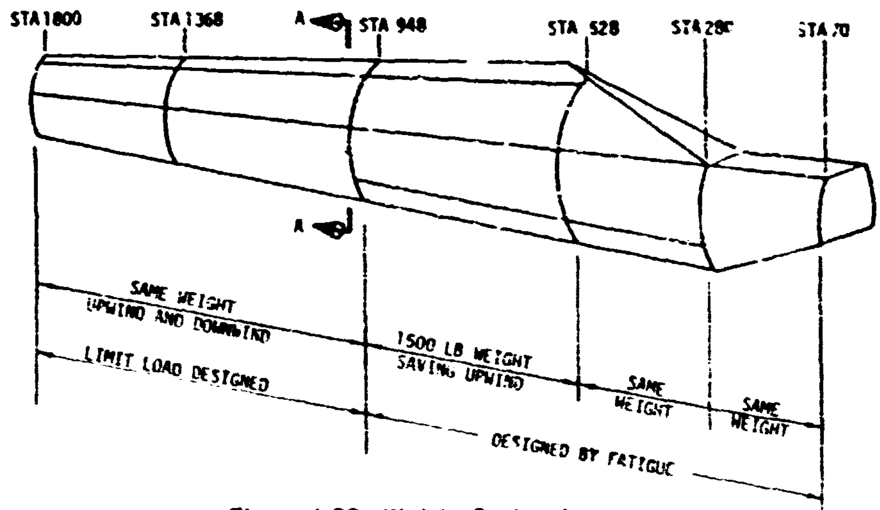


Figure 4-20. Weight Saving Areas

Yaw Drive System - The loads were of nearly the same magnitude (although at different wind velocities) as shown in Figure 4-21. Therefore, configurations were practically the same and were of like complexity and cost, and do not impact the cost of electricity.

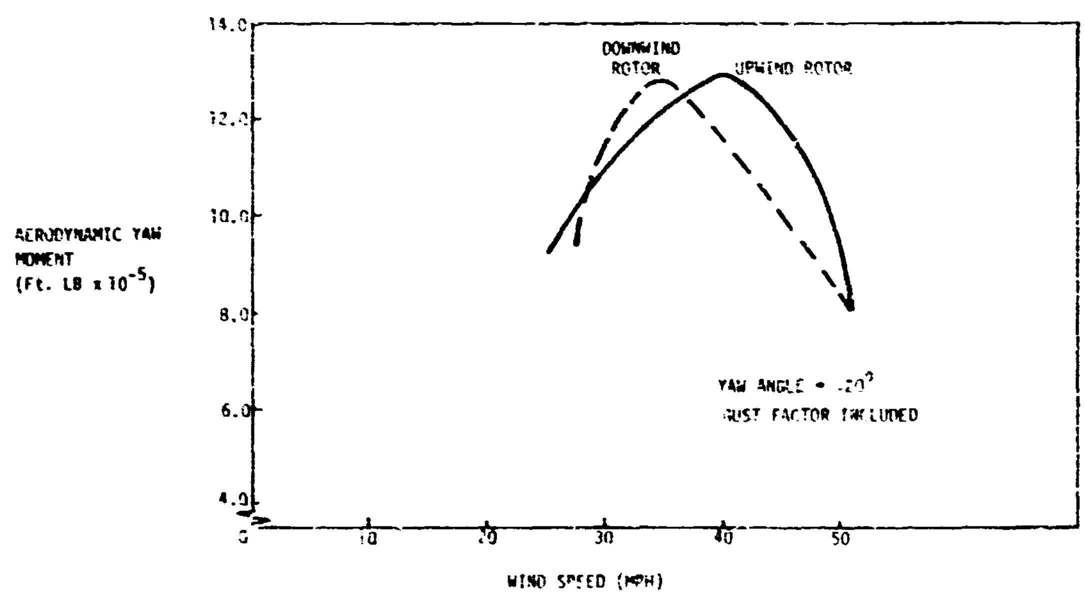


Figure 4-21. Yaw Torque Versus Wind Speed

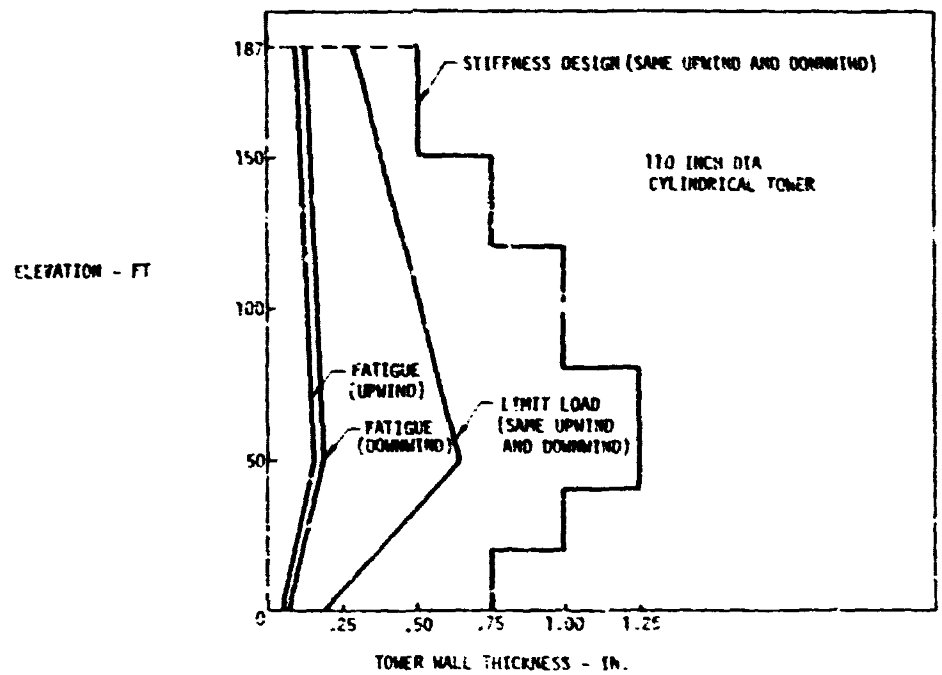


Figure 4-22. Tower Design Drivers

Tower - at the time of this study the structure was designed for torsional stiffness and therefore was of the same configuration whether upwind or downwind (Figure 4-22). Thus, the cost of electricity was not affected. It should be noted that the tower described in section 3.2.4 is now designed by fatigue and limit loads. However, this did not change the validity of the subject study nor of its conclusions.

Nacelle - The study indicated that the system loads were not sensitive to upwind or downwind rotor location, because the tower shadow effect on hub loads was minimal.

Wind Sensors - In an upwind rotor configuration, this system may be more complex and it was estimated that a cost penalty of about $5000 could be incurred.

Energy Output - The near elimination of the tower wake in an upwind rotor installation results in a 2.6% increase in annual energy output, thus further reducing the cost of electricity.

Costs - The upwind blade cost is $3000 less than for the downwind configuration but the upwind wind sensor cost is $5000 more. This resulted in an overall WTS cost increase of $2000; however, the cost of electricity was reduced because of the higher energy output of the upwind rotor configuration.

This trade study is summarized in Table 4-25.

Table 4-25. Upwind/Downwind Trade Study Summary

	Upwind	Downwind	Remarks
Design complexity	Same		
Erection	Same		
Transportation	Same		
Weight	Baseline	1.2% higher	Higher fatigue loads downward
Annual energy output	Baseline	2.6% lower	Output increase with upwind rotor because tower shadow is eliminated
Cost	Baseline	0.1% lower	100th WTS costs $2,000 more (upwind)
Reliability	Same		
Maintainability	Same		
Safety	Same		
Technical risk	Same		
Cost of electricity	Baseline	Adds 0.1 ¢ / KW hr	2.6% less (upwind)

CONCLUSION: Based on the results of this study, it is concluded that the upwind rotor location was superior for wind turbines of MOD-2 size as it results in lower cost of electricity.

4.2.2.12 Tip to Ground Clearance

The current tower configuration provides a rotor tip to ground clearance of 50 feet. This trade study provided the basis for the selection of this configuration.

OBJECTIVE: The objective of this study was to compare the cost of electricity generated using the baseline 50 foot rotor tip to ground clearance configuration with that generated using 25 foot, 75 foot, and 100 foot rotor tip to ground clearance configurations.

APPROACH: The two major factors that vary with rotor height and affect the cost of electricity are the annual energy produced by the WTS and the initial costs of the tower and foundation. Due to the effect of the wind gradient, an increase in rotor height results in an increase in the annual energy produced by the WTS. However, an increase in rotor hub height also results in an increase in tower height and overturning moment at the tower base. Thus, the tower weight and foundation size increase results in higher initial costs of these two items. The study approach was to design towers and foundations for 25 foot, 50 foot, 75 foot, and 100 foot rotor tip to ground clearance configurations and to compare the effect of the annual energy produced and the initial costs of these configurations on the cost of electricity.

RESULTS: The study resulted in a slight increase in cost of electricity for the 75 foot and 100 foot tip clearance configurations as compared to the 50 foot tip clearance baseline. The cost of electricity for the 25 foot tip clearance and the 50 foot tip clearance configurations were essentially the same. However, the 50 foot tip clearance configuration was selected based on a considerable increase in power produced per year, resulting in less units required to produce a specific power requirement. See Table 4-26 for a summary of study result

Table 4-26. Rotor Tip to Ground Clearance Study

Criteria	Ground Clearance/Tower Height				Remarks
	25/175	Baseline 50/200	75/225	100/250	
Tower costs (\$) (Δ)	-47,100	0	53,400	113,100	Increased costs proportional to Δ tower weight
Transportation cost (\$) (Δ)	0	0	1,800	1,800	Increased tower size and weight resulted in increased trucking costs
Site preparation costs (\$) (Δ)	-18,900	0	20,200	40,900	Higher towers require larger foundations due to increased base moment
Erection costs (\$) (Δ)	-6,200	0	6,500	14,000	Increased tower parts, weight and heights results in higher erection costs
Rotor costs (\$) (Δ)	6,700	0	-2,600	-3,900	Fatigue loads decrease with height
System cost - 100th unit (\$) (Δ)	-65,500	0	79,300	165,900	
Operation and maintenance costs per year (\$) (Δ)	0	0	0	0	Maintainability changes are negligible
Machine power coefficient (Cp) (Δ)	-.008	0	.004	.005	Rotor Cp increases due to less relative change in wind velocity at higher rotor heights
Energy out (Kwh per year) (Δ)	-353,400	0	353,400	636,200	
Cost of electricity (Δ)	0	0	Increases .03¢/kwh	Increases .09¢/kwh	Increased power out from higher towers offset by increased system costs

CONCLUSION: The optimum tip to ground clearance for wind turbines the size of MOD-2 is 50 feet.

4.2.2.13 Tilted Vs. Non-tilted Rotor

The MOD-2 WTS configuration has the rotor shaft positioned horizontally without any tilt (section 3.0).

OBJECTIVE: The main purpose of this study was to assess the potential advantages of a tilted rotor shaft, especially with respect to the resulting cost of electricity.

RESULTS: Figure 4-23 shows that the cost of electricity could be reduced slightly for small angles of tilt. This is because the tilt allows less nacelle overhang for constant blade to tower clearance. At higher angles of tilt, the increased blade alternating flap loads cause higher rotor weight and costs. This increase, plus the lower energy output, soon negates any advantage of tilt.

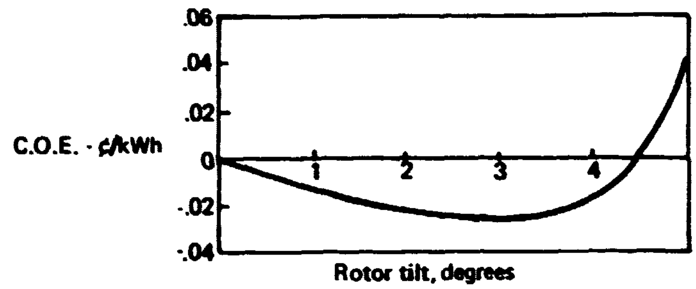


Figure 4-23. Cost of electricity Versus Rotor Tilt

CONCLUSION: The non-tilted rotor will be maintained for MOD-2 at the present time. The only significant advantage of small angles of tilt would be to gain blade clearance if it should be found necessary.

4.2.3 Drive Train

This section reports the results of trade studies conducted to evaluate alternative drive train configurations.

4.2.3.1 Epicyclic vs. Parallel Shaft Gearbox

This section reports the last of several trade studies conducted to select the optimum gear box for the MOD-2 wind turbine.

OBJECTIVE: The objective of this trade study was to compare the relative merits of a parallel shaft gearbox and an epicyclic type gearbox. They were compared primarily with respect to cost of electricity and impact on other WTS subsystems.

RESULTS: The two candidate gearboxes were the parallel shaft design by Philadelphia Gear Company (USA) and the epicyclic gearbox by STAL LAVAL (Sweden). The general features of each gear box and their effect on the WTS are shown in Table 4-27. As is noted, the large size of the parallel shaft gear box requires a much larger nacelle and its higher weight significantly increases the weight of both the nacelle and tower. The selection matrix is shown in Table 4-28 and shows that the cost of electricity is 16% higher for the parallel gear box wind turbine configuration.

Table 4-27. Comparison of Alternative Gearbox Designs

PARALLEL SHAFT

- PROVEN AS HIGH TORQUE DRIVE
- HEAVY AND LARGE
- MOD-1 CONCEPT
- ANTI-FRICTION BEARINGS
  - LOWER BREAKAWAY TORQUE
  - FINITE LIFE
- REQUIRES TWO LOW SPEED FLEX COUPLINGS
- VERTICALLY OFFSET SHAFTS OF HIGH CONFIGURATION PERMIT HOLLOW SHAFT ACCESS
- LOW CONFIGURATION SENSITIVE TO STRUCTURAL DEFLECTIONS OF FOUNDATION AND NACELLE

COMPACT PLANETARY GEAR (C.P.G.)

- IMPROVEMENT OVER STOECKICHT DESIGN
- PROTOTYPE C.P.G. TESTED SUCCESSFULLY
- 1/3 THE SIZE AND WEIGHT OF PARALLEL SHAFT CONFIG.
- SLEEVE BEARINGS
  - HIGHER BREAKAWAY TORQUE
  - LONG LIFE (WITH ADEQUATE LUBRICATION)
- SOFT MOUNT ELIMINATES ONE LOW SPEED FLEX COUPLING AND FOUNDATION INFLUENCE
- FLEX LUBE CONNECTION REQ'D

Table 4-28. Philadelphia Versus Stal Laval Evaluation Summary

	PHILADELPHIA GEAR	STAL-LAVAL	REMARKS
SYSTEM IMPACT	PENALIZED SYSTEM BECAUSE OF SIZE	FAVORABLE	
RELIABILITY	SAME		BOTH SATISFACTORY
MAINTAINABILITY	REQUIRES REMOVAL OF GEAR BOX FROM NACELLE FOR MAJOR REPAIR	ALL MAINTENANCE WORK CAN BE ACCOMPLISHED IN NACELLE	
TECHNICAL RISK	LEAST	SCHEDULE RISK DUE TO NEW DEVELOPMENT	BOTH BOXES NEW DESIGN. STAL-LAVAL NEW CONCEPT
PROCUREMENT SCHEDULE	52 WEEKS	56 WEEKS	BOTH ACCEPTABLE
WEIGHT (GEAR BOX)	115,000 LB	35,000 LB	STAL-LAVAL REDUCES SYSTEM WEIGHT 109,315 LBS
ANNUAL ENERGY	8,911,224 kWh	9,153,965 kWh	STAL-LAVAL HAS HIGHER EFFICIENCY
SYSTEM COE	3.89 ¢/kWh	3.34 ¢/kWh	STAL-LAVAL REDUCES COE BY 0.55¢/kWh

CONCLUSION: For wind turbines the size of the MOD-2 WTS, an epicyclic gearbox is the optimum configuration.

4.2.3.2 Shaft Configuration Studies

4.2.3.2.1 Low Speed Shaft Configuration

The current low speed shaft configuration of the drive train is described in section 3.2.2. This trade study provided the basis for the configuration selection; however, the study design may differ in certain details from that shown in section 3.2.2.

OBJECTIVE: The purpose of this study was to compare alternate shaft support systems for the rotor. Configurations studied are shown in Figure 4-24.

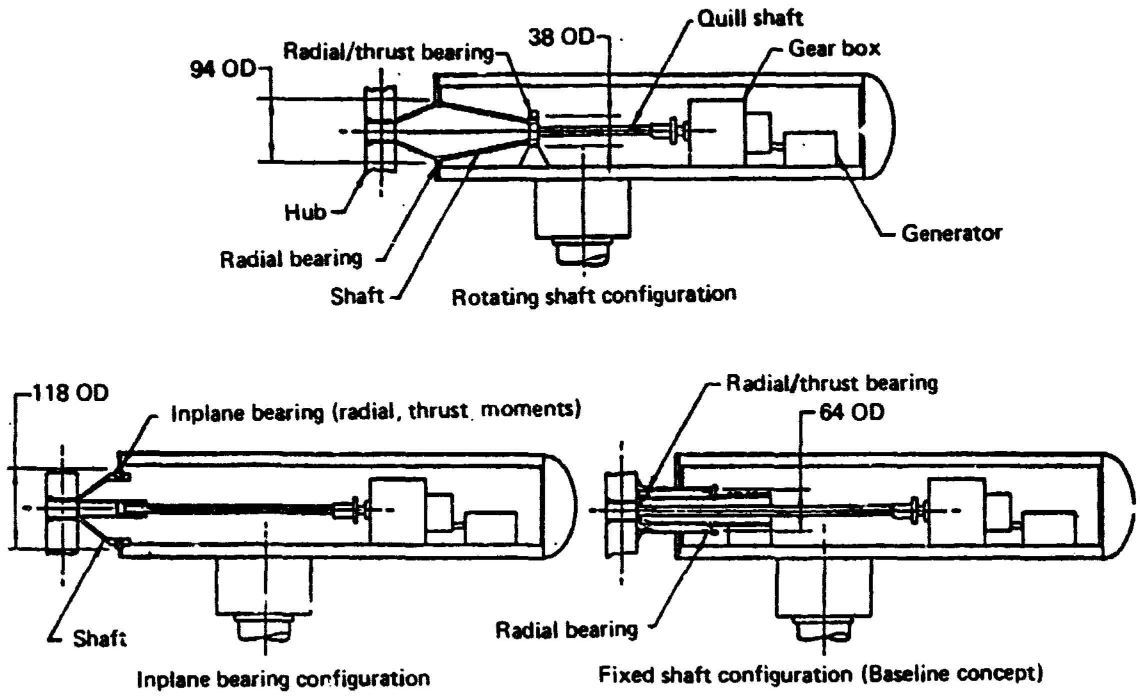


Figure 4-24. Low Speed Shaft Design

RESULTS/CONCLUSION: It was determined that the rotating shaft configuration concept supported by the fixed and floating bearings was the best choice on the basis of low cost and low weight. The study results are summarized in Table 4-29.

Table 4-29. Study Summary - Alternative Shaft Support System

CONCEPT	INPLANE	ROTATING SHAFT	FIXED SHAFT	REMARKS
DESIGN COMPLEXITY	SIMILAR		MORE COMPLEX	CONSIDERS SHAFT, BEARING, PITCH SYSTEM; SUPPORT STRUCTURE
ERECTION	SIMILAR			ALL CONFIGURATIONS INSTALLED IN NACELLE PRIOR TO ERECTION
TRANSPORTATION	\$2900	\$1200	\$2100	PER THOUSAND MILES
WEIGHT	103,500 lb	44,500	75,000	INCLUDES SHAFT BEARINGS, QUILL SHAFT AND NACELLE MOUNTING STRUCTURE.
SHAFT ASSY COST	\$231,000	\$97,000	\$156,000	
ANNUAL ENERGY - KWH	SAME			
RELIABILITY	SAME			
MAINTAINABILITY	NORMAL		DIFFICULT	BEARINGS NOT ACCESSIBLE ON FIXED SHAFT
SAFETY	SAME			
TECHNICAL RISK	HIGHER	LOWER	NORMAL	MORE GROWTH CAPABILITY WITH ROTATING SHAFT
TORSIONAL STIFFNESS	SATISFACTORY			
COST OF ELECTRICITY, ¢/KWH	ADDS .16	SUBTRACTS .12	BASELINE	FIXED SHAFT BASELINE

4.2.3.2.2 Hydraulic System Trades

The current pitch control hydraulic system design was developed after considering several factors as follows:

Emergency Feather - Being capable of reliably stopping the rotor in the case of an emergency including a loss of power or a loss of electrical control was considered to be one of the most important design features of the pitch control system. This required an auxiliary energy source independent of the primary source. The choice of a compressed gas emergency hydraulic accumulator was made. The hydraulic system also has the capability of controlling the pitch rate toward the feather position without the benefit of the electrical control system. Other methods, such as electrically-driven actuators and mechanically driven actuators, could not meet both requirements reliably and cost effectively.

System Stiffness - A tip controlled WTS requires a mechanism that has high stiffness, low lost motion and high response. The hydraulically controlled linear actuator met these requirements without undue costs.

Maintenance - Although many hydraulic systems have troublesome leakage problems, the MOD-2 pitch hydraulic system is designed to keep these to a minimum. For example, no hydraulic swivel joint is used between the nacelle and the rotating shaft. The hydraulic system is mounted to the shaft thus avoiding the necessity of the swivel. The hydraulic reservoir has been tested and found to function satisfactorily in the rotating environment.

No transfer bearing is required with the MOD-2 system. Mounting the hydraulic system on the low speed shaft eliminated the need for transferring tip motion from fixed actuators in the nacelle to the rotating system.

4.2.3.2.3 Drive Train Shaft Alignment

The drive train flexible couplings are limited to the high speed shaft and are defined in section 3.2.2.

OBJECTIVE: The purpose of the initial trade study was to determine the most cost effective and compatible shaft couplings that would satisfy the alignment requirements. Subsequent gearbox configuration decisions (sec 4.2.3.1) eliminated the need for low speed shaft flexible couplings.

RESULTS/CONCLUSIONS: The coupling configuration trade study is summarized in Table 4-30. Subsequent evaluations of high speed couplings ccnfirmed the choice of flex-disc types based on cost and lack of lubrication maintenance.

Table 4-30. Shaft Alignment Couplings

	GEAR TOOTH	DIAPHRAGM	FLEX PACK	REMARKS
COMPLEXITY	MOST	LEAST	MULTIPLE (IDENTICAL) PARTS	FLEX PACK STANDARD VERTOL DESIGN PRACTICE
WEIGHT	>14,000 #	1660 #	8000#	DIAPHRAGM WT. DOES NOT INCLUDE HUBS TO ATTACH TO SHAFT & GEAR BOX
COUPLING COST	$72,000	$88,000	$34,000	
RELIABILITY	GOOD	BETTER		
MAINTAINABILITY	REQ'D	NONE	NONE	
TECHNICAL RISK	LOW	HIGHER	LOW	FLEX PACK FAILURE MODE IS PROGRESSIVE & CAN BE SEEN DURING PERIODIC INSPECTION

4.2.3.2.4 Quill Shaft Stiffness Study

The current quill shaft configuration is described in section 3.2.2. This section reports the results of studies conducted to optimize quill shaft torsional stiffness.

OBJECTIVE: To minimize drive train torsional fatigue problems and to maintain steady electrical power output by isolating the gearbox and generator from torsional oscillations of the rotor, which are caused by Coriolis accelerations due to blade teetering and flapping.

RESULTS: The objective was met by using a "soft" quill shaft, which places the rotor/generator torsional natural frequency at .5 per rev, which is well below the torsional forcing frequency of 2 per rev. This limits the drive train alternating torques at 2 per rev to less than ± 10% of rated torque under all operating conditions. This soft shaft configuration becomes essential with the choice of the teetering partial span rotor, due to its somewhat higher alternating shaft torques. The soft quill shaft is preferable to a fluid coupling, which has a substantial energy loss and therefore results in much higher cost of electricity. A graphical comparison of drive train alternating torques with a stiff quill shaft, fluid coupling and soft quill shaft is shown in Figure 4-25. It was determined that a shaft torsional stiffness of not greater than 2 x 10^8 in-lb/rad would satisfy the requirement. With this stiffness requirement established, various quill shaft designs were evaluated such as solid and hollow, high strength and low strength materials, and with and without flexible couplings. Early configurations, where spring rate was adjusted primarily as a function of quill shaft length, with an equivalent change in nacelle length, were replaced with the present configuration where the quill shaft was nested within the low speed shaft assembly. The deletions of the flexible coupling (s) due to the choice of gearbox (reference paragraph 4.2.3.1) simplified this portion of the drive train design.

CONCLUSION: A "soft" quill shaft design was selected to satisfy dynamic requirements of the MOD-2 drive system.

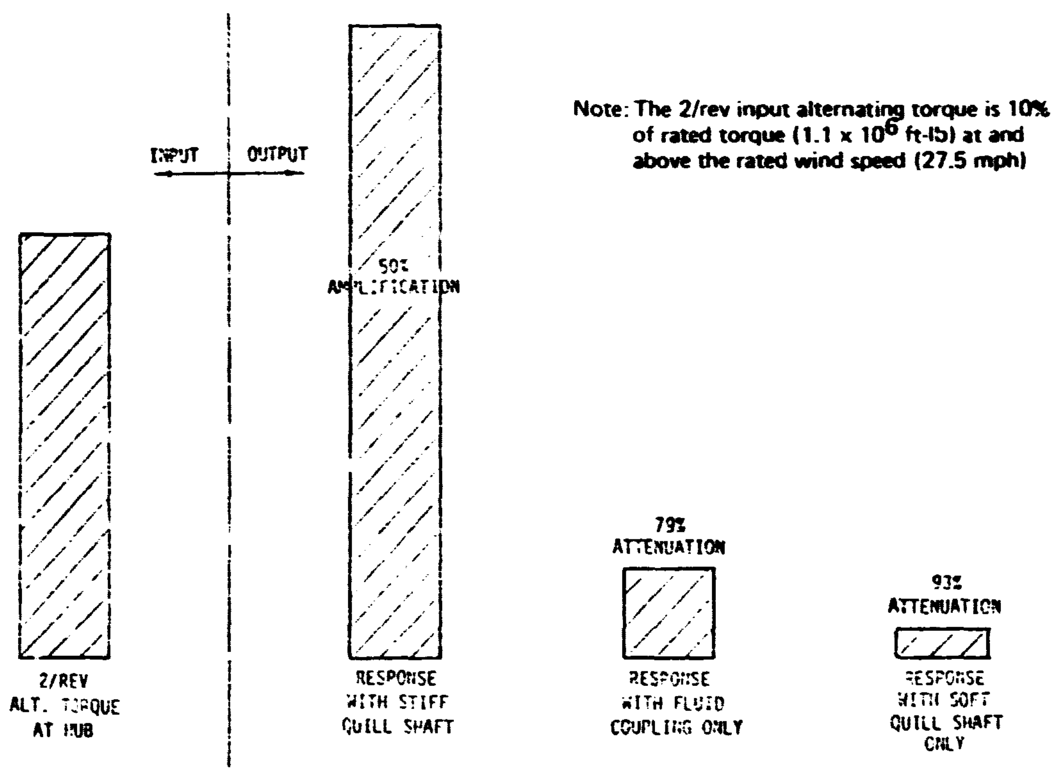


Figure 4-25. Response to 2/Rev Torsional Forcing Functions

4.2.4 Electrical Power System

This section reports the results of trade studies which evaluated alternative electrical power system characteristics.

4.2.4.1 Generator Selection

As noted in section 3.0, a 4 pole, 1800 rpm, 4.16 kV synchronous generator has been identified for use in the MOD-2 WTS. A high speed synchronous machine was selected after comparing it to induction and direct current machines applied in a number of system configurations and operated at a variety of speeds. Figure 4-26 illustrates the organization of the configurations considered; it defines their relationship, and it identifies the major trade studies conducted. Minor trade studies, not shown in the Figure, refined some of the details of the final configuration.

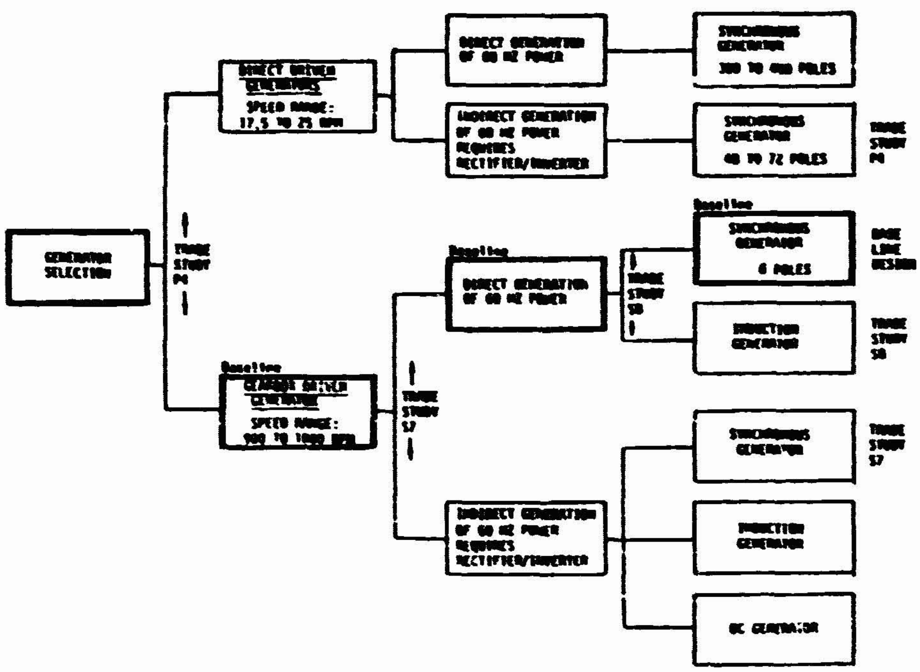


Figure 4-26. Generator Selection Trade Studies

4.2.4.1.1 Direct vs. Gearbox Driven Generator

OBJECTIVE: The generator selected is coupled to a gearbox that increases the 17.5 rpm speed of the rotor to the 1800 rpm used by the generator. Elimination of the costly gearbox by directly driving a generator was considered. A study was conducted to compare the cost of electricity as generated by systems reflecting the two approaches.

RESULTS: As shown in Figure 4-26 two direct drive electrical system concepts were explored and compared to the baseline design. In the first concept the generator was designed to provide a 60 Hertz output while driven at the low rotor speed. Depending on exact speed, the generator could have up to 400 poles. A 2500 kW machine with this many poles could require a rotating field up to 50 feet in diameter while its thickness need be less than one foot. A pancake machine of this size and proportions would require a great deal of material to structurally support the active electrical elements.

Weight and cost of the generator would be 20 to 40 times that of a conventional machine. Because the cost was so high, the concept was given no further consideration. The second concept for a direct drive system was a more reasonably proportioned pancake machine to generate low frequency to be rectified to direct current and then inverted back to 60 Hertz power by a line commutated static device. Figure 4-27 describes the power conversion component sizing for this type of direct drive system which would be more reasonably sized at an increased rotor speed of 22.5 RPM. Two parts of this system are very high cost: the pancake generator and the power conversion electronics. Either of these two electrical items would cost more than a conventional gearbox. A trade study summary is provided on Table 4-31. It became very clear during the study that to provide an output of 2500 kW with an input of 17.5 rpm would involve the use of an extremely costly generator. At very low rotor speeds, the cost effective approach was to use a gearbox and conventional generator.

CONCLUSION: The study concluded that the direct drive system was not economically competitive with a conventional generator and gearbox.

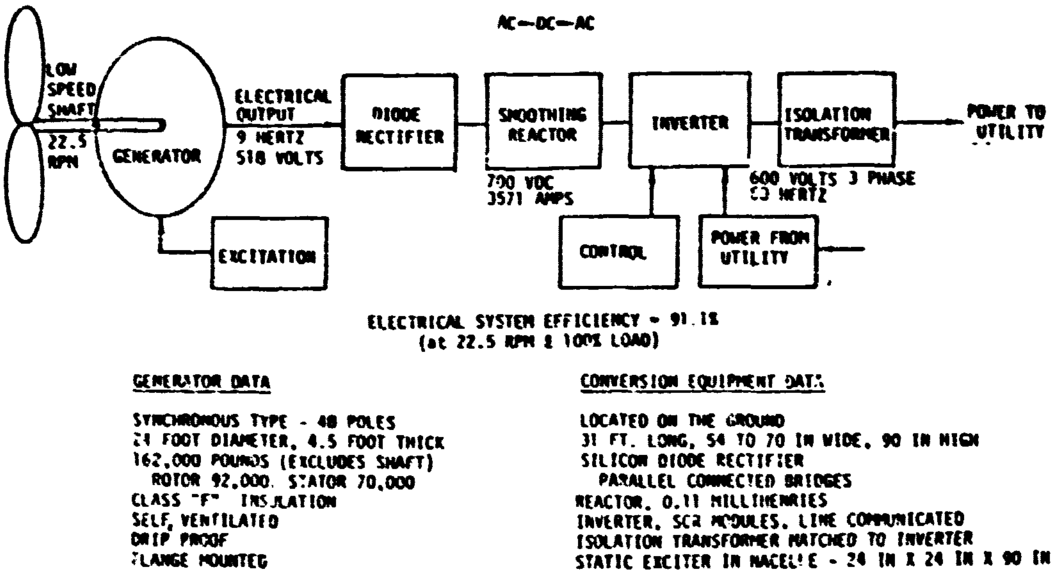


Figure 4-27. Direct Drive System

4.2.4.1.2 Direct vs. Indirect Power Generation

OBJECTIVE: Indirect Power Generation for this study was defined as the generation of unregulated AC power and the use of a static inverter type of device to condition the WTS output to 60 Hertz. Direct generation was defined as use of a generator that provided a 60 Hertz output. The purpose of the study was to compare the cost of electricity as generated by a variable speed high efficiency rotor with that produced by a constant speed rotor. By permitting variable speed the rotor could be operated at closer to its maximum aerodynamic efficiency at all wind velocities within the working range of the MOD-2 WTS. For the electrical power system to accept a variable speed at the generator shaft, it was necessary to make provisions to convert its unregulated output to 60 Hertz. Figure 4-28 illustrates functionally the two systems that are compared and the method of providing a regulated output from an unregulated generator.

Table 4-31. Direct Drive Versus Gearbox Driven Generator Study Summary

	Direct Driven	Gearbox Driven (Baseline)	Remarks
Design complexity	Diodes, filters, SCRs for rectifiers & inverter	Gearbox & high speed shaft	AC-DC-AC direct driven system
	48 pole generator	8 pole generator	[illegible] use 4P generator in baseline
	Larger nacelle cross-section 24' dia. generator	6' dia. generator, 12' wide x 10' high gearbox	Nacelle size must increase for direct drive
	Relaxed speed control will accept wind gust and wind shadow torque variations	Must be synchronous RPM requiring detail control of system design	Major advantage of direct drive system. Other options are available
Facility impact	31' x 6' x 6' high electronic package & transformer on ground	8' x 8' x 8' high transformer on ground	
Control system impact	Inverter control is self contained		Both have similar generator controls
Utility interface	Requires utility power for operation	Capable of independent operation	Power for inverter commutation required on direct drive
Efficiency	91.1%	92.1% (96% generator, 96% gearbox)	Rated load and speed
Weight impact	Generator weighs 192,000 pounds	Generator and gearbox less than 100,000 pounds	Expect additional weight in nacelle and tower on direct drive
Reliability	Significantly greater parts count	Analytically more reliable	Both satisfactory
Maintainability	Requires electronic component replacement	Uses simple scheduled mechanical maintenance	Both satisfactory
Technical Risk	Mature designs available for low risk		Both satisfactory
Power quality	Smoother power delivery	Considered satisfactory	Potential for interference from inverter on direct drive
Equipment procurement lead time	14 months minimum, 17 months normal, (are source required)	6-8 months for generator, 12 months minimum for gearbox	
System cost - 100kh unit	$1,853,000	$280,000 to $800,000	$350,000 most likely for generator & gearbox
Cost of electricity	Adds more than 1¢ per kwh		

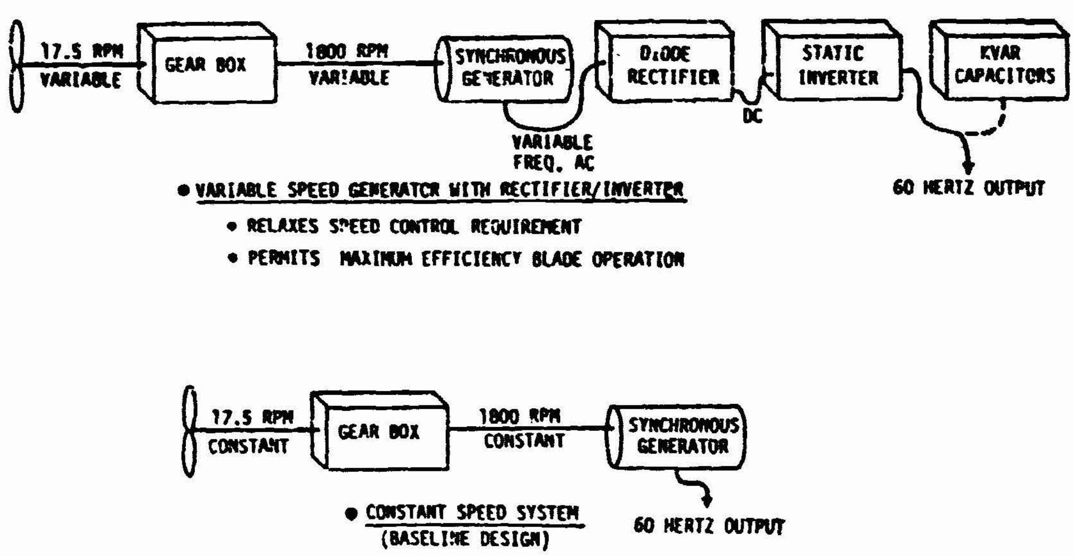


Figure 4-28. Indirect and Direct Power Generation Functional Block Diagram

RESULTS: The study determined that for a 2500 kW output:

1. A variable speed rotor could capture 5.9 percent more energy than a constant speed rotor.

2. The resultant variable speed electrical system could produce 3.0 percent more energy than the equivalent constant speed system.

3. Electrical equipment for the variable speed system would cost $335,000 more (at hundredth unit) than that for the constant speed system.

4. The electrical system efficiency at rated output would be 90.8 percent for the variable speed system as compared to 95.0 percent for the constant speed system.

5. Over the life of the wind turbine system, in the wind spectrum defined, variable speed operation would increase the cost of electricity by more than 0.57 ¢/kWh.

The results of this study are summarized on Table 4-32.

Table 4-32. Direct vs. Indirect Power Trade Study Summary

	Variable Speed System	Constant Speed System (Baseline)	Remarks
Design complexity	Permits maximum efficiency blade operation	Wind energy conversion penalized by constant RPM	Variable speed captures 5.9% more wind energy
	Requires special regulator and conversion electronics	Standard regulator only	Variable speed requires more equipment
	May require power factor correction capacitors	Excitation variation provides stepless power factor control	Baseline system is more advantageous
Technical risk	Higher due to increased complexity	Low-uses existing WTS technology	Increased risk is not excessive
Drive train & structure impact	Becomes more costly due to speed variation		Quantitative assessment not made
Facilities impact	31' x 6' x 8' high electronics and transformer package	8' x 6' x 8' high transformer	Variable speed increases cost
Reliability	Significantly higher parts count	Analytically more reliable	Both systems acceptable
Maintainability	Higher maintenance cost		Increase is small
Efficiency	96% x 95.5% x 99% = 90.8%	96% x 99% = 95.0%	Electrical system only
Electrical equipment cost	$397,000	$62,000	Based on 100th unit
Cost of electricity	Adds more than 0.57¢/KWH		Variable speed system costs include CVR, added equipment & reduced efficiency

CONCLUSION: It was concluded that variable speed, or indirect power generation, when compared to constant speed direct power generation, was not cost effective.

4.2.4.1.3 Induction vs. Synchronous Generators

OBJECTIVE: A synchronous generator will be used in the MOD-2 wind turbine. However, an induction generator had the apparent advantage of simplicity, low cost and desirable operating characteristics. To assist in making the choice between the two generator types, a detailed trade study was conducted. The purpose of the study was to identify the relative performance of the two types of generators, their operational characteristics, and their cost. This effort was to provide a basis in fact to support the "common knowledge" that induction generators are simpler, cheaper, easier to control, and have better performance in wind gusts than synchronous generators. The study used 2500 kW as an output for both units. Typical induction generator characteristics are shown in Figure 4-29.

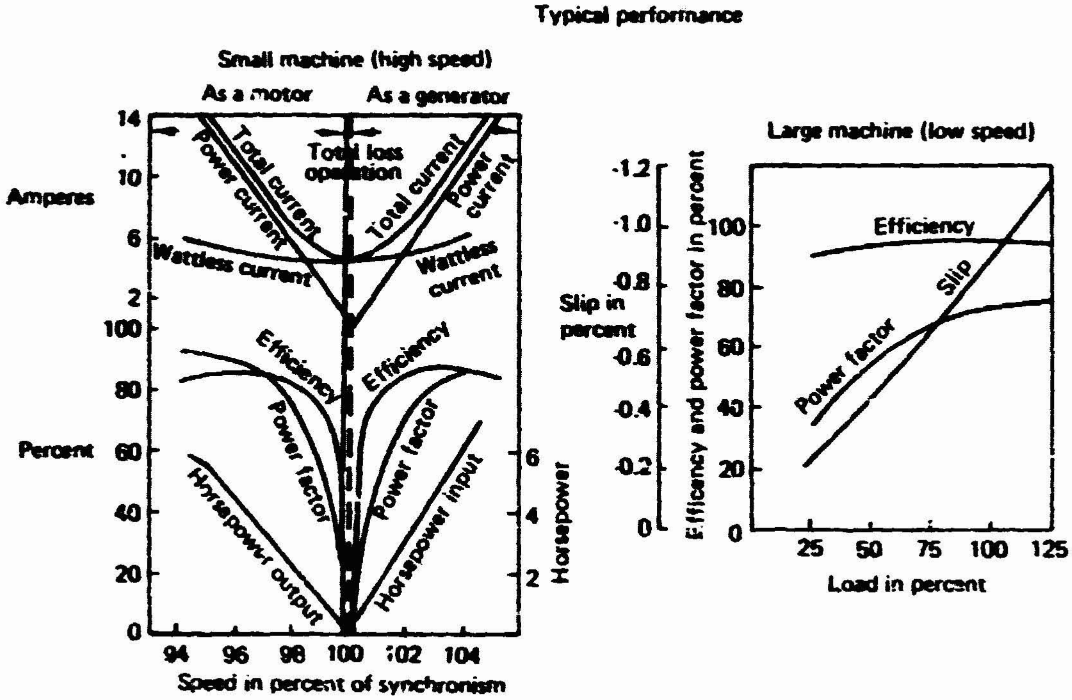


Figure 4-29. Induction Generator Characteristics - Typical Performance

RESULTS: Results of the study indicate that:

1. The induction generator system must be closely engineered to fit each WTS application because it has no effective controls to modify its operational performance as contrasted to the synchronous generator which could be controlled over a reasonable range of operating conditions. Control requirements are shown in Table 4-33.

Table 4-33. Generator Control Requirements & Characteristics

OPERATIONAL PHASE	INDUCTION GENERATOR	SYNCHRONOUS GENERATOR
PRIOR TO CONNECTION TO THE UTILITY BUS	GENERATOR HAS NO USEFUL OUTPUT VOLTAGE	GENERATOR VOLTAGE EXISTS. ADJUST GENERATOR VOLTAGE TO UTILITY BUS LEVEL. ADJUST TORQUE FOR SLIGHTLY MORE THAN SYNCHRONOUS SPEED
UPON CONNECTION	A SURGE OF REACTIVE CURRENT FLOWS INTO GENERATOR TO PROVIDE MAGNETIZATION. NO REAL POWER FLOWS	THERE IS NO REAL OR REACTIVE POWER FLOW
CHANGING POWER OUTPUT	CHANGING TORQUE INPUT CHANGES GENERATOR SPEED. REAL POWER OUTPUT CHANGE IS PROPORTIONAL TO SPEED CHANGE	CHANGING TORQUE INPUT AT CONSTANT SYNCHRONOUS SPEED CHANGES REAL POWER (KW) OUTPUT. CHANGING EXCITATION LEVEL CHANGES REACTIVE POWER (KVAR) OUTPUT
AT RATED OUTPUT	RATED NEGATIVE SLIP IS LESS THAN 1%, I.E., RPM IS LESS THAN 1818	SPEED MAINTAINED AT SYNCHRONOUS. TORQUE IS RATED VALUE

2. Operation of the induction generator is very simple while that of the synchronous generator is relatively complex. However, the controls for synchronous machines are well understood and in everyday use by the utilities.

3. The act of paralleling an induction generator is very simple while that for a synchronous machine is complex. See Table 4-34.

Table 4-34. Paralleling with the Utility

INDUCTION GENERATOR	SYNCHRONOUS GENERATOR	REMARKS
INRUSH CURRENT UPON CONTACTOR CLOSING IS ALL REACTIVE-REQUIRED FOR ROTOR MAGNETIZATION	NO POWER FLOW UPON CLOSING CONTACTOR	INRUSH CURRENT LEVEL MAY BE EXCESSIVE
POSSIBLE NEED FOR A REDUCED VOLTAGE STARTER - CONTACTOR TO LIMIT INRUSH. USE RESISTOR TYPE	REQUIRES SYNCHRONIZER.	
RPM MUST BE SLIGHTLY HIGHER THAN SYNCHRONOUS SPEED FOR CONTACTOR OPERATION	GENERATOR MUST BE EXCITED. RPM MUST BE SLIGHTLY HIGHER THAN SYNCHRONOUS. EXCITATION LEVEL MUST PROVIDE BUS VOLTAGE LEVEL FROM GENERATOR. SYNCHRONIZER MUST VERIFY VOLTAGE AND RPM AND PROVIDE IN PHASE VOLTAGE SIGNAL TO CONTACTOR	SYNCHRONOUS GEN OPERATION IS SIGNIFICANTLY MORE COMPLEX BUT WELL UNDERSTOOD AND DONE DAILY BY THE UTILITIES: STANDARD PARALLELING EQUIPMENT IS READILY AVAILABLE
RECONNECTION AFTER INTERRUPTION MUST BE DELAYED FOR FIELD DECAY	RECONNECTION REQUIRES SYNCHRONIZATION	

4. In wind gusts, both types of machines will deliver additional power until either the wind velocity declines or the rotor blade pitch is adjusted. Extreme gusts may cause the maximum torque limit of the generator to be exceeded. Excessive torque into the induction generator would result in very high slip and most likely an overspeed condition, while excessive torque into the synchronous machine would cause poles to be slipped creating high fault current conditions. In the induction generator, the maximum torque is a fixed value established by the design of the machine whereas in a synchronous generator, maximum torque may be controlled during operation by varing the excitation of the rotor. In a somewhat similar manner, the power factor of the delivered current will vary with the output of the generator. Again, the induction generator has a fixed, designed-in characteristic while the output of the synchronous machine may be varied by excitation control. See Tables 4-35 and 4-36.

Table 4-35. Power Factor Correction & Control

INDUCTION GENERATOR	SYNCHRONOUS GENERATOR	REMARKS
REQUIRES PF CORRECTION CAPACITORS AND ASSOCIATED SWITCHING AND SUPPORT HARDWARE	EXCITATION LEVEL CONTROLS PF. NO ADDITIONAL MAJOR EQUIPMENT REQUIRED.	INDUCTION GENERATORS REQUIRE ADDITIONAL EQUIPMENT
P.F. CORRECTED IN DISCREET STEPS	INFINITELY VARIABLE WITH NO ADDITIONAL EQUIPMENT	INDUCTION GENERATOR PF CONTROL IS POOR
FOR 2500 KW, 0.5 PF GEN. REQUIRES 1350 KVAR CONTINUOUSLY PLUS 500 KVAR ADDITIONAL FOR 0.98 PF at FULL LOAD	NOT REQUIRED	CAPACITOR COST IS $4 PER KVAR AT 4KV. CONTACTOR COST MUST BE ADDED.

Table 4-36. *Wind Gust Response*

INDUCTION GENERATOR	SYNCHRONOUS GENERATOR	REMARKS
SPEED PERMITTED TO VARY	SPEED MUST BE HELD CONSTANT	INDUCTION GENERATOR PERMITS RELAXED CONTROL SYSTEM REQUIREMENT
INHERENT EXCITATION COMPENSATION	REQUIRES ADDITIONAL EXCITATION CONTROL	SYNCHRONOUS GENERATOR PROVIDES MORE REACTIVE POWER (KVAR) VARIATION
FAST SYSTEM RESPONSE	SPEED OF RESPONSE DETERMINED BY CONTROLS	
HIGH DAMPING COEFFICIENT	LOWER DAMPING COEFFICIENT	HEAVY DAMPING PREFERRED
PEAK LOAD CAPABILITY DETERMINED BY SYSTEM DESIGN & OPERATION	CAPABILITY CONTROLLED BY EXCITATION LEVEL	INDUCTION SYSTEM MAY REQUIRE OVERDESIGN FOR PEAK LOAD CAPABILITY

5. Efficiency of the machines was found to be quite similar. At full load the synchronous generator has about one-half percent advantage. The difference increases with reduction in load. Impact on the cost of electricity is real but small. Efficiency characteristics are shown in Figure 4-30.

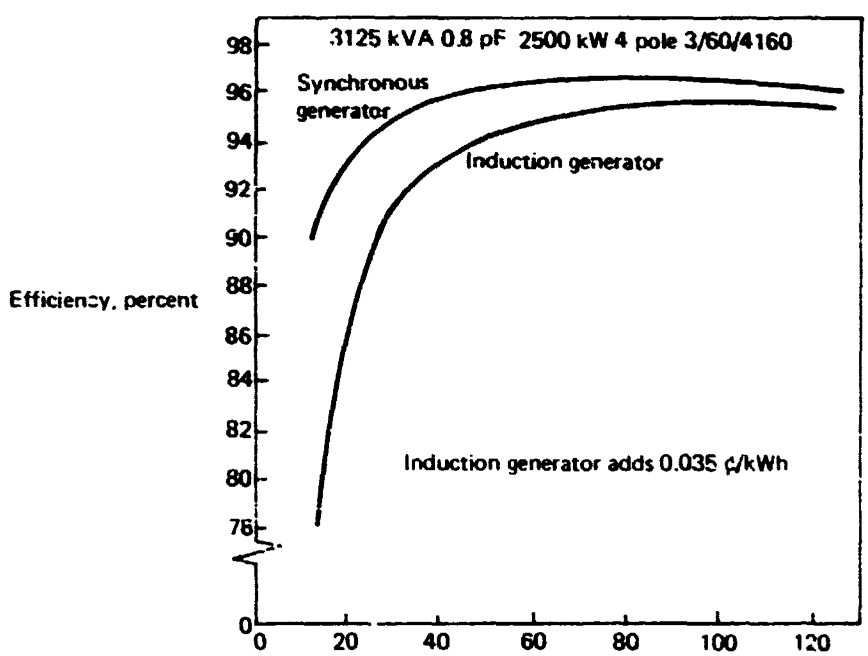
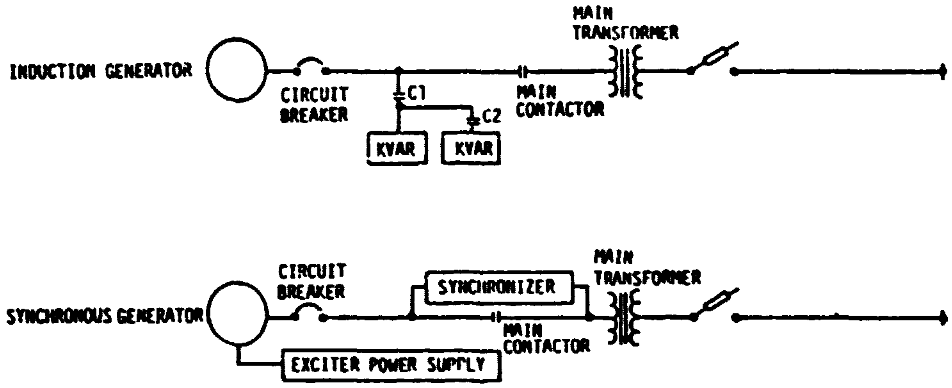


Figure 4-30. Generator Efficiency Characteristics

6 Induction Generators are less costly than synchronous, but the difference is absorbed by required associated equipment. For practical purposes the cost of the two systems is identical. One assessment of equipment cost is shown in Figure 4-31. Of major concern is the need and high cost of a reduced voltage starter to limit induction generator inrush current.



	INDUCTION GENERATOR	SYNCHRONOUS GENERATOR	REMARKS
GENERATOR	$40,000	$52,000	NOMINAL
EXCITER POWER SUPPLY & CONTROL	NOT APPLICABLE	1,225	BASLER SRBA
SYNCHRONIZER	NOT APPLICABLE	1,000	BASLER
CONTACTORS FOR CAPACITORS	8,000	NOT APPLICABLE	G.E.
CAPACITORS (1850 KVAR)	7,400	NOT APPLICABLE	WESTINGHOUSE
	$55,400	$54,225	

Figure 4-31. Equipment Differences and Costs

7. Speed controls are required for both types of units. Impact of the choice of generator on the control system will be apparent only in its software.

The results of the trade study are summarized in Table 4-37.

Table 4-37. Induction Versus Synchronous Generator Trade Study Summary

Criteria	Induction Generator	Synchronous Generator (Baseline)	Remarks
Design Complexity	Limited control capability	Full control available	Narrows induction generator applications
	Requires reactive power (KVAR) for excitation	Self-excited	Induction cannot operate independently
	Pull-out torque limited by transmission line performance	Pull-out torque governed by excitation	Induction has limited peak wind gust capability
	Does not support short circuits	Requires fast fault clearing protective devices	Both are satisfactory
	Drive speed must vary	Constant speed	Induction requires more complex dynamic analysis of drive train
	Self compensating for wind gusts	Requires additional electrical controls	Induction is simpler
Technical risk	Greater due to limited experience in WTS	Problems are known	Major induction system disadvantage
Generator efficiency	95.5%	96.0%	Light load difference is greater
Weight in nacelle	Synchronous is slightly greater		Insignificant
Control System Impact	Software changes only		No cost impact
Facility impact	Requires space for capacitors and switching equipment	No special requirement	Difference not significant
Utility interface	Limited & stepped KVAR control	Fully adjustable	Either acceptable
Transmission losses	Off optimum - poor P.F. control	Can be minimized	Induction losses are higher
Reliability	Analytically more reliable	Higher parts count	Both are satisfactory
Maintainability	Requires contactor maintenance	Requires control relay maintenance	Essentially the same
Electrical equipment cost	$55,400	$54,225	Insignificant difference
Cost of Electricity	Adds 0.035 c/KWH		

CONCLUSION: It was concluded that while the induction generator was mechanically simpler than the synchronous generator, the additional equipment required (i.e. capacitors, reduced voltage starters, etc.) made the system equally or more complex. This added equipment also caused the loss of the apparent cost and performance advantage.

As an additional effort, a study was conducted to determine if an induction generator could be used to reduce the need for a Quill Shaft in the MOD-2 WTS Drive Train. As a result of the study described below, it was concluded that the use of an induction generator with about 11% slip at full load would permit the elimination of a Quill Shaft, but that the use of a very high slip machine of this type would cause a significant increase in the cost of electricity.

The study was initiated by estimating the speed-torque characteristics of the induction generator and using them in the system simulation along with soft and stiff quill shafts. More accurate generator characteristics were established using the traditional equivalent circuit and 2500 kW machine circuit parameters from the machine manufacturer. It was found that the normal induction machine was much stiffer than anticipated. Slip at full load was 0.68% for a speed of 1812 rpm and maximum torque occurred at 1859 rpm. Performance of a high slip machine was calculated and found to be only a little better. Its full load speed was only 1837 rpm and maximum torque occurred at 1980 rpm. It was concluded that an induction generator with the characteristics as defined, and used with a stiff shaft, would produce excessive two-per-rev alternating torques. Undesirable continuous blade pitch actuation would be required to provide acceptable system performance. Under these conditions the use of an induction generator to replace a quill shaft was not cost effective (See Table 4-38 and Figure 4-32).

Table 4-38. Summary - Induction Generator Versus Quill Shaft Study

- BACKGROUND
  - QUILL SHAFT DAMPENS 2 PER REV ALTERNATING TORQUES
  - ALTERNATING TORQUE LIMIT IS ± 10% OF RATED TORQUE
  - TRANSMISSIBILITY MUST BE LIMITED TO 14%
- STUDY RESULTS
  - STANDARD INDUCTION GENERATOR: 0.7% SLIP, 117% TRANSMISSIBILITY
  - HIGH SLIP INDUCTION GENERATOR: 3.0% SLIP, 31% TRANSMISSIBILITY
  - 11% SLIP REQUIRED IN GENERATOR FOR ACCEPTABLE TRANSMISSIBILITY WITHOUT QUILL SHAFT
    - GENERATOR EFFICIENCY FALLS 9% BELOW STANDARD MACHINE
    - COE INCREASES 0.21¢/kWh (EXCLUDES ADDED GENERATOR COSTS)
- CONCLUSIONS:
  - INDUCTION GENERATOR WITH 11% SLIP AND STIFF SHAFT CAN REPLACE SYNCHRONOUS GENERATOR AND SOFT QUILL SHAFT
  - LOSS OF GENERATOR EFFICIENCY CAUSES A SIGNIFICANT INCREASE IN COE
- RECOMMENDATION
  - RETAIN SOFT QUILL SHAFT AND SYNCHRONOUS GENERATOR

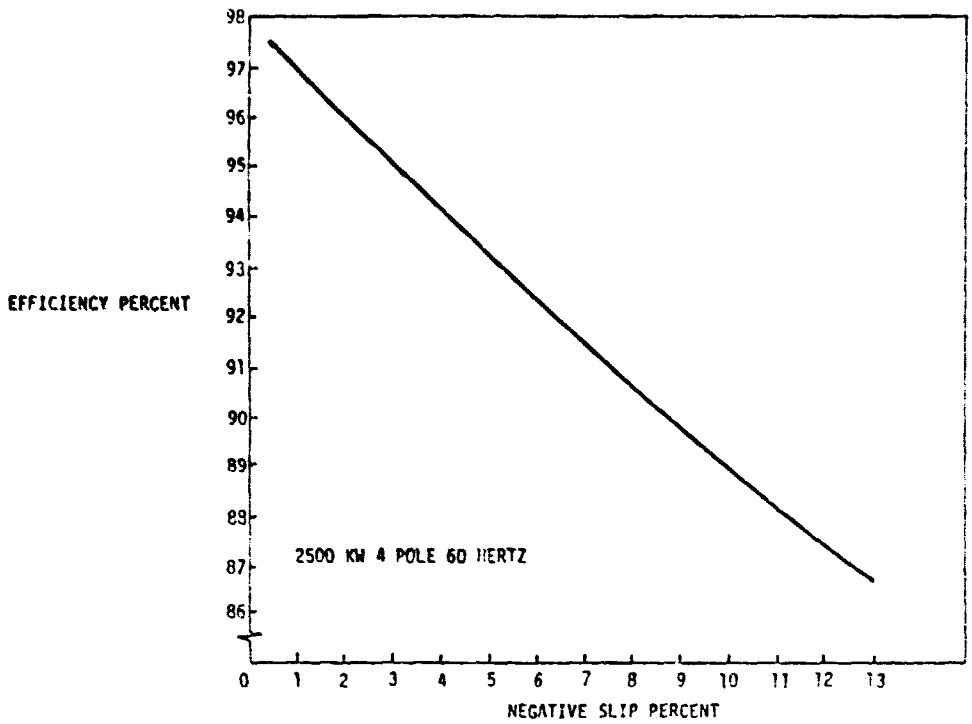


Figure 4-32. Induction Generator Full Load Efficiency as a Function of Slip at Full Load

4.2.4.1.4 Generator Speed

OBJECTIVE/RESULTS: The initially proposed configuration of the MOD-2 WTS used a 1200 rpm generator. A viable alternative is 1800 rpm. Choice of speed has an impact on the generator, high speed drive train, and the gearbox. A 2500 kW generator operating at 1800 rpm costs approximately 10% less than a 1200 rpm generator of the same capacity. The diameter of the four pole machine (1800 rpm) is slightly less than that of the six pole unit (1200 rpm). In the drive train an increase of speed at a constant 2500 kW load results in a proportional reduction in load torque. The gearbox reduction ratio increases from about 65:1 for 1200 rpm to about 100:1 for 1800 rpm; however, the gearbox cost does not change appreciably.

CONCLUSIONS: It was concluded that because there was a cost and weight saving the generator speed should be changed from 1200 rpm to 1800 rpm.

4.2.4.1.5 Generator Voltage

The concern with generator voltage is associated with the economics of transmitting the output of individual wind turbines to a common point where they are summed and then transmitting the resulting block of power to the utility tie point. Power losses in the transmission system, stability of operation, and cost of the equipment to the tie point must be considered. Electrical design considered these concerns and evolved a configuration that may be adapted to individual multi-unit installations efficiently and economically.

The MOD-2 WTS generator will provide its rated output at 4160 Volts (4.16 kV). This value was initially selected because it is the industry standard for generators rated to provide 2500 kilowatts. Changes increase the generator cost and may change its procurement lead time. Other than 4.16 kV, both 6.9 kV and 13.8 kV are identified as NEMA standard output voltages and generators can be built with either of these outputs without increasing technical risk. Use of 4.16 kV as the WTS generator voltage is highly restrictive because it cannot be used to transmit 2500 KW for any significant distance. Practical distances with it are considerably less than one mile. The 6.9 KV level is more practical but again, the range is quite short. At two miles, the line losses will be high and the high reactance of the line may cause operational stability problems. The 13.8 KV level will permit transmission over reasonable distances and is the practical upper limit for the generator output.

OBJECTIVE: A trade study was conducted to compare a system using a 4.16 KV generator and 13.8 KV output transformer to a system using a 13.8 KV generator. Elimination of the output transformer appeared attractive.

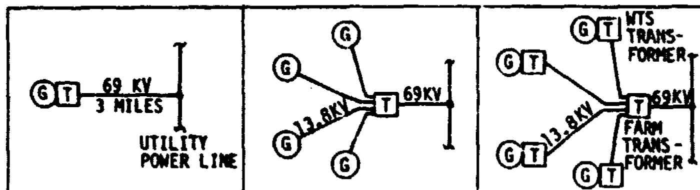
RESULTS: Information obtained from vendors confirmed that 4.16 KV was the standard voltage for a 2.5 MW generator and its cost would be about $50,250. A 13.8 KV generator of the same capacity was non-standard. It would be larger, heavier , and cost $76,380, or $26,130 more than the standard unit. Weight would increase 4550 pounds from 15,250 to 19,800. The size would increase slightly, and there would be minor size and weight growth in the nacelle and tower. The cost of the output transformer is about $27,900. Transformer costs vary as a function of KVA rating and over reasonable ranges are relatively independent of input and output voltage levels. The cost of the applicable electrical equipment is identified on Figure 4-33. The cost of the 13.8 KV generator electrical equipment is only $1770 less than that of the standard, and the costs associated with the enlarged nacelle and tower are not listed.

CONCLUSION: Use of a wind turbine electrical power system configuration that contains a power output transformer at the foot of the tower provides the flexibility required for single unit and farm installations. The output voltage of the transformer may be specified to accomodate the WTS usage. In the baseline design a 69KV output voltage is specified to assure delivery of WTS power to a utility transmission line rather than a distribution line. For other single unit installations the output voltage can be specified to match the power lines at or near the site. For multiple unit installations or farms the WTS output voltage may be optimized for power transmission within the farm and the farm output voltage set to match the utility lines. As noted below, this can be done at no cost disadvantage.

4.2.4.1.6 High Altitude Operation

OBJECTIVE: The purpose of this study was to evaluate high altitude operation of a generator.

RESULTS: The capacity of a generator is limited by its maximum permissible temperature. This value is a function of the class of electrical insulation used in the machine's construction. The power output rating of a generator is based on it reaching this limiting temperature with cooling air of a defined maximum temperature at a given maximum altitude. A constant volume of air flow is assumed.



WIND TURBINE PARAMETERS			
CONFIGURATION NOMENCLATURE	BASELINE	HIGH VOLTAGE GENERATION	LOW VOLTAGE GENERATION
GENERATOR VOLTAGE	4.16 KV	13.8 KV	4.16 KV
WTS TRANSFORMER RATING	2.5 MW, 0.8 PF	NOT APPLICABLE	2.5 MW, 0.8 PF
COST PER GENERATOR	\$50.25K	\$76.38K	\$50.25K
COST PER TRANSFORMER	\$27.90K	NOT APPLICABLE	\$27.90K
TOTAL COST PER WIND TURBINE FOR GENERATOR AND TRANSFORMER	\$78.15K	\$76.38K	\$78.15K
FARM PARAMETERS			
WIND TURBINES PER FARM	1	4	4
FARM TRANSFORMER RATING	NOT APPLICABLE	10 MW 0.8 PF	10 MW 0.8 PF
FARM TRANSFORMER COST	NONE	\$63.24K	\$63.24K
TRANSMISSION LINE VOLTAGE WITHIN FARM	NOT APPLICABLE	13.8KV	13.8KV
TRANSMISSION LINE LENGTH WITHIN FARM	NONE	3.236 MILES	3.236 MILES
TRANSMISSION LINE COST WITHIN FARM	NONE	\$24.27K	\$24.27K
TOTAL COST OF GENERATORS, TRANSFORMERS AND TRANSMISSION LINES FOR FARM	\$78.15K	\$393.03K	\$400.11
COST PER WIND TURBINE FOR GENERATOR, TRANSFORMERS AND TRANSMISSION LINES	\$78.15K	\$98.26K	\$100.03K

G = GENERATOR
T = TRANSFORMER

Figure 4-33. 4.16kV Versus 13.8kV Generators Cost Analysis

For commercial machines, the National Electrical Manufacturers Association has standardized the rating conditions as 50 degrees F (10°C) to 104 degrees F (40°C), at altitudes up to 3300 feet (1000 meters). Operation of a standard commercial machine at conditions beyond these limits requires that the machines' power output be derated.

For generators of the MOD-2 size, the following derating factors may be used for initial approximations. Final rating data must be provided by the generator manufacturer.

Derating for Ambient Temperature

- 10 KW per degree F above 104°F or
- 18 KW per degree C above 40°C

Derating for Altitude

- 45 KW for each 1000 feet above 3300 feet or
- 15 KW for each 100 meters above 1000 meters

Based on these two factors, a generator rated 2500 KW at the standard conditions of 3300 feet altitude and 105°F ambient temperature would provide only 2183 KW at 7000 feet and 120°F. Using 120°F at 7000 feet fails to recognize the natural environment. The design temperature limits will be 120°F maximum at sea level and decrease linearly to 95°F at 7000 ft. Under these more realistic conditions for temperature at altitude the standard machine would have a rated capacity of approximately 2333 kW at 7000 feet.

Data provided by a generator vendor indicated that the required 2500 kW rated output can be provided at 7000 feet if the machine size was slightly increased. Cost increase is negligibly small.

CONCLUSION: Based on the above information, it was concluded that a single generator design should be employed for all WTS installations from sea level to 7000 feet above sea level.

4.2.4.2 Deletion of the Diesel Driven Generator

OBJECTIVES: The baseline electrical power system proposed for the MOD-2 wind turbine contained a diesel driven generator. The diesel driven generator (DG) functions to provide long duration stand-by electrical power to WTS auxiliary equipment when power is not available from the WTS rotor driven generator or from the utility. This loss of normal power could occur as a result of utility outages or faults on the transmission line connecting the WTS to the utility infinite bus. The DG is an item of emergency equipment that can be automatically started immediately after loss of the utility and run for as long as needed. This study evaluated the need for a diesel driven generator.

RESULTS: A survey of the requirements of the WTS auxiliary equipment has not identified any item of equipment that requires long duration standby power. It was concluded that the fail-safe design employed in each subsystem would cause the controlled shut down and securing of the WTS in the case of total loss of electrical power. Continued absence of electrical power will not degrade the equipment. It is expected that some equipment will cool below its minimum operating temperature. After the utility connection is restored, operation of the power generation system will have to be delayed while this temperature sensitive equipment is heated to acceptable temperature limits. This constraint appears reasonable.

Deletion of the DG from the electrical system will not eliminate a standby power capability. Energy will continue to be available from the Uninterruptable Power Supply (UPS). As presently sized, this supply will provide 48 VDC from 50 amp-hour batteries. Loads on this supply are such that based on an 8 hour rating, the battery can support the continuous load for over 8 hours. If required, the size of this battery can be increased at low cost and without any increase in the required maintenance time. To date, there is no indication the battery size will be required to change from the 50 AH size.

Elimination of the DG was encouraged by the utilities. During Utility Technical Interchange meetings concern was expressed about the need for the DG. They pointed out that it would require additional maintenance and would require periodic inspection by a fire inspector. They found this unattractive and encouraged the development of a design that would not require a DG.

CONCLUSION: Based on the above assessment the diesel driven generator was deleted from the MOD-2 design.

4.2.4.3 Location of the Generator Circuit Breaker

OBJECT: This study evaluates the location of the generator circuit breakers.

RESULTS: The generator circuit breaker (GCB) functions to prevent the generator from loading the drive train to a torque level that exceeds the capacity of the gear box. Excessive generator loads are possible during some unusually high wind gusts and during some electrical fault situations. Mounting the GCB in the nacelle puts it as close as practical to the generator terminals, which allows it to interrupt current to faults in the nacelle downstream of the GCB, allows it to respond very rapidly to excessive currents, provides maximum assurance that faults can be isolated, permits earliest detection of generator faults, and reduces the number of circuits that must be carried across the yaw bearing.

The 4.16 KV GCB functions to interrupt the 4.16 KV circuit from the generator under all possible current levels. In doing this, it reduces the power output of the generator and the driving torque it requires to essentially zero. Operation of the GCB is controlled by a standard complement of protective relays which responds to the 4.16 KV power and current. These relays, which operate on signals from current and potential transformers, are adjusted to provide the GCB with a trip signal when the monitored parameters indicate that excessive torque levels are approached.

The alternative to mounting the GCB in the nacelle is to mount it on the ground at the foot of the tower. This location infers that the generator differential protection zone would include the path around the yaw bearing and the 4.16 KV power cables that extend from the nacelle to the foot of the tower and then to the GCB equipment enclosure. A part of this path may be underground. In the alternative GCB location, fault currents due to cable failure would be reduced more slowly by removal of generator excitation rather than by the direct opening of the 4.16 KV power feeders at the generator terminals. An equipment enclosure for electrical power apparatus would continue to be required in the nacelle. It would house the generator excitation control apparatus, potential and current instrumentation transformers, and a number of protective relays.

CONCLUSION: The best location of the GCB is in the nacelle.

4.2.4.4 Wiring Transfer Across the Yaw Bearing

OBJECTIVE: A requirement exists to transfer power, control, and signal electrical circuits from the rotatable nacelle to the stationary tower. Initially it was proposed to do this with flexible cables and to limit nacelle rotation to ± 360 degrees from an index position. As a result of the study described below the design was changed to slip rings rather than flexible cables,and the limits applied to nacelle rotation were eliminated.

RESULTS: Detailed preliminary requirements defined a total of 64 individual circuit paths between the nacelle and tower. Most critical design item was a 350 MCM path for the flow of 2500 kW of 4160 volt, three phase 60 Hertz power. Individual discussions with six different cable manufacturers provided a unanimous recommendation that the high voltage power cables not be twisted. Using the full height of the tower, the cable twist would be at least plus and minus two degrees per foot of cable length. This would cause the cable electrical insulation to fail after relatively few operational cycles. This information plus the fact that the twisted cable would claim the space inside the tower allocated to the hoist caused twisted cables to be dropped from further consideration.

Cable manufacturers indicated that controlled and limited bending of the high voltage power cables was acceptable. Life would be limited with no prediction available for duration. Preliminary designs that exploited the bending concept indicated that the cable must lay in a toroidal tray,thus limiting access to the nacelle to an 80 inch diameter space in the center of the tower. The movable part of the cable would weigh in excess of a thousand pounds and would have to be attached to a skid plate and a push plate. Not analyzed was the impact of the magnetic field of the required coil configuration. Concern existed about impact of the magnetic field on power line impedance, mechanical properties during electrical faults, and of induction heating of local structure. When compared with slip rings the bending cable configurations were unattractive.

Slip rings are being used successfully in the MOD-0 and MOD-0A wind turbines and are in the MOD-1 design. Technical risk for slip rings is low and reliability has been demonstrated. Cost for slip rings is known. Fixed price quotes, based on MOD-1 experience, are available.

CONCLUSION: Based on the above assessment, the baseline design was revised to delete flexible cables across the yaw bearing and to carry all circuits through slip rings. This study is summarized in Table 4-39.

4.2.5 Nacelle

This section shows the results of trade studies which evaluated alternative nacelle configurations.

4.2.5.1 Electric Vs. Hydraulic Yaw Drive

OBJECTIVE: The subject trade study was conducted, comparing the baseline hydraulic rotor driven yaw drive system (including caliper disk brakes) with an electric motor and worm-gear drive system.

RESULTS/CONCLUSION: It was concluded that the hydraulic motor driven yaw drive system offers the following advantages over the electric motor and worm-gear drive:

Table 4-39. Summary - Circuit Transfer Across the Yaw Bearing

	Twisted cable (Vertical)	Coiled cable (Horizontal)	Slip ring assembly
Design complexity	Requires cable supports and twist controller at 50 ft intervals	Requires cable tray, cable support push plate, and cable termination clamps	Derived from existing MOD-1 design
	Twisted high voltage cable not recommended by cable vendors	Cable to be pushed will weigh over 1,000 lbs	
	Uses space preferred for elevator	Limits Nacelle access hole to 60-in diameter	Provides adequate and convenient Nacelle access
Test program	Requires development and qualification testing	Requires qualification testing	No testing required
Technical risk	Very high	Moderate	Relatively low
Weight impact	High	Highest	Relatively low
Maintainability	High - expect frequent cable replacement	Medium - inspect and lube skid plate	Low - detailed procedure available
Reliability	Low	Medium	High - experience data available
Cost	Considered highest - difficult to predict with confidence due to design risk and impact on elevator	Medium - uses the most cable and mechanical equipment	Lowest - fixed price quote available

1. Lower cost ($103,000 less for 100th unit).

2. Smaller size, allowing smaller mounting plate and structure.

3. Better location and accessibility for maintenance of prime mover (motor-pump).

4. Will overrun safely without special precautions (slip clutches).

5. Higher stall torque - also may be stalled without damage.

The trade study was terminated and no further consideration was given to the electric motor/worm-gear drive yaw system.

4.2.5.2 Nacelle Configuration Studies

Nacelle configuration is impacted by every major wind turbine element. Successful integration of the nacelle requirements for support, access, hoisting, maintenance, etc., appears to be a dominant factor in the successful design and operation of a wind turbine system. As a result, 47 major layouts were prepared during the evolution of the current MOD-2 nacelle configuration. Some of the more important trades associated with this evolution are reported below.

OBJECTIVE: To develop a nacelle configuration that most efficiently integrates the requirements discussed above.

RESULTS: The five concepts selected as baseline at different points in the program are briefly summarized in Table 4-40. General arrangement views are shown in Figures 4-34 through 4-39. Configuration 107 has been adopted for the MOD-2 WTS.

Table 4-40. Baseline Nacelle Configuration Data

Width-Height-Length	Gearbox	Weight - Lbs
153 x 153 x 663	Parallel Shaft - Wide	30,675
208 x 168 x 580	Parallel Shaft - Tall	45,041
139 x 139 x 712	Parallel Shaft - Wide	67,472
116 x 109 x 420	Parallel Shaft - Wide	30,362
132 x 112 x 442	Epicyclic	32,836

Three basic nacelle structural concepts were evaluated as shown in Table 4-41.

Table 4-41. Basic Nacelle Concepts

Concept	Weight - Lbs.	Comparative Cost
Truss	30,675	1.00
Heavy Bed Beam	32,875	2.06
Semi-monocoque	31,828	2.15

CONCLUSION: The truss concept was adopted because of lowest weight and cost.

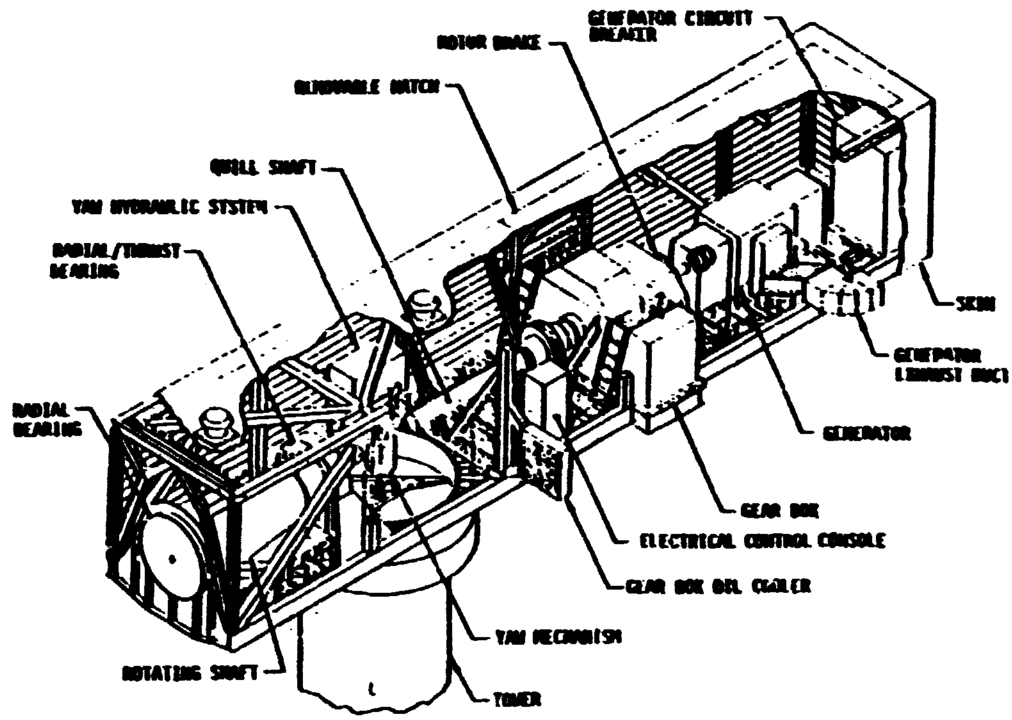


Figure 4-34. Nacelle Arrangement MOD-2-103

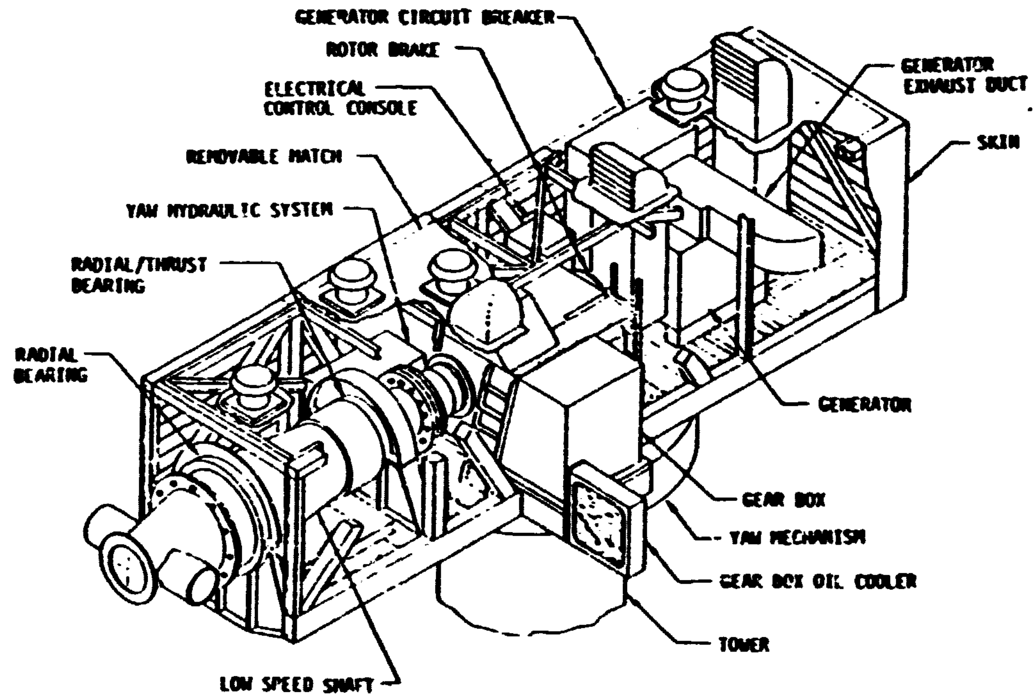


Figure 4-35. Nacelle Arrangement MOD-2-104

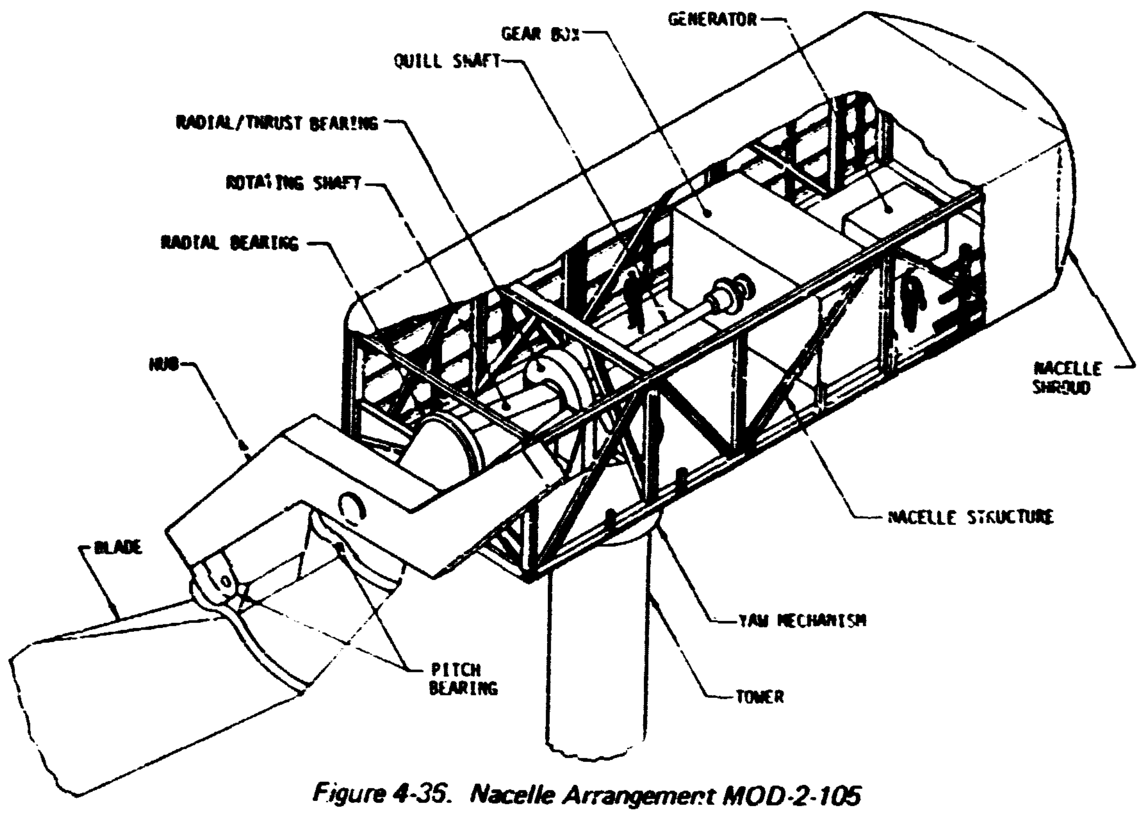


Figure 4-35. Nacelle Arrangement MOD-2-105

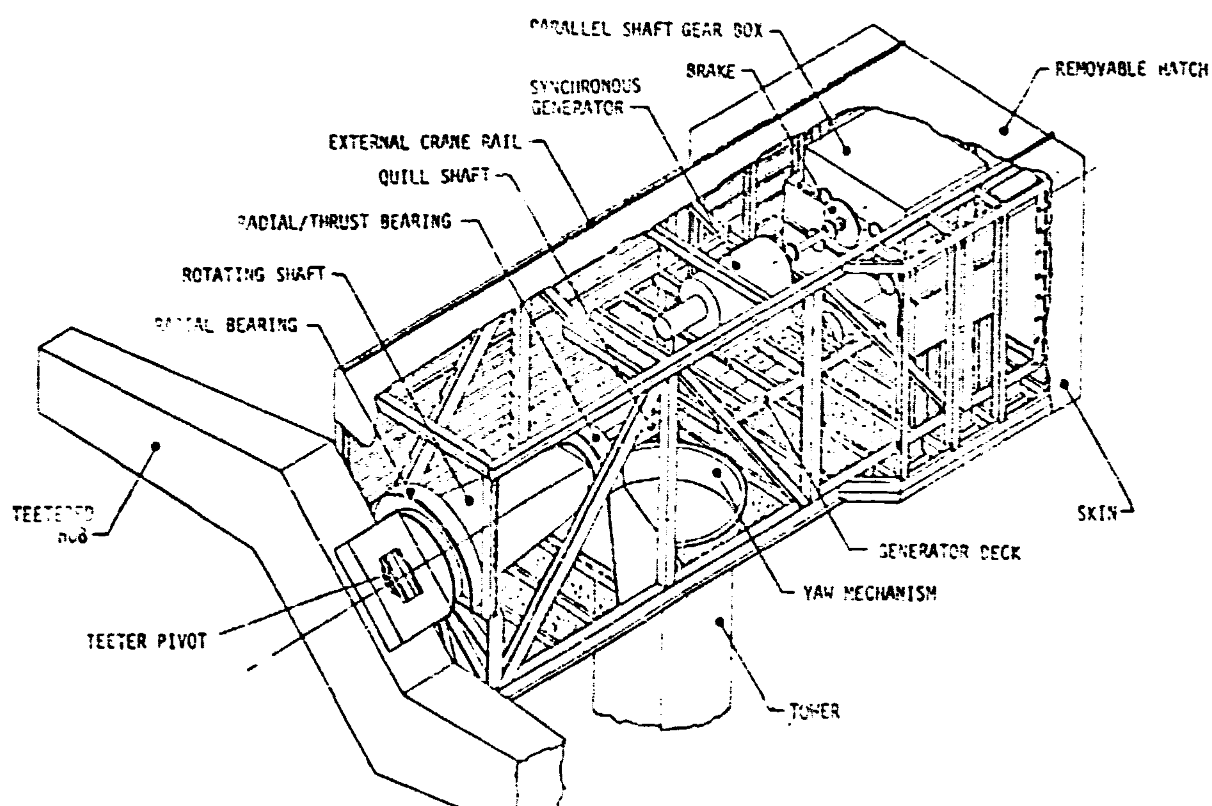


Figure 4-37. Nacelle Arrangement MOD-2 106

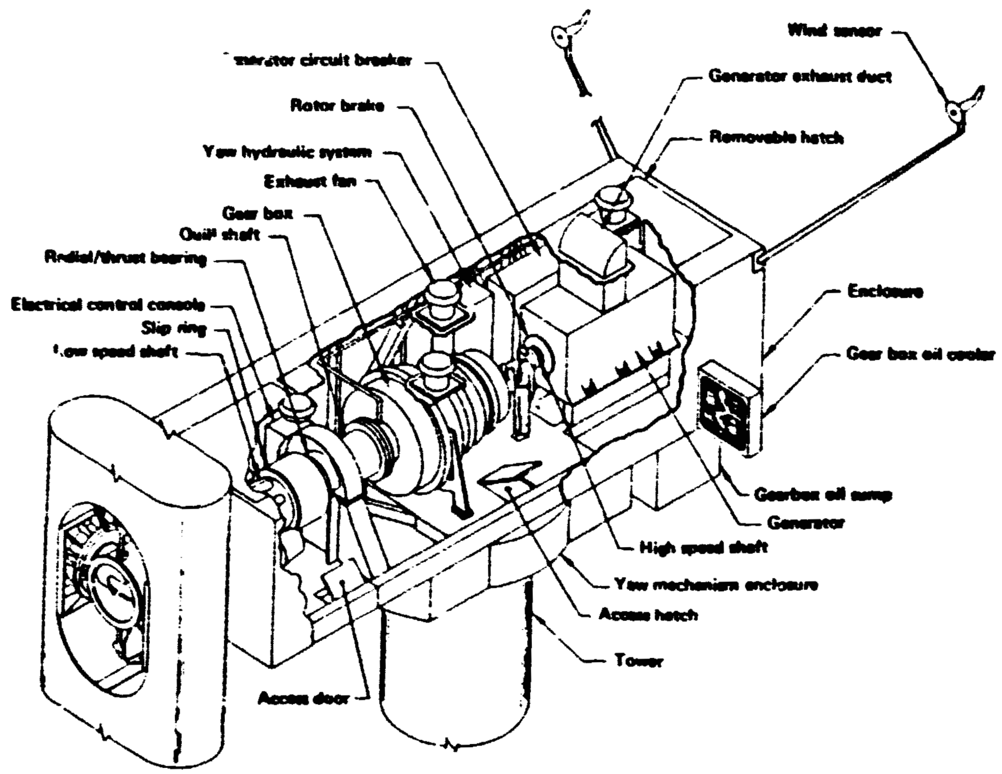


Figure 4-38. Nacelle Arrangement MOD-2-107

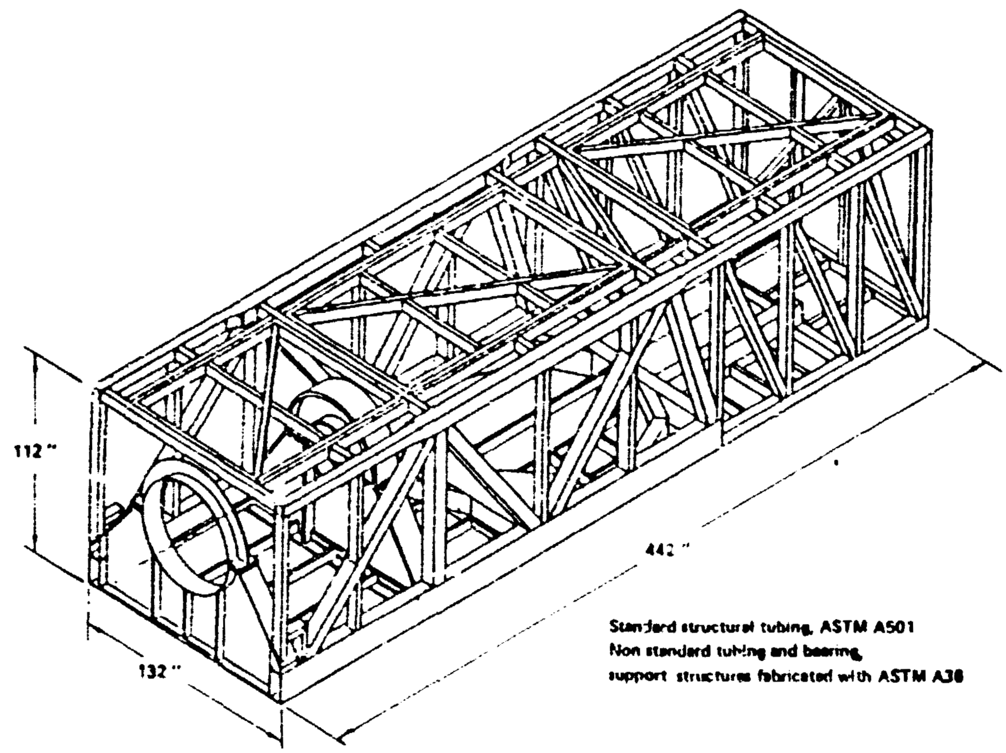


Figure 4-39. Nacelle Structure MOD-2-107

Nacelle Enclosure Studies: Several enclosure concepts were evaluated as shown in Table 4-42.

Table 4-42. Comparison of Nacelle Enclosure Concepts

Skin Concept	Weight - Lbs	Comparative cost
Fiberglass - Integral Stiffener	13,355	1.72
Fiberglass - Bonded Stiffener	12,075	1.58
Corrugated Steel Skin	12,556	1.00
Sheet Steel - Brake Formed Stringers	12,225	1.30

CONCLUSION: Corrugated steel skin was adopted on the basis of lowest hardware cost.

Rotor Bearing Support Studies: Several methods of supporting the rotor bearing system were evaluated for structural weight effect on the nacelle as shown in Table 4-43.

Table 4-43. Comparison of Nacelle/Rotor Bearings Support Concepts

Layout	Concept	Weignt - Lbs
37	Single Forward Bearing	13,028
38	Live Shaft	7,856
39	Fixed Shaft	7,913

CONCLUSION: The live shaft system was adopted on the basis of lowest weight.

Nacelle Joint Study: Steel fabricator provided comparative costs for the following types of structural joints (Figure 4-40):

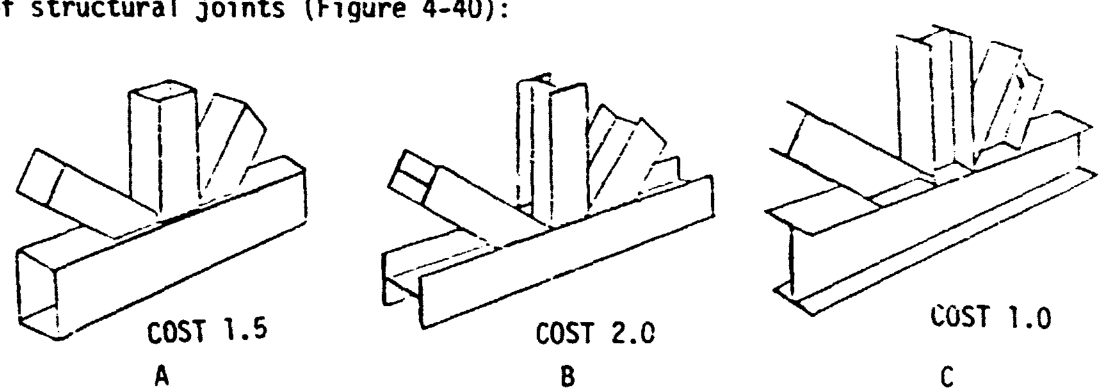


Figure 4-40. Types of Structural Joints

CONCLUSION: The "C" type structural joint concept was selected and is being incorporated into the detail design.

4.2.5.3 Yaw Stiffness Requirements

The current hydraulic yaw drive system is described in section 3.2. This study provided the basis for the definition of stiffness and brake torque requirements.

OBJECTIVE: The objective of this study was to determine a cost-effective yaw mechanism configuration having adequate stiffness to ensure optimum efficiency of the WTS.

RESULTS: Analysis of the operating conditions indicated that the torques applied to the yaw mechanism by the rotor had a predominant two per rev frequency content, and a significant non-zero mean bias for both upwind and downwind conditions. The system torsional frequency requirement was established to avoid the predominant rotor forcing frequency. To assure proper placement of the coupled torsional frequency, requirements were established for various uncoupled system frequencies as shown in Table 4-44.

Table 4-44. Yaw Frequency Design Requirements

ITEM	REQUIREMENT
Tower/Yaw System	Fundamental coupled torsional frequency to be ≥ 2.3 per rev and free of resonance with integers of rotor speed
Rigid Yaw System	Uncoupled nacelle/mast nat. frequency to be > 7.0 per rev.
Rigid Nacelle & Tower (Max. brake torque)	Yaw mech. torsional frequency to be > 15.0 per rev.
Rigid Nacelle & Yaw System	Tower torsional frequency to be > 5.0 per rev.

The yaw system brake requirement was established to hold the rotor without slipping under all operating conditions to provide structural integrity across the yaw interface and attain the yaw frequency requirement. In addition, the brakes hold the rotor without slipping in winds in excess of cut-out speed, in all rotor and nacelle orientations, up to steady winds of 81 mph. Above this wind speed, the nacelle will slowly yaw (at a rate less than 3 rpm) to a downwind position. This protects the yaw drive motor from damage, while keeping blade loads to acceptable levels. The predicted oscillating torques indicated the need for a yaw drive brake, applied while yawing, to assure that yaw oscillations are adequately damped.

CONCLUSION: The MOD-2 yaw system structural stiffness requirements were defined. Analysis were conducted which showed that the MOD-2 design, with 6 brakes providing a torque of 432,000 ft-lb, met the stiffness requirement and did not adversely affect other structural subsystems.

4.2.6 Tower

This section reports the results of trade studies conducted to evaluate alternative tower designs.

4.2.6.1 Soft vs Stiff Tower

The soft, monocoque shell tower of the WTS configuration, described in sections 3.0 and 3.2.4, was selected on the basis of this trade study. Although the study was performed for a braced tubular tower and a downwind rotor, the conclusions drawn from the results of the study are equally applicable to the pure monocoque configuration and an upwind rotor shown in the above referenced sections.

OBJECTIVE: The objective of this study was to compare the technical and economic feasibility of the baseline soft tower configuration with that of a stiff tower by examining the relative cost of electricity and the technical risks associated with each concept. The study was conducted assuming a downwind rotor with baseline rotor weight of 177,000 lb and total elevated system weight, tower excluded, of 356,000 lb.

RESULTS: Stiffness Requirements - The distinction between soft tower and stiff tower is made on the basis of combined tower/system fundamental frequency relative to that of the principal alternating hub forces which occur at two cycles per revolution for the two-bladed rotor. The permissible frequency ranges established for the fundamental bending mode were 1.3 to 1.5 cycles per revolution for the soft tower, and 3.2 to 3.8 cycles per revolution for the stiff tower. These ranges were selected to reduce dynamic amplification of the principal 2 per revolution excitation force to acceptable limits, while still avoiding other undesirable frequencies at integral numbers of cycles per revolution.

Tower Configurations - The tower configurations compared in the study are shown in Figure 4-41. A monocoque shell design was selected for the soft tower because of the slenderness required to achieve the low fundamental bending frequency with minimum tower shadow effect on the downwind rotor. The truss configuration was selected for the stiff tower design as the most efficient way to achieve the much greater stiffness required without causing excessive wind blockage. The truss tower configuration is of welded tubular construction to improve air flow and minimize tower shadow effects. Both towers are designed by the stiffness requirements and both towers meet all static and fatigue strength requirements.

Tower Shadow Effect - Blade flapwise bending moments for a complete rotor cycle at maximum operating wind speed of 45 mph at hub height were obtained from the MOSTAS computer program for both soft and stiff towers. The MOSTAS results, Figure 4-42, show that the alternating blade loads are 27% higher for the stiff tower due to increased wind blockage. This results in a 7% increase in rotor weight to satisfy the increased fatigue strength requirement, which in turn increases the stiff tower weight by approximately 6% to maintain the same bending and torsional frequencies.

Start-up/Shut-down Load Conditions - During shutdown the 2 per revolution rotor load frequency will pass through the soft tower fundamental frequency. A transient analysis was made to simulate this condition as follows:

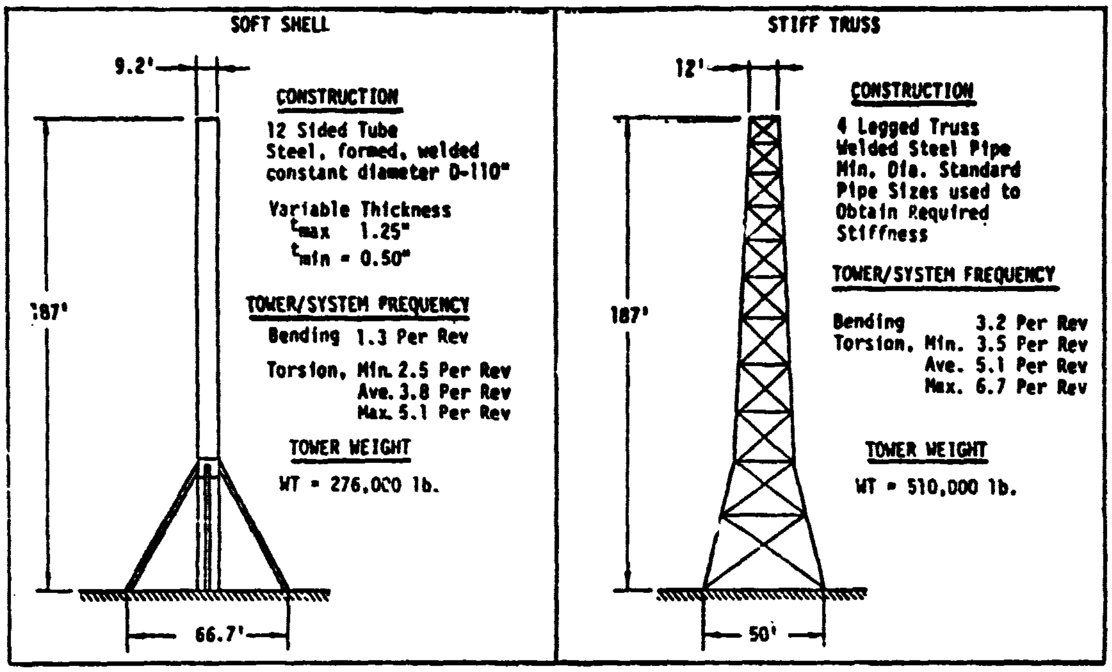


Figure 4-41. Tower Configurations

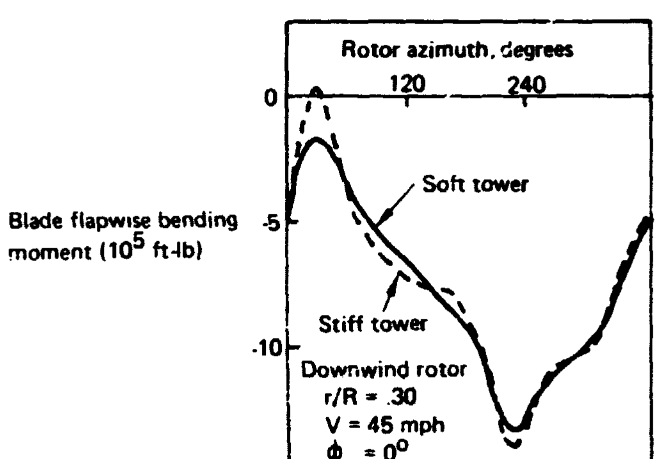


Figure 4-42. Tower Shadow Blade Load Effect

(1) Rotor loads were assumed decoupled from tower response
(2) Full RPM rotor loads were computed by MOSTAB for 45 mph wind at 20 degrees yaw
(3) Rotor loads were assumed proportional to RPM during shutdown
(4) Shutdown from full RPM was accomplished in 15 rotor cycles
(5) Total linear system damping of 2.5% was assumed.

An exact solution of the above system for uniform deceleration resulted in a maximum tower deflection of 2.6 inches, which is only 3 percent of the static limit load tower deflection. Therefore, shutdown transient loads are considered negligible.

The high speed startup problem is similar to the shutdown problem caused by 2 per revolution rotor loads crossing the soft tower resonant frequency. Normal startup at 45 mph with one degree per second pitch rate will occur in approximately 15 rotor cycles. Therefore, maximum tower deflection will be approximately the same as for the assumed shutdown case (2.6 inches) and the resulting transient tower loads can be assumed negligible.

Low speed startup under certain wind conditions might conceivably cause continuous idling with 2 per revolution rotor loads at or near tower resonance, causing large tower oscillations to build up. This potential problem will be solved by incorporating provisions in the startup control system to prevent such continuous idling, and to limit startup transient loads to not greater than those occuring during normal high speed startup.

A wind tunnel test was conducted on a 1/20 dynamically scaled model of the soft tower configuration. This test proved the feasibility of the soft tower because it verified the above analytical results. No significant tower vibration was noted during start-up or shut-down.

CONCLUSIONS: The results of the trade study, summarized in Table 4-45, indicate that the baseline soft tower is technically and economically more feasible than the stiff tower.

Table 4-45. Trade Study Summary - Soft Versus Stiff Tower

Concept	Soft shell	Stiff truss	Remarks
Rotor weight	177,000 lb	190,000 lb	Difference due to tower shadow effect
Tower load amplification	1.0	1.8	Relative load magnitudes affecting fatigue deisgn of tower
Tower weight	276,000 lb	510,000 lb	Increased weight due to stiffness requirement
Design complexity	44 parts Startup control	196 parts	Startup control to prevent continuous idling at soft tower resonance
Rotor cost	$268,000	$288,000	Difference due to increased rotor weight
Tower cost	$188,000	$578,000	
Transportation	$8,000	$15,000	Includes cost of additional rotor weight
Erection cost	$22,000	$66,000	
Annual energy out	1.000	0.99	Relative values based on greater tower shadow effect for truss
Technical risk	None	Soft foundation Weight growth	
Cost of electricity	Baseline	Adds 1 ¢/kWh	Assumed same O & M costs

4.2.6.2 Soft vs Soft-soft Tower

A brief investigation was made into the feasibility of a soft-soft tower with a fundamental system frequency of 0.8 cycles per revolution or less. Such a tower design would permit greater attentuation of one per revolution alternating loads caused by any dynamic imbalance in the rotor. It was found that to meet both strength and flexibility requirements for this concept the maximum tower diameter was 8 feet or less, depending on rotor speed and total elevated weight. For the baseline rotor speed of 17.5 rpm the maximum tower diameter was 7 feet. Such small tower diameters were not considered practical for use with the desired internal lift and power lead configuration; therefore, the soft-soft tower concept was rejected.

4.2.6.3 Braced vs Conical Base

Transferring loads from the tower to the foundation requires a larger base than that provided by the 120 inch tower diameter. The current tower configuration utilizes a conical base to increase the tower diameter to 250 inches. This trade study provided the basis for the selection of this configuration.

OBJECTIVE: The objective of this study was to compare tower weight, foundation volume, and technical risks of conical based towers with braced towers. Three braced tower configurations with rotor tip to ground clearances of 50 feet, 37.5 feet and 25 feet, and three conical based tower configurations with these same clearances were designed. These configurations are shown in Figure 4-43.

RESULTS: The technical risks associated with the braced tower configuration are much greater than those associated with the conical based tower configuration. The braced tower is structurally indeterminate and its foundation configuration makes it susceptable to differential settlement. These two conditions make prediction of the loads in the braces and foundation uncertain. The braces cause the base plate shear reaction of the braced tower concepts to be substantially higher than the lateral input loads applied above the bracing. However, the base plate shear reaction is not amplified with the conical based tower configurations (Figure 4-44).

Conical based towers are widely used in standard practice for structures such as water towers and chimneys which are designed to resist large overturning moments.

The resulting tower weights and foundation concrete volumes for each of the study configurations are plotted in Figure 4-45. The conical base tower weight decreased linearly with a decrease in rotor tip to ground clearance. The braced tower weight decrease was not linear with rotor tip to ground clearance, and its rate of decrease was less than the conical based configurations. This was a result of the unfavorable geometry of the braces at the lower rotor tip to ground clearances. The foundation concrete volume required for the conical base configurations decreased linearly with decreasing rotor tip to ground clearance while the foundation concrete volume required for the braced configurations remained constant. This was again due to the unfavorable geometry of the braces at the lower clearances. With a 25 foot rotor tip to ground clearance both the tower weight and the concrete volume are less for the conical base configurations.

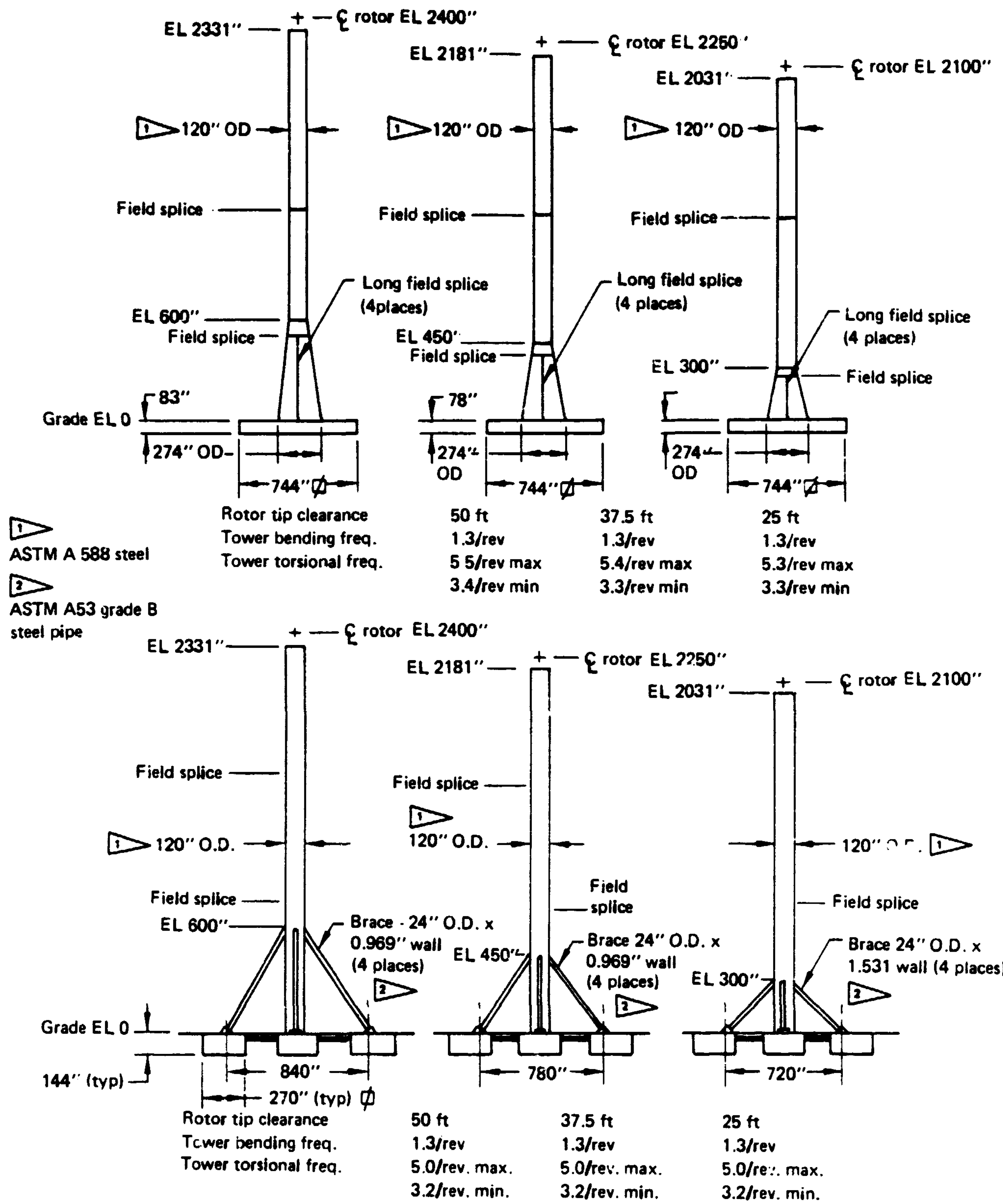


Figure 4-43. Study Configurations

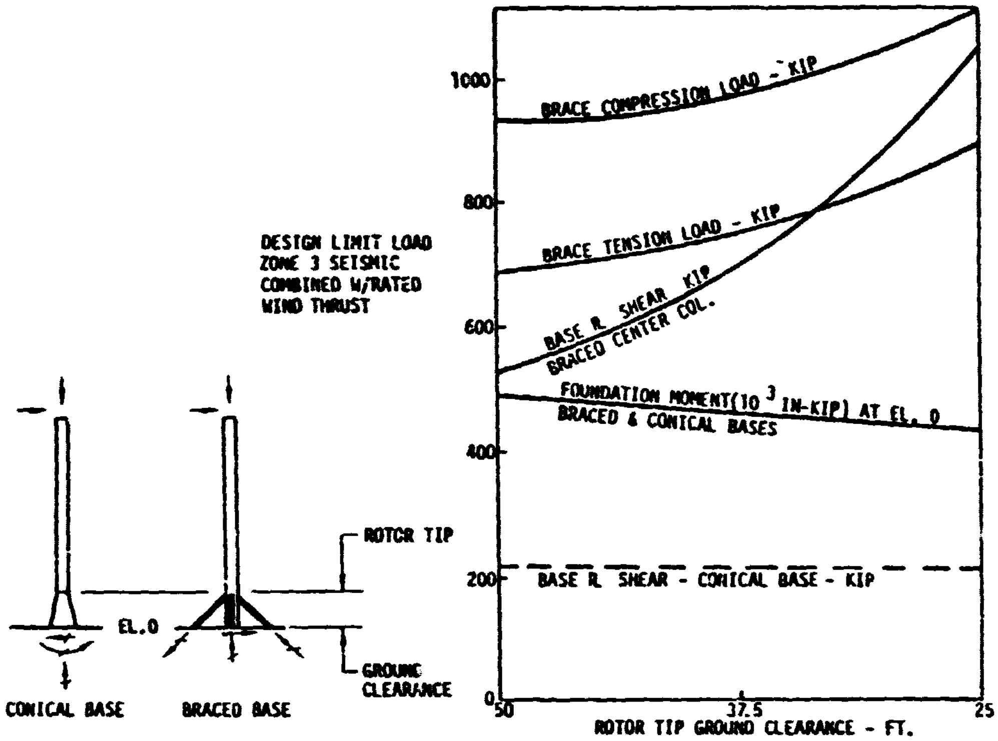


Figure 4-44. Design Loads

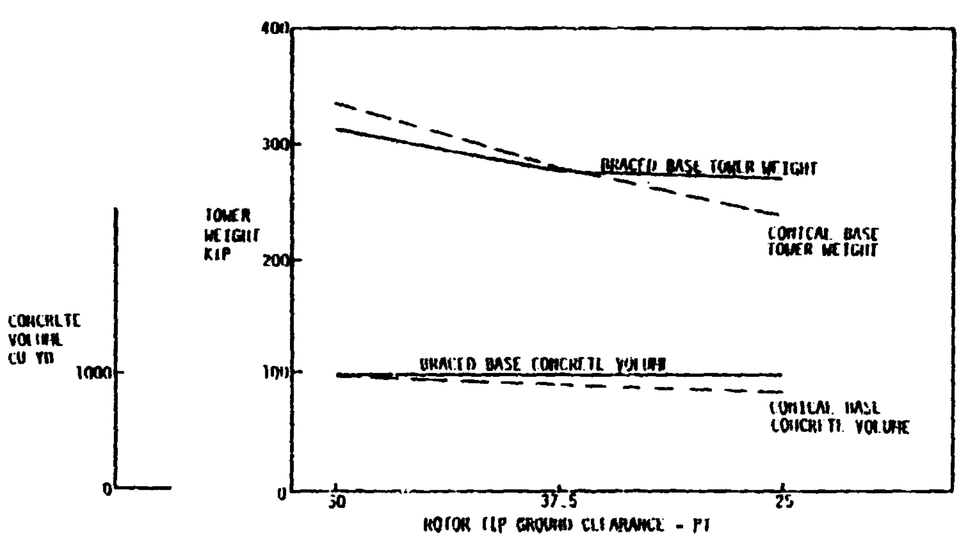


Figure 4-45. Tower Weight & Foundation Concrete Volume

C-2

With a 37.5 foot rotor tip to ground clearance both tower configurations weighed approximately the same but the foundation for the conical base configuration required less concrete. With a 50 foot rotor tip to ground clearance the braced tower was lighter than the conical based tower and both configurations required the same foundation concrete volume.

CONCLUSIONS: Based on results of this study the conical based tower configuration was considered best. The primary reason was the greater technical risks associated with the braced tower configuration. Only at the 50 foot rotor tip to ground clearance was the conical base tower found to be heavier than the braced tower. It was concluded that even for this configuration the technical risks, the complexity of the foundation, and the uncertainty of the loads outweighed the benefit of a weight savings in the tower.

4.2.6.4 Transition Section

The selection of the conical base tower required the design of a cone to a cylinder transition section. The initial tower configuration had an abrupt transition section. A circular T-ring weighing approximately 2700 lbs. and 72 gussets weighing approximately 900 lbs. were required due to local stresses produced by this abrupt transition. The selected tower configuration has a hyperbolic shaped transition section which eliminates the abrupt cone to cylinder transition. This greatly reduced the local discontinuity stresses and eliminated the need for the ring and gussets, (Figure 4-46).

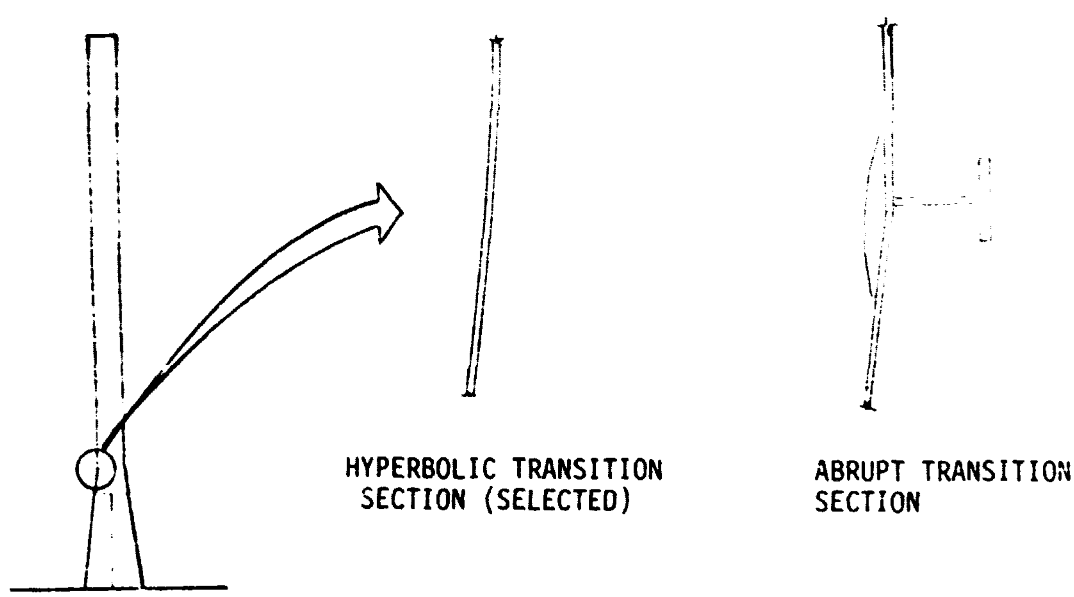


Figure 4-46. Tower Cone to Cylinder Transition Selection

4.2.7 Machine Size Optimization

The features and characteristics of the MOD-2 WTS are the result of extensive conceptual and preliminary design studies. Thorough consideration has been given to the sizing of the WTS for best economic performance in the specified design wind characteristics. The programmatic goal of the MOD-2 project is to produce a system which in production will minimize the cost of electricity (less than 4¢/Kwh for the 100th production unit).

4.2.7.1 Approach

Figure 4-47 illustrates the approach to machine size optimization. In general, the cost and performance of point designs were evaluated in depth. Fundamental design relationships were formulated and empirical cost trends were fitted to the point design data. The derived cost trends make use of the physical parameters such as rotor diameter, torque, RPM, tip speed ratio, power output, design wind velocity, structural loads and weight trends to arrive at system cost trends as a function of rotor diameter, power rating, design wind velocity, etc.

The cost algorithms thus developed were programmed with the annual energy output program to yield the cost of electricity (COE) as a function of machine size parameters.

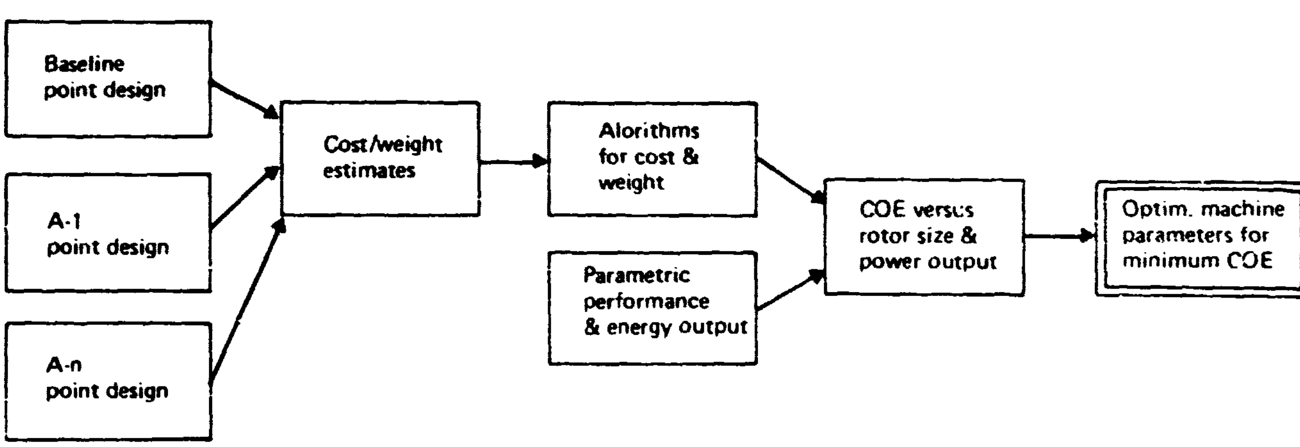


Figure 4-47. Flow Chart for Selection of MOD-2 Optimum Parametric Size

The point designs for which detailed cost estimates were prepared are defined in Table 4-46, and the blade-tower geometry is illustrated in Figure 4-48. Table 4-47 provides the point design cost analysis utilized in developing the cost trend algorithms.

Table 4-46. Cost Validation Design Matrix

Designation	Rotor diameter ft	Power rating kW	V_0/V_r mph	Rotor speed rpm	Tip speed ft/s	Rotor torque 10^6 in-lb
Baseline	300	2,500	20/28.2	17.5	275	13.2
A-1	300	3,500	22.1/32.2	19.34	304	16.7
A-2	300	4,500	23.5/34.7	20.55	323	20.2
A-3	400	4,445	20/28.2	13.13	275	31.3
A-4	500	6,945	20/28.2	10.50	275	61.1

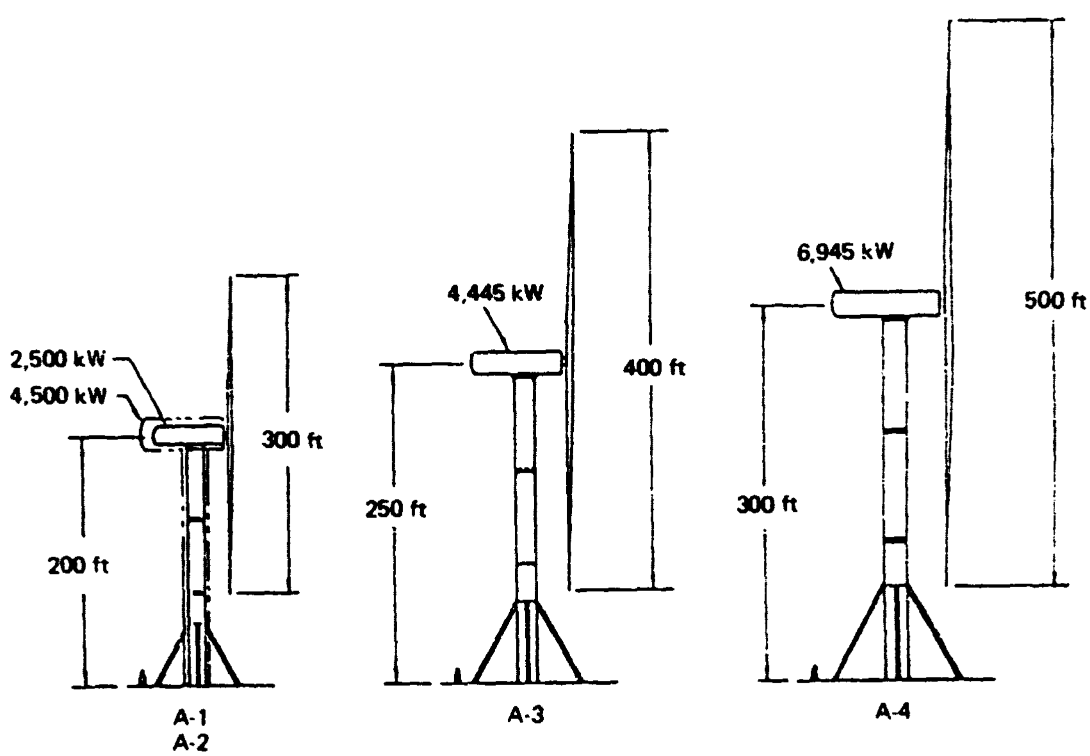


Figure 4-48. Cost Validation Design Matrix

Table 4-47. Point Design Cost Analysis (100th Unit)
($ Thousands)

	Baseline	A–1	A–2	A–3	A–4
1.0 Site preparation	153	153	153	280	450
2.0 Transportation	40	40	42	66	
3.0 Erection	197	250	300	470	
4.0 Rotor	268	296	314	731	
5.0 Drive train	350	480	571	841	
Generator	56	65	79	79	96
Gearbox	249	360	430	636	Incomplete point design analysis
Shafting	45	55	62	126	
6.0 Nacelle	216	275	314	463	
Structure	131	166	176	272	
Yaw system	85	109	138	191	
7.0 Tower	313	381	409	540	
8.0 Initial spares	20	29	32	52	
9.0 Total turnkey costs	1,557	1,904	2,135	3,443	
10.0 Annual O&M	20	25	28	37	

4.2.7.2 Cost Algorithms

The cost algorithms were developed for each of the major cost elements as discussed in the following paragraphs. Table 4-48 is an alphabetical listing of all symbols used for computer programming.

4.2.7.2.1 Site Preparation

The site preparation cost is primarily that of yardwork and of the foundation. The yardwork cost is assumed to be a function of yard size which is proportional to the rotor disk area. The foundation cost depends primarily on the volume of concrete which is a function of the overturning moment due to seismic loads. This moment is a function of the tower height, tower bending frequency and the mass on top of the tower. The tower bending frequency is a function of rotor RPM for a given frequency separation. The resulting algorithm is:

$$SIT = SITYW \left(\frac{DIA}{300}\right)^2 + SITF \left(\frac{WM}{WM_0}\right) \left(\frac{NR}{17.5}\right)^{1.1} \left(\frac{\frac{DIA}{2} + GC}{200}\right)$$

where $WM = WR + WN + WS + WB + WG +$
$+ WNGA + WNIC + WY + WYB +$
$+ WYBG$

All weight equations are provided in section 4.2.7.3.

TABLE 4-48

SYMBOLS FOR COST AND WEIGHT MODELS

A = rotor cost power exponent
AF = availability factor
AKWH = annual KWHR's
AOM = annual O & M cost
CF = capacity factor
CL = lift coefficient
COE = cost of electricity
CP = rotor power coefficient
DIA = rotor diameter-
DRI = cost of drive train
EA = annual energy output
ERE = cost of erection
ES = specific energy
FCR = cost carrying factor
FMAT = % of rotor material cost
FMFG = % of rotor manufacturing cost
GBC = gearbox cost
GC = blade ground clearance
GEC = generator cost
GR = gear step-up ratio
HT = tower (hub) height
IC = initial cost of WTS
KE = baseline erection cost
KBX = program code number
KGB = baseline gearbox cost
KNA = baseline nacelle cost
KNGA = gener. accessory cost
KNI = nacelle instrum. cost
KPC = pitch control cost factor
KRO = baseline rotor cost
KS = spares cost factor
KSHT = baseline shafting cost
KT = transportation cost factor
KTO = baseline tower cost
KW = kilowatt
LF = load factor
LM = tip speed ratio
NAC = nacelle assembly cost
NACS = nacelle structure cost
NG = generator RPM
NR = rotor RPM
NRE = non-recurring cost
NRO = baseline rotor RPM
PS = specific power
ROT = rotor cost
SHC = shafting cost
SIT = site preparation cost
SITF = foundation cost factor
SITYW = yardwork cost factor
SP = spares cost
TOR = rotor torque
TOW = tower cost
TRA = transportation cost
VC/VCO = concrete volume ratio
VO = design wind speed
VR = rated wind speed
VC = concrete volume
WB = gearbox weight
WBO = baseline gearbox weight
WG = generator weight
WGO = baseline generator weight
WM = weight on top of tower
WMO = baseline wt. on top of tower
WN = nacelle structure weight
WNA = baseline nacelle struct. weight
WNAC = nacelle assembly weight
WNGA = gener. accessory weight
WNIC = nacelle instrum. weight
WR = rotor weight
WRO = baseline rotor weight
WS = shafting weight
WSH = baseline shafting weight
WSUM = total WTS weight
WT = tower weight
WTE = weight of electrical equipment
WTO = baseline tower weight
WY = yaw mechanism weight
WYB = yaw brake weight
WYBG = yaw bearing weight
WYBGO = baseline WYBG
WYBO = baseline WYB
WYDO = baseline WYDS
WYDS = yaw drive system cost

4.2.7.2.2 Transportation

Transportation cost will ultimately be very much site dependent and the scenario of truck and rail transport will be further refined. At this time, the transportation cost algorithm states $.05 per lb per 1000 miles.

$$TRA = (.05)\ (WSUM)$$

4.2.7.2.3 Erection

The primary cost items in erection are the assembly and installation of the tower, nacelle and rotor plus the cost of checkout, acceptance testing and specialized equipment. All these items are driven by size and weight which are proportional to the torque and inversely proportional to the design wind speed.

$$ERE = KE \left(\frac{Kw}{2500}\right) \left(\frac{DIA}{300}\right) \Big/ \left(\frac{Vo}{20}\right)$$

4.2.7.2.4 Rotor

The rotor cost is the sum of material cost, manufacturing cost, and the pitch control system cost.

Material cost will vary proportionally to the structural weight. As shown in section 4.2.7.3, the weight varies with the ratio $\left(\frac{D}{300}\right)^{3.4-2a} \times \left(\frac{Kw}{2500}\right)^{a}$.

The manufacturing cost depends highly on the cost of welding and forming. Boeing manufacturing statistical data shows welding cost for Vee double groove to be proportional to the 1.77 power of the plate gauge. The plate gauge is proportional to the diameter and the length of weld is also proportional to the diameter. As shown in the weight expression, the weight (plate gauge) is also a function of the power rating. Therefore, the manufacturing cost algorithm is adjusted for power rating similar to the structural weight.

From point design cost analysis, it was found that material was approximately 37% and manufacturing 63% of the total structural cost. The pitch control system cost was found to be 14.85% of the rotor structural cost.

The resulting algorithm is:

$$ROT = (KRO) \left[(FMAT) \left(\frac{D}{300}\right)^{3.4-2a} \left(\frac{Kw}{2500}\right)^{a} + (FMFG) \left(\frac{D}{300}\right)^{2.77-2a} \left(\frac{Kw}{2500}\right)^{a} \right] (KPc)$$

where: KMAT = 0.37
KMFG = 0.63
KPc = .1485
KRO = baseline rotor cost
a = 0.4

4.2.7.2.5 Drive Train

The drive train cost is the sum of three major subsystems, i.e.:

- o Generator
- o Gearbox with lub systems
- o Shafting with couplings and rotor brake

o <u>Generator</u>

The generator cost trend expression correlates with actual vendor cost data as shown in Figure 4-51.

$$GEC = 44.42\ (KW)^{.898}$$

o <u>Gearbox</u>

Gearbox cost is proportional to torque which is therefore expressed as:

$$GBC = KGB\ \left(\frac{Kw}{2500}\right)\ \left(\frac{D}{300}\right)\ \left(\frac{20}{Vo}\right)$$

This trend correlates with information from gearbox manufacturers as shown in Figure 4-52.

o <u>Shaft with Couplings and Rotor Brake</u>

The shafting and associated hardware is designed primarily by torque. The exponents in the cost algorithms provide adjustments to correlate with actual cost analysis of design points representing variations in power rating and rotor diameter.

$$SHC = KSHT\ \left(\frac{Kw}{2500}\right)^{0.8}\ \left(\frac{DIA}{300}\right)^{1.5}\ \left(\frac{20}{Vo}\right)$$

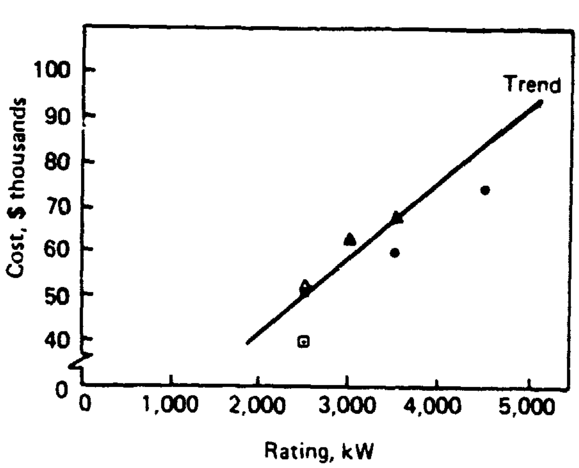


Figure 4-51. Generator Cost

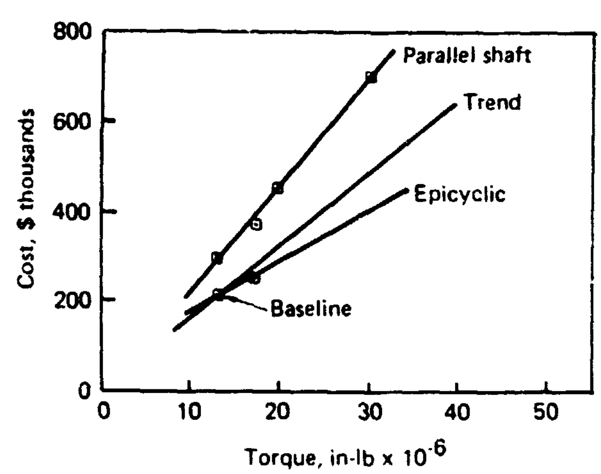


Figure 4-52. Gearbox Cost

4.2.7.2.6 Nacelle

The nacelle cost includes the nacelle structure and the yaw system. The instrumentation and generator accessory unit costs have been programmed into the nacelle structure cost.

- <u>Structure</u>

  Based on the actual point design analysis, the following cost ratios and principal cost drivers have been established.

COMPONENT	% OF COST	COST DRIVER	
Bed Structure	42	Rotor torque 21% Static moment 21%	(1)
Shroud	31	Volume	(2)
Rotor Support	19	Rotor Weight	
Environmental Control	7	Volume	(2)
Cabling	1	Power Rating	(3)

Notes: (1) The effects of torque and static moment are of approximately equal magnitude.
(2) Nacelle volume varies with DIA $^{2.5}$
(3) Include in the torque term

The resulting cost trend expression is:

$$NACS = KNA \left[.21 \left(\frac{DIA}{300}\right) \left(\frac{Kw}{2500}\right) + .22 \left(\frac{Kw}{2500}\right) \left(\frac{20}{Vo}\right) + .19 \left(\frac{WR}{WR_0}\right) + + .38 \left(\frac{DIA}{300}\right)^{2.5} \right] + KNI + KNGA$$

where: KNA = baseline nacelle structure cost
WR = weight of rotor (see 4.2.7.3)
KNI = constant instrumentation cost
KNGA = constant generator accessory unit cost

- <u>Yaw System</u>

  The yaw system is subdivided into the drive system, the brake, and the bearing. The cost trend of each subsystem is expressed as a direct function of the subsystem weight. The weight expressions (see section 4.2.7.3) are expressed as a function of the design parameters.

$$YDS = YDO \left(\frac{WYDS}{WYDO}\right)$$

$$YDB = YBO \left(\frac{WYB}{WYBO}\right)$$

$$YBG = YBGO \left(\frac{WYBG}{WYBGO}\right)$$

4.2.7.2.7 Tower

The tower cost expression includes the tower structure, electrical equipment including slip rings, bus tie contactor and disconnect switch plus the cost of the transformer and power cable. The tower structure is designed to a first mode bending frequency below the two per rev forcing frequency of the rotor. The bending stiffness requirement directly determines the weight and cost of structure. The cost trend equation is therefore:

$$TOW = (KTO)\left(\frac{HT}{200}\right)\left(\frac{NR}{17.5}\right)\left(\frac{WM}{WMo}\right)^{1/3} + 52{,}000 + (17)(KW)$$

where: 52,000 = constant electrical equipment

17 KW = transformer and power cabling

4.2.7.2.8 Initial Spare Parts

Spares cost are expressed as a function of the total WTS hardware cost.

$$SP = KS\ (ROT + DRI + NAC + TOW)$$

Analysis for machine size optimization utilized KS = .0184.

4.2.7.2.9 Initial Turnkey Cost

$$IC = SIT + TRA + ERE + ROT + DRI + NAC + TOW + SP$$

4.2.7.2.10 Annual Operations and Maintenance

The cost trend expression assumes a 25 unit cluster of WTS with crew costs fixed and some of the consumables vary with size.

$$AOM = 12{,}500 + 4500\left(\frac{Dia}{300}\right)^{2.85}$$

4.2.7.3 Weight Algorithms

The weight trend relationships were developed similarly to the cost trend expressions. The fundamental design parameter relationships were formulated and empirical weight trends were fitted to the point design data. The following expressions were incorporated with the cost expressions into the energy output computer program to perform the machine size optimization.

$$\frac{VC}{VCO} = \left(\frac{WM}{WMO}\right) \left(\frac{NR}{NRO}\right)^{1.1} \left(\frac{\frac{DIA}{2} + GC}{200}\right)$$

$$\frac{WR}{WRO} = \left(\frac{DIA}{300}\right)^{3.4-2a} \left(\frac{KW}{2500}\right)^{a}$$

where a = 0.4 for teetering rotors

$$\frac{WG}{WGO} = \left(\frac{KW}{2500}\right)^{0.7}$$

$$WB = WBO + 3450 \left(\frac{TOR - 13.2 \times 10^6}{10^6}\right)$$

$$\frac{WS}{WSH} = \left(\frac{TOR}{13.2 \times 10^6}\right)^{1/3}$$

$$WN = WNA \left[(.36) \left(\frac{DIA}{300}\right) \left(\frac{KW}{2500}\right) + (.36) \left(\frac{KW}{2500}\right) \left(\frac{20}{Vo}\right) + (.15) \left(\frac{WR}{WRO}\right) + (.13) \left(\frac{DIA}{300}\right)^{2.5}\right] + WNIC + WNGA$$

where:
- WNIC = baseline weight of instruments
- WNGA = baseline weight of generator accessories

$$WYDS = (WYDO) \left(\frac{KW}{2500}\right) \left(\frac{17.5}{NR}\right)$$

$$WYB = WYBO \left(\frac{DIA}{300}\right)^3$$

$$WYBG = WYBGO \left(\frac{DIA}{300}\right)^3$$

$$WT = WTO \left(\frac{HT}{200}\right)^2 \left(\frac{NR}{NRO}\right) \left(\frac{WM}{WMO}\right)^{1/3} + WTE$$

4.2.7.4 Machine Size Optimization Results

Utilizing the foregoing cost and weight expressions combined into the energy output computer program allowed a comprehensive analysis of design optimization for minimum cost of electricity. This program allows rapid evaluation of all parameters and optimization for various wind models or various mean wind speeds.

As the MOD-2 program progressed, the baseline values in the cost and weight expressions have been updated to reflect the continuing depth of point design analysis. The program was also updated to utilize the NASA recommended variable power wind gradient instead of the originally specified gradient. These changes resulted in a continuous adjustment of the value of COE (as noted in other data in this document) but the machine optimization results remain valid.

As shown in Figure 4-55, the optimum COE is only slightly affected by power rating but is more sensitive to optimizing the rotor diameter for a given power rating. Examination of the cost and weight trend equations give insight into the reasons for the buckets of these curves. Some of the more significant factors are:

1) Large rotor diameters for a given power, even as RPM decreases, significantly influences size and cost of the total system.

2) Small rotor diameters for a given power, require higher RPM and reach rated power only at the higher wind speeds. This results in significant loss in annual energy output.

3) A higher power rating optimizes at higher rotor diameters due to lower RPM and achieving rated power at a lower wind speed thereby increasing annual energy.

Table 4-48 summarizes the trade study performed at the time of the preliminary design review. Based on least cost of electricity, the 300 foot diameter rotor and 2500 KW generator rating was selected for MOD-2 at the specified 14 mph mean wind site.

The computer optimization program allowed a rapid optimization analysis at various values of mean wind speed as may be typical of different sites. Figure 4-56 shows the relative COE of the MOD-2 baseline as a function of wind speed and compares to the COE of an optimum sizing (KW and Diameter). It is noted that the MOD-2 baseline is very near optimum over a significant range of mean wind speeds.

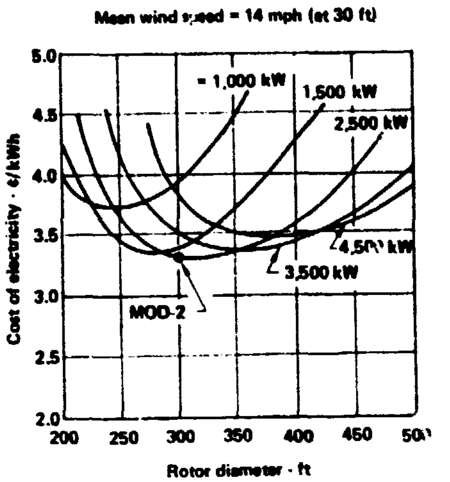


Figure 4-55. Parametric COE Trends

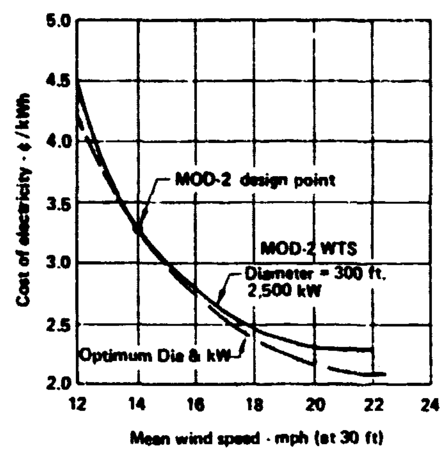


Figure 4-56. Effect of Mean Wind Speed on COE

Table 4-48. Optimum Machine Size Trade Study Summary

CRITERIA	CONFIGURATION				REMARKS
	BASELINE 300' DIA/2500 kW	A-1 300'DIA/3500 kW	A-2 300'DIA/4500 kW	A-3 400' DIA/4450 kW	
DESIGN COMPLEXITY				ROTOR BLADES REQ. FIELD ASSY.	
		TOWER REQUIRES LONGITUDINAL SPLICE			
			LARGER NACELLE REQ. ADDITIONAL FIELD ASSY.		
			HIGH TORQUE GEARBOX AVAILABILITY QUESTIONABLE		
ANNUAL ENERGY OUTPUT - 10^6 kWH	8.680	9.60	10.21	16.44	PER UNIT-INCLUDES .96 AVAIL FACTOR
SYSTEM COST 100TH UNIT	1,560,000	1,840,000	2,077,000	3,284,000	
$/kW	625	526	461	739	
FARM SIZE - N UNITS	25	22.6	21.3	13.2	BASED ON EQUAL ANNUAL ENERGY OUTPUT PER FARM
FARM COST - 10^6 $	39.0	41.6	44.1	43.3	
COST OF ELECTRICITY ¢/kWHR	3.47	3.66	3.86	3.75	

4.2.8 Control System

This section describes the results of trade studies which evaluated alternative control systems.

4.2.8.1 Analog vs Microprocessor

OBJECTIVE: Several choices of technology are available to the designer for implementing the electronic portion of the MOD-2 control system. These choices grossly can be categorized under the headings programmed digital (computer), hardwired digital, and analog electronics. Table 4-47 describes functions that are candidates to be performed by a control computer. The prime criteria used in selecting the technology for a function are cost, reliability, and performance. Other criteria normally used in aerospace applications such as size and power usage are not a significant consideration in the Wind Turbine System.

Table 4-47. Potential Control Functions in the MOD-2 Wind Turbine System

- ALL MODES
  - Monitor for alarms
  - Notify nearest utility substation of alarms
  - Respond to substation commands/requests
  - Respond (with safety limitations) to on-site manual commands/requests
- STARTUP/SHUTDOWN MODE
  - Sequence mechanical & electrical systems
    - Turn on/off pumps, brakes, generator exciter, etc.
    - Monitor for equipment performance (oil press, etc.), intrusion
    - Command pitch profile for startup & shutdown
- UP & RUNNING MODE
  - Operate on algorithms and command pitch and yaw
  - Monitor for equipment performance/intrusion
  - Command shutdowns (separate system will back up critical failures, e.g., vibration.
  - Monitor wind for yaw or shutdown criteria
- STANDBY MODE
  - Monitor for equipment performance
  - Monitor for startup conditions (wind, etc.)

The development of a prototype system such as MOD-2 brings in another design criteria, that of flexibility of the system to change with minimum impact on costs or schedule. The automatic control system of an unattended site will have an interface with many of the mechanical, hydraulic, and electrical elements of the system and is consequently sensitive to changes in those elements. Flexibility is particularly an attribute of the digital computer where changes can be accomodated by software rather than hardware.

RESULTS: Reliability - The most compelling reason for considering digital computer implementation for performing Wind Turbine functions is its design simplicity and consequently reliability. This is particularly true with the microcomputer. In the microcomputer the same few monolithic circuits (CPU, timing module, memory, etc.) can be used for the numerous sequencing, timing, discrete state monitoring, and discrete command functions of the system. With analog technology, each of these functions requires its own set of hardware.

One functional element that will be implemented outside of the computer is the safety shutdown system. The reason for a hardwired design in lieu of a programmed digital system is the simplicity of this system and the predictability of the failure modes of the former rather than reliability.

Performance - In general, the performance requirements of the Wind Turbine System (accuracy, resolution, speed, etc.) are so modest that either digital or analog technology may be used. An example of potential performance superiority of a digital implementation may be found in a filter application. The control system will respond to drive train parameters (e.g., generator output, low speed shaft acceleration) to control the effects of variable wind. It is desireable that the control system not respond (at least in the same way) to drive train periodic oscillations. One solution is to filter those effects out of the sensed parameter signal.

Analog notch filters, at the very low frequencies in question, must be finely tuned to restrict their bandwidth. Further, both the analog filter's bandwidth, and center frequency are subject to change as its individual component parameters drift with temperature and age. The computer notch filter performance is determined by initial design implementation (word size, processing speed) and will not change with time or environment.

Cost - The potential cost advantages of a microprocessor are tied to the same design simplification goals mentioned under reliability. Fewer parts result in a simplified manufacturing process. In addition, the costs of microcomputer components (particularly CPU and memory) are in a significant down trend stage.

Another potential cost advantage of the digital computer systems is in the usage of off-the-shelf electronics. An analog or hardwired digital board is designed especially for the particular application and must bear the small quantity production costs of that application. Digital computers (micro or mini) are commercially available in off-the-shelf board/module form. It is the software and the amount (rather than design) of memory and I/O in the computer that is unique to the application.

CONCLUSIONS: It was determined that a microprocessor-based digital system is the best technology for the majority of MOD-2 control functions for the following reasons:

1. Superior availability/reliability as the result of reduced quantities of electrical parts and solder connections.

2. Reduced costs due to a simplified manufacturing process for the less complex boards and potential cost benefits from the current down-trend on prices of the microprocessor and associated modules.

3. Reduced costs due to use of commercially available microprocessor boards.

4. Flexibility of the system to change with minimum program impact.

5. Superior performance in some areas, where the analog system's drifting with temperature and age is critical.

4.2.8.2 Multiplexer, Ground Computer Vs. Nacelle Microcomputer

OBJECTIVE: The current control system (Nacelle Microcomputer) is described in section 3. This trade study provided the basis for its selection over the original baseline (Multiplexer, Ground Computer).

System Requirements: The control system must perform the functions listed in Table 4-48. To perform these functions it interfaces with the remaining WTS systems as shown in Table 4-49.

Configuration traded: The configurations studied are shown in Figure 4-52. The original baseline consists of a microcomputer-based system, located at the base of the tower, which interfaces with nacelle WTS equipments through an adaptation of the Boeing model 1014 Multiplexer. The proposed alternate locates the microcomputer in the nacelle and uses direct wiring for the few sensors and controls located at the base of the tower.

RESULTS: The results of this study are summarized in Table 4-50. The difference in cost and MTBF are directly related to the lower parts count of the nacelle microcomputer system. This lower parts count is possible because the majority of the control system interfaces originate in the nacelle thus eliminating the multiplexer/Demultiplexer hardware required in the original baseline.

CONCLUSION: The Nacelle Microcomputer System offers improvements in reliability, 100th unit cost and cost of electricity with no known penalties. Therefore, it is recommended that the original baseline be replaced with the Nacelle Microcomputer System.

Table 4-48. Control System Functions

- FUNCTIONAL REQUIREMENTS
  - AUTOMATIC OPERATION OF UNATTENDED SITES
    - OPERATING MODE SELECTION/SEQUENCING
      1) START UP
      2) CONNECTION TO UTILITY
      3) SYNCHRONIZED OPERATION WITH UTILITY
      4) SHUT DOWN
         - LOW WINDS/HIGH WINDS
         - EMERGENCY
      5) STANDBY
    - SYSTEM PROTECTION
    - SYSTEM STABILITY CONTROL
      1) WIND VARIATIONS
      2) UTILITY LOAD VARIATIONS
    - SYSTEM DATA COLLECTION
      1) MONITORING FOR OPERATION & STATUS
      2) STORAGE
      3) TRANSMISSION TO UTILITY DISPATCHER
  - MANUAL INTERFACE
    - UTILITY DISPATCHER CONTROL & DISPLAY
    - SITE MANUAL CONTROL & DISPLAY

Table 4-49. Control System Interfaces

	DISCRETE INPUTS
1)	GEARBOX OIL OVERTEMP
	" OIL PRESS
	" OIL LEVEL
	" PARTICLES
	" OIL START TEMP (SUMP)
	" OIL START TEMP (INLET)
	" OIL FILTER STATUS
	MAIN BEARING TEMP #1
	" " " #2
10)	" " OIL FLOW
	YAW PULSATION ACCUM PRESSURE
	" BRAKE " "
	" HYDR OIL LEVEL
	" " " PRESSURE
	" " " FILTER #1
	" " " FILTER #2
	" " " TEMP
	" CABLE WRAP LIMIT (360° CW)
	" " " " (360° CCW)
20)	ENVIRONMENTAL OVERTEMP
	PITCH UNLOCK STATUS
	PITCH EMERGENCY ACCUM PRESS #1
	" " " " #2
	" OPERATIONAL " "
	" HYDR OIL LEVEL
	" " " PRESS
	" " " TEMP
	" " " FILTER #1
	" " " FILTER #2

	DISCRETE INPUTS
30)	ROTOR HUB VIBRATION
	HIGH SPEED SHAFT VIBRATION
	ROTOR BRAKE TEMP #1
	" " " #2
	" CLOCK POSITION #1
	" " " #2
	" PITCH LOCK STATUS
	" OVERSPEED
	NACELLE INTRUSION
	GENERATOR/BUS TIE BREAKER ▶
40)	GROUND INTRUSION ▶
	MANUAL EMERGENCY STOP ▶
	" MODE SELECT ▶
	" SHUTDOWN/DISABLE ▶
	" UTILITY POWER PRESENT ▶
	FAILSAFE MONITOR
	SHAFT BRAKE STATUS
47)	NACELLE MANUAL EMERGENCY STOP

▶ GROUND DATA SOURCES - ALL OTHERS LOCATED IN NACELLE

Table 4-49. Control System Interfaces (Cont)

	ANALOG INPUTS
1)	ROTOR PITCH ANGLE
	SHAFT SPEED
	WIND SPEED #1
	WIND SPEED #2
	WIND DIRECTION #1
	WIND DIRECTION #2
	GENERATOR POWER
8)	NACELLE POSITION

	ANALOG OUTPUTS
1)	PITCH COMMAND

	DISCRETE/DIGITAL OUTPUTS
1)	PITCH HYDR. PUMP ON
	" " " OFF
	YAW " " ON
	" " " OFF
	LUBE OIL " ON
	" " " OFF
	ROTOR BRAKE RELEASE
	YAW BRAKE RELEASE
	YAW LEFT
10)	YAW RIGHT
	EMERGENCY FEATHER SOLENOID
	GENERATOR FIELD ENABLE
	AUTOMATIC SYNCHRONIZER ENABLE ▶
	BUS TIE BREAKER DISCONNECT ▶
	. . . ▶
	RS232 DATA INTERFACE #1 ▶
16)	" " " #2 ▶

▶ GROUND INTERFACES - ALL OTHERS LOCATED IN NACELLE

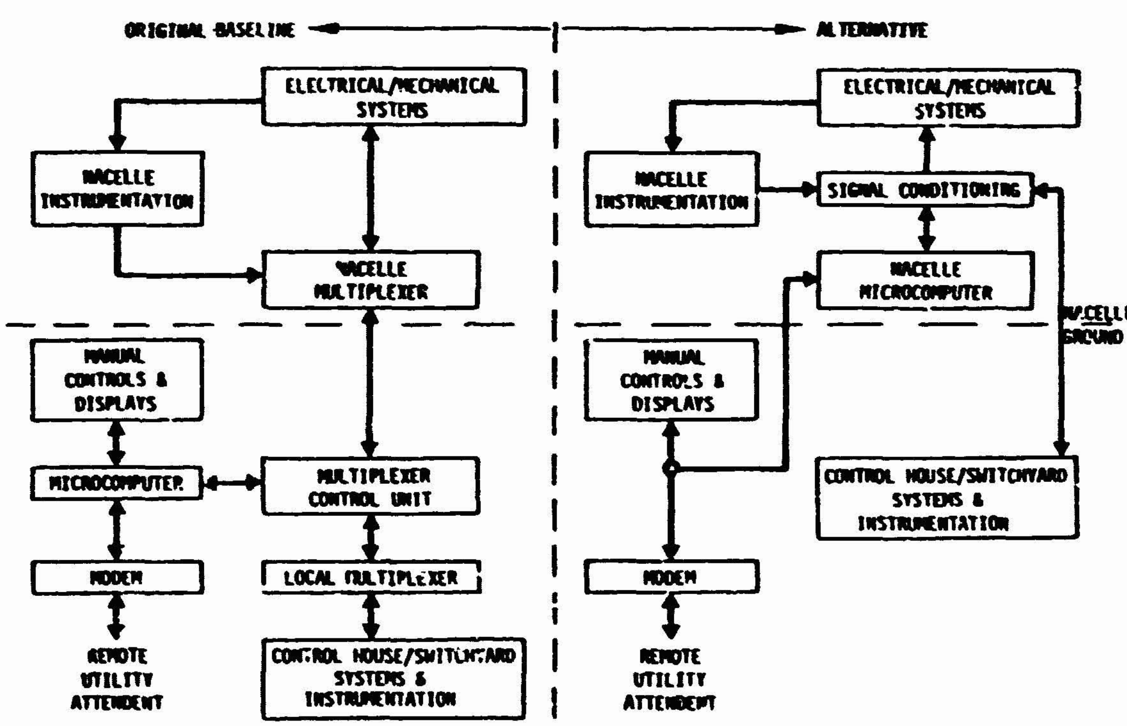


Figure 4-52. Control System Alternatives

Table 4-50. Control System Alternative Summary

	BASELINE GROUND COMPUTER + DISTRIBUTED MULTIPLEXER & SIGNAL CONDITION	ALTERNATIVE NACELLE COMPUTER & SIGNAL CONDITIONING
DESIGN COMPLEXITY	SAME - STATE OF ART	
RELIABILITY	1472 HR MTBF (SIGNIFICANTLY MORE PARTS)	5000 HR MTBF
MAINTAINABILITY	SAME .	
TECHNICAL RISK	SAME - LOW	
SYSTEM COST (100 UNIT)	BASELINE	40% LESS
COST OF ELECTRICITY	BASELINE	.10¢/kWh REDUCTION

4.3 ALTERNATE ROTOR DEVELOPMENT

4.3.1 Introduction

On August 10, 1978, more extensive study was initiated to include the preliminary design and evaluation of a composite (Fiberglass) rotor for the MOD-2 WTS. Structural Composites Industries (SCI) was subcontracted to provide preliminary design, analysis, program definition, and cost estimates. BEC supported the study with definition of MOD-2 requirements, loads, design and analysis of the metal components, and the programmatic comparison to the steel rotor.

The objective of this study was to bring the preliminary design of a MOD-2 composite rotor to sufficient maturity to enable evaluation of the technical and programmatic impacts of developing the composite rotor.

4.3.2 Summary

The composite rotor configuration has the features which, when fully developed, may offer advantage to the WTS in quantity production. These features and the comparison to the steel rotor are discussed more fully in section 4.2.2.10. The design is based upon the SCI experience in fabrication and test of the 150 foot test blade. Several features as discussed in section 4.3.5 differ from the test blade design. These differences accommodate the MOD-2 requirements, allow interchangability with steel blades on a MOD-2 WTS and, most significantly, offer reduced technical risk and reduced cost in production quantities.

The configuration, as defined in section 4.3.5, utilizes non-rotating mandrels and a ring winding process* to apply Transverse Filament Tape (TFT)* to the major portion of the multi-spar blade. The fiberglass is wound integrally onto steel torque boxes at the tip blade pivot area and onto a steel hub to provide the field assembly joint. The steel components comprise 33 percent of the rotor weight. The fiberglass components (67 percent) are 98 percent applied by the low cost winding process. The small trailing edge minimized afterbody cost and eliminates the risk of system damage due to possible failure of a trailing edge bond. The conclusion from the trade study is that the composite blade is feasible. As compared to the steel rotor, it offers additional technical risk, offers equivalent cost for production units, is less fully developed, and would incur program delay for prototype hardware. The steel rotor is therefore recommended as the baseline configuration for MOD-2.

4.3.3 Design Requirements

At the time the composite rotor study was initiated, the MOD-2 WTS had completed the system optimization studies of the Conceptual Design Phase and was well into the Preliminary Design Phase. The composite rotor therefore had the requirements to be compatible with the existing MOD-2 system. Tables 4-51 and 4-52 summarize the requirements and goals imposed on the composite rotor design.

*Patents Pending

Table 4-51. Design Requirements

- Compatibility With MOD-2 System Parameters
  - 2500 KW Generator
  - 300 Ft Diameter Rotor
  - Tip Control Blade (30 percent span)
  - NACA 230XX Airfoil
  - $+ 2.5^\circ$ to $- 4^\circ$ Twist
  - Tip Rotation from -5° to $+ 95^\circ$
  - Teeter $\pm 5^\circ$
  - Upwind Rotor
  - 17.5 RPM
  - 30-year life
  - NASA Specified Environment
  - Natural Frequence Separation
    - From Integer values by 0.25 cycles/rev
    - First Flapwise Frequency between 2.5 and 2.75 cycles/rev.

Table 4-52. Design Goals

- Minimize Cost of Electricity (< 4¢/kWh)
  - Minimum material costs
  - Automatic Production for Minimum Labor Cost
  - Minimum Maintenance Cost
- Minimize Technical Risk
  - Use existing technology
  - Conservative Design with Margins of Safety
  - Redundant Load Paths at Critical Hub Joint

4.3.4 Prelimary Design Load Conditions

Table 4-53 itemizes the load conditions analyzed for the composite blade preliminary design. The limit operating, extreme wind, and control system fault loads occur only infrequently over the design life of the wind turbine, and consequently were used only in buckling and maximum strength calculations. Fatigue loads were determined based on the MOD-2 gust criteria to establish a loads spectrum over the operating wind speeds of 14 to 45 mph at 20° yaw.

All blade loads were calculated using the MOSTAB computer program, with the exception of extreme wind loads, which were hand-calculated. Composite blade mode shapes and frequencies used in the MOSTAB program were computed by SCI using the SPAR computer program. Overspeeds and trim conditions used in the MOSTAB quasi-steady shutdown analysis were computed in a detailed aerodynamics program which assumes uniform time-varying flow and a rigid blade, with pitch rate specified in the input. Stresses in the blade were calculated in a MOSTAB postprocessor, which first translated loads from the blade reference line to the elastic axis at each cross section, and then summed the stresses due to chordwise bending, flapwise bending, and centrifugal force at locations around the airfoil. Fatigue stresses were calculated for the MOD-2 wind spectrum which result in a maximum discrete stress value for each mean wind speed having a 99.9 percent probability of non-exceedance. Number of cycles of loading were computed based on a Weibull distribution of mean wind speeds, and the MOD-2 WTS operating schedule.

Table 4-53. Composite Blade Preliminary Design Load Conditions

DESIGN CONDITION	DESCRIPTION
Limit Operating (0.01% probability of non-exceedance)	17.5 rpm, 16.5 mph mean wind
Control System Fault	
150% Power	17.5 rpm, 54 mph, 5000 hp
Emergency Shutdown	19.6 rpm, 45 mph, loss of generator load
Extreme Wind	0 rpm, 120 mph, tip feathered
Fatigue (Preliminary)	17.5 rpm, 45 mph, 30° Yaw with gust factors
Fatigue (Final	17.5 rpm, 14 to 45 mph mean wind spectrum, extrapolated to 20° yaw

4.3.5 Configuration

The composite rotor was configured to meet the design requirements of Section 4.3.3 and to be compatible with the MOD-2 System, as discussed in Section 4.3.8. The design goals of low cost and minimum technical risk were strong influences on the final configuration, as was the recent experience gained on the DOE/NASA 150 Ft. Composite Blade Program. Wherever possible, overall configuration and design details from the BEC MOD-2 steel blade were incorporated into the composite rotor configuration, with modifications as appropriate.

4.3.5.1 General Arrangement

The composite rotor general arrangement is shown in Figures 4-53 and 4-54. The rotor consists of two composite inner blade sections, two composite tip sections, and a steel hub. Rotor external geometry is identical to the BEC MOD-2 steel blade from Station 900 to Station 1800. Inboard of Station 900 the chord and thickness of the composite rotor have been increased to provide adequate stiffness to meet the MOD-2 natural frequency requirements. A full airfoil is maintained from Station 360 outboard. Inboard of Station 360, the cross-section transitions to an ellipse at the hub.

4.3.5.2 Inner Blade Assembly

The inner blade assembly is shown in Figure 4-55. The basic load carrying composite structure consists of an airfoil-shaped outer blade shell, with an inner "box spar". The box spar provides two vertical webs which serve to reduce unsupported panel width for enhanced buckling stability, and additional thickness at the outer fiber for reacting flapwise bending loads. Planforms of the box spar and shell are tapered linearly to facilitate mandrel extraction (see Section 4.3.7).

Structural and manufacturing considerations preclude a sharp trailing edge on the structural blade shell; therefore, small, light-weight, non-structural molded fairings are bonded onto the trailing edge to provide the required shape. The elliptical hub end of the blade shell is detailed for joining to the steel hub (see Section 4.3.5.4). The outer end of the blade shell contains a wound-in steel torque box which mounts the pitch control spindle, actuator and controls (Section 4.5.5.5).

4.3.5.3 Tip Assembly

The basic composite structure of the tip assembly differs from the inner blade assembly due to the requirement for the chordwise center of gravity to be located farther forward. This is accomplished by using two inner "D" shaped spar members and an outer shell as shown in Figure 4-56. Again two vertical webs are provided, with linear planform taper for mandrel extraction. The trailing edge fairing is similar to but smaller than that of the inner blade. The steel and composite spindle box details are bonded in after mandrel extraction. An aluminum closure is fastened to the tip by bolts and clips.

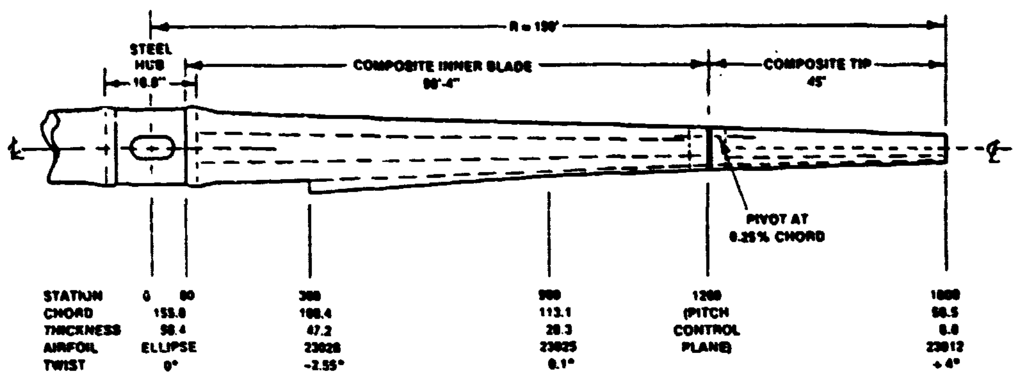


Figure 4-53. Composite Rotor General Arrangement

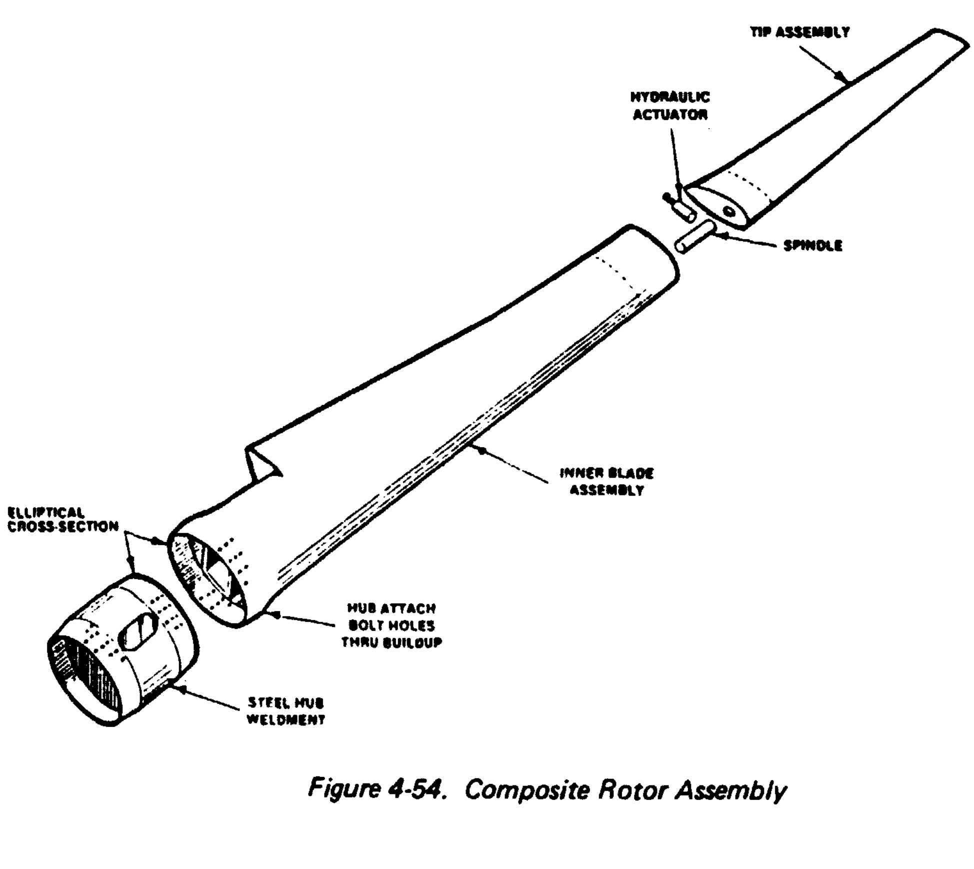


Figure 4-54. Composite Rotor Assembly

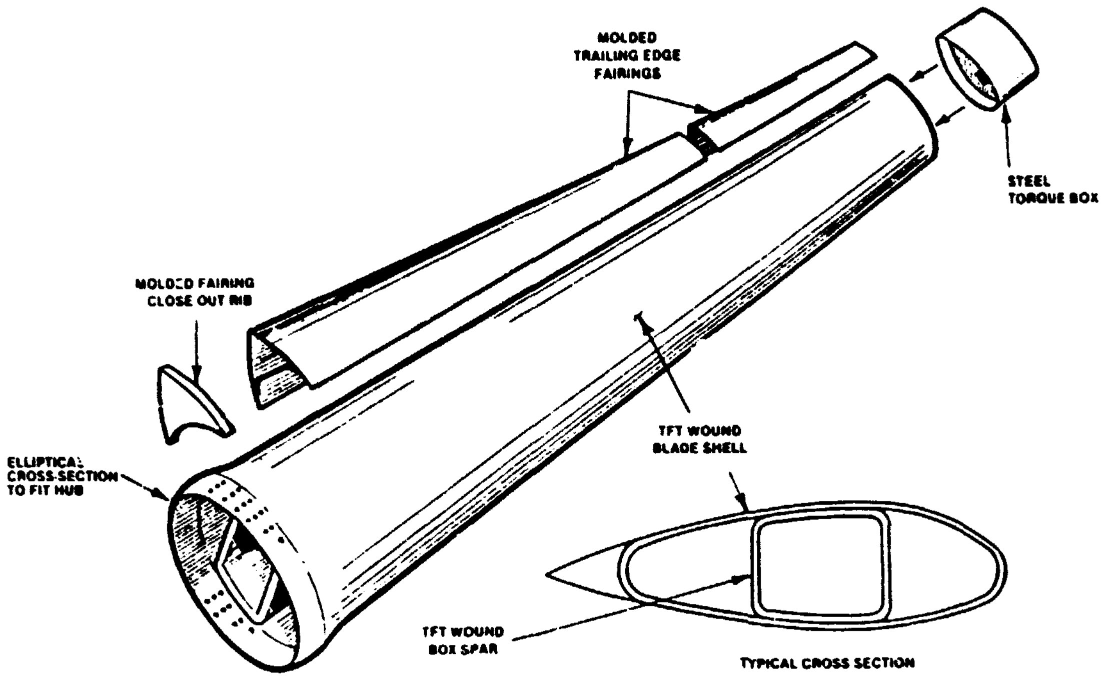


Figure 4-55. Composite Rotor Inner Blade Assembly

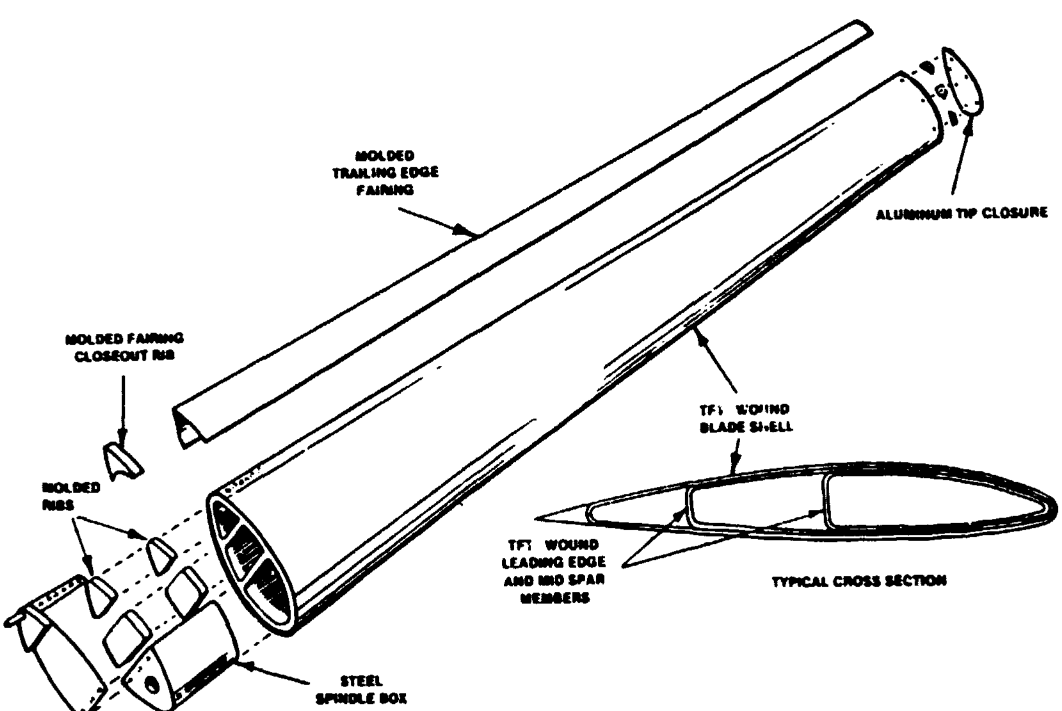


Figure 4-56. Composite Rotor Tip Blade Assembly

4.3.5.4 Hub and Hub Joint

The steel hub is shown in Figure 4-57 and a typical hub joint detail is shown in Figure 4-58.

The hub is of welded steel construction with shaft teeter bearing details similar to the MOD-2 steel rotor. A drilled, tapered, elliptical flange is provided for mating to the composite blade shell and the vertical webs are extended to splice to the composite box spar.

In order to ensure a good fit between the hub and blade shell, the composite shell is wound using the hub as a mandrel to form the joint area. Extra reinforcements and steel shim stock are incorporated into the composite joint area to give the desired strength, bolt bearing, and stiffness properties (see Section 4.3.6).

The field joint is both bonded and bolted, with either the adhesive bond or the bolts capable of carrying the maximum joint loads.

4.3.5.5 Pivot Joint

The pitch control pivot joint is shown in Figure 4-59. The air-foil-shaped torque box, contained in the inner blade assembly, is a steel weldment with pivot and actuator assemblies similar to the MOD-2 steel blade. It is retained by bonding and bolting, as well as being mechanically inter-locked by being wound into the composite blade shell. A molded composite clip and a rib are used to terminate the box spar webs.

The tip section construction differs in that the steel shaft receptacle weld-ment is inserted in the forward D-spar only. A steel rib assembly carries the actuator loads, assisted by molded composite internal ribs in the afterbody area. All components are assembled by bolting and bonding. As with the hub, the shaft receptacle torque box is used as a winding mandrel to ensure a proper fit for subsequent bonding.

4.3.5.6 Trailing Edge Fairing

Figure 4-60 shows some typical cross-sections of the molded composite trailing edge fairing. It is attached to the blade structure by bonding, (similar to the afterbody on the 150 ft. blade). Molded composite webs, and ribs bonded in place, are used to stiffen and terminate the fairing and to locate it on the blade shell. (see Section 4.3.7).

4.3.5.7 Environmental Protection

Lightning protection is provided by imbedding a grounded 120 mesh aluminum screen in the outer surface of the composite. This is followed by a coating of polyurethane paint for UV and moisture resistance. Abrasion protection of the tip assembly leading edge is provided by a polyurethane rubber boot.

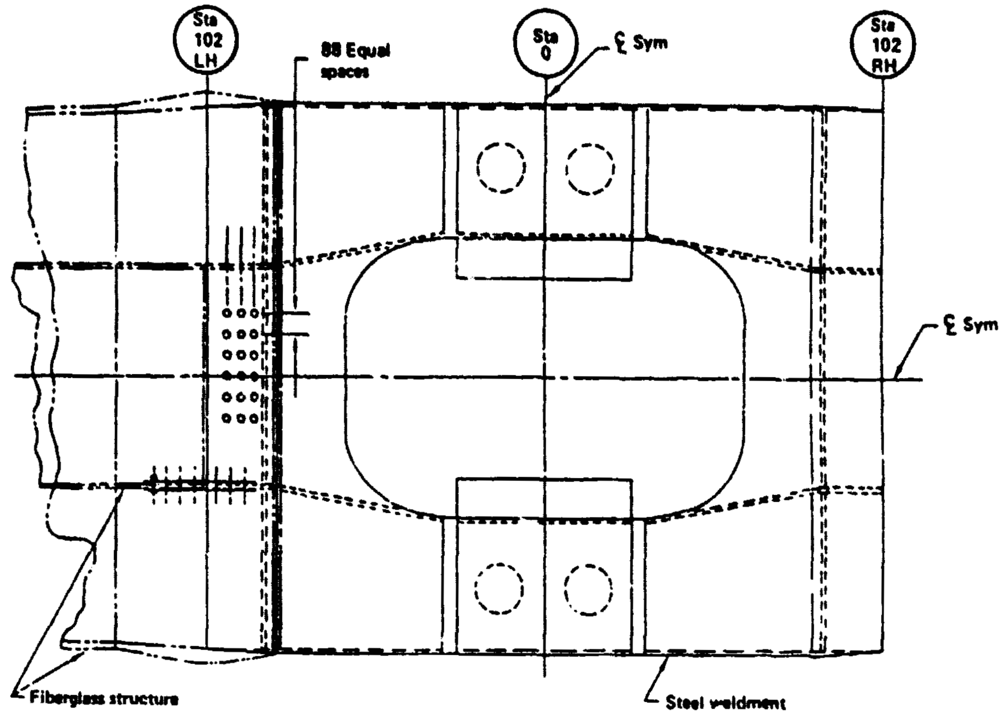


Figure 4-57. Composite Rotor - Steel Hub & Field Joint

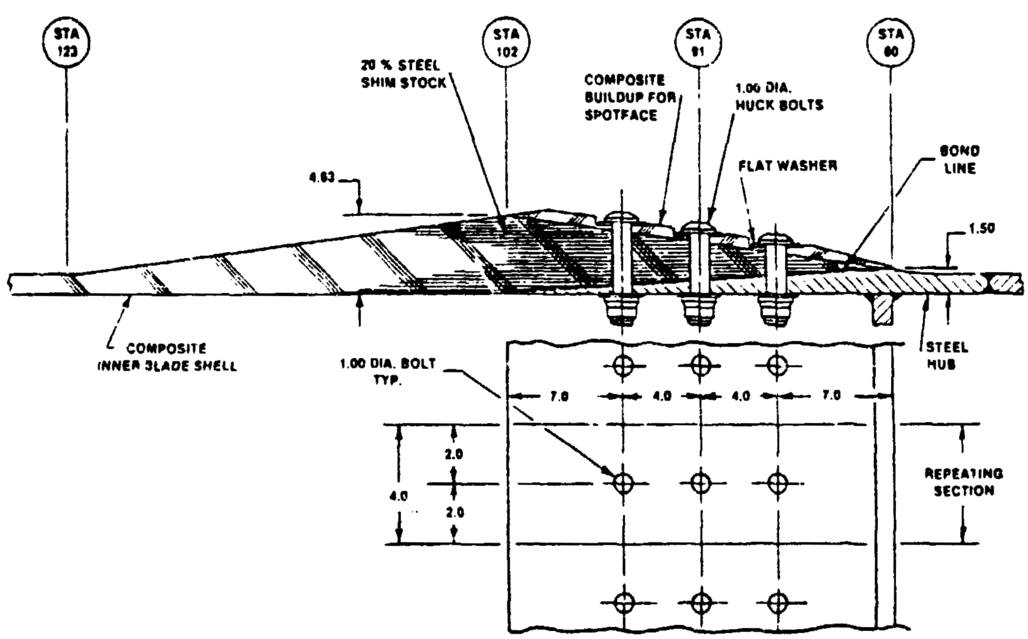


Figure 4-58. Composite Rotor Hub Joint Detail

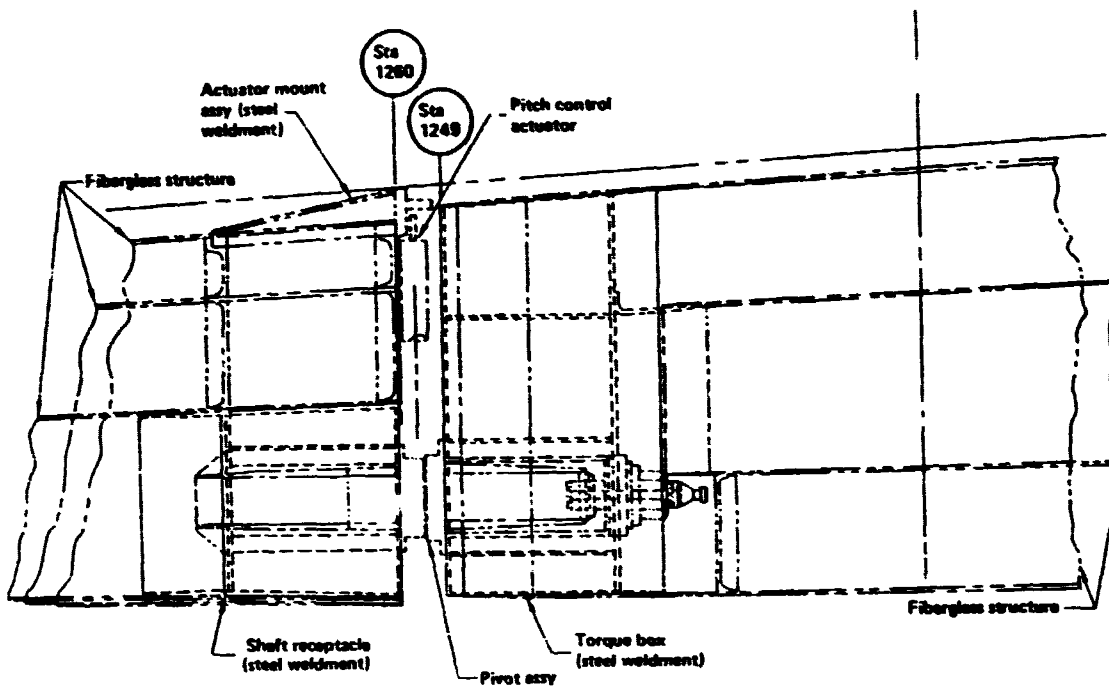


Figure 4-59. Composite Rotor Pivot Joint

INNER BLADE – 0.2 < r/R < 0.5

STA r/R	L1 In.	L2 In.	L3 In.	L4 In.	h In.	Wt./Ft. lb.
.3	21.06	29	15.5	13.5	.4	28
5	5.05	7.5	4.5	3	.125	2.2

OUTER HALF OF SPAN

STA r/R	L1 IN.	L2 IN.	h IN.	Wt./Ft. LB.
.6	5.00	8.5	125	1.4
.8	6.33	6.0	↑	1.06

Figure 4-6^ Composite Rotor Trailing Edge Fairing

4.3.5.8 Materials of Construction

Table 4-54 lists the recommended materials. The E-glass reinforcements, epoxy resin system and structural epoxy adhesive are the same as used on the 150 ft. composite blade.

These materials are used in a patent-pending, modified filament winding process called the TFT process, which can produce a composite with highly oriented properties, at low cost, in a tapered, airfoil-shaped structure. This process was used to fabricate the D-spar of the 150 ft. composite blade.

Table 4-54. Composite Rotor Recommended Materials

E-GLASS FILAMENT[1]		MANUFACTURER
0° REINFORCEMENT[2] TFT	STYLE D360 WEFT UNIDIRECTIONAL 36 OZ. PER SQ. YD.	PROFORM
±45° REINFORCEMENT[2] TFT	STYLE DB-240 45° BIAS TAPE, 24 oz. PER SQ. YD.	PROFORM
90° REINFORCEMENT	CONTINUOUS ROVING (OCF 410 AA-450) 450 YDS/LB.	OWENS-CORNING FIBERGLASS
RESIN SYSTEM[1]	DER 332 RESIN (80 PBW) RD-2 DILUENT (20PBW) TONOX 6040 HARDENER (22.5 PBW) (CURE TEMPERATURE: 180°F, 5 HOURS; 250°F, 5 HOURS)	DOW CHEMICAL CIBA/GEIGY UNIROYAL
ADHESIVE[1]	EPON 913 PASTE EPOXY	SHELL CHEMICAL
SYNTACTIC FOAM	No. 7018	FIBER-RESIN
PAINT	MIL-C-81773 POLYURETHANE	ADVANCED COATINGS
JOINT REINFORCING SHIMS	CRES (CLEANED AND PRIMED)	COMMERCIAL

(1) ALSO USED ON 150 FT. BLADE PROGRAM
(2) SCI PATENT PENDING

4.3.6 Structural Analysis

4.3.6.1 Material Properties

The proposed E-glass composite has typical tensile strength of about 50 ksi. The blade design, however, based on stiffness, fatigue, and compression buckling requirements, results in tensile stresses less than 10 ksi. Consequently, attention is focused on the stiffness and fatigue properties of the material rather than the tensile strength.

The stiffness properties of the composite laminate vary with the lay-up and are shown for the proposed materials and proposed glass content in Figure 4-61. The winding process requires that about 10 percent of the filaments are hoop (90^o) windings, for proper compaction of other filaments. The proposed MOD-2 composite will have about 70 percent of the filaments oriented axially (0^o), and 20 percent at $\pm 45^o$ orientation. These proportions will vary slightly along the span as the total wall thicknesses taper. When the webs are thickened near joints and discontinuities, the percentage of 45^o

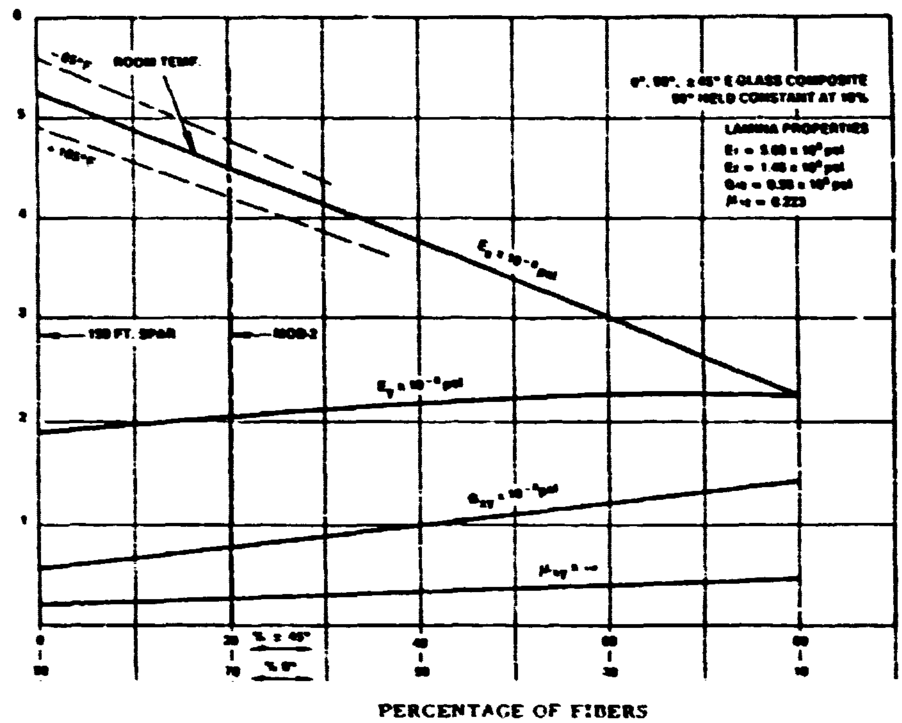


Figure 4-61. Composite Rotor Predicted Elastic Properties

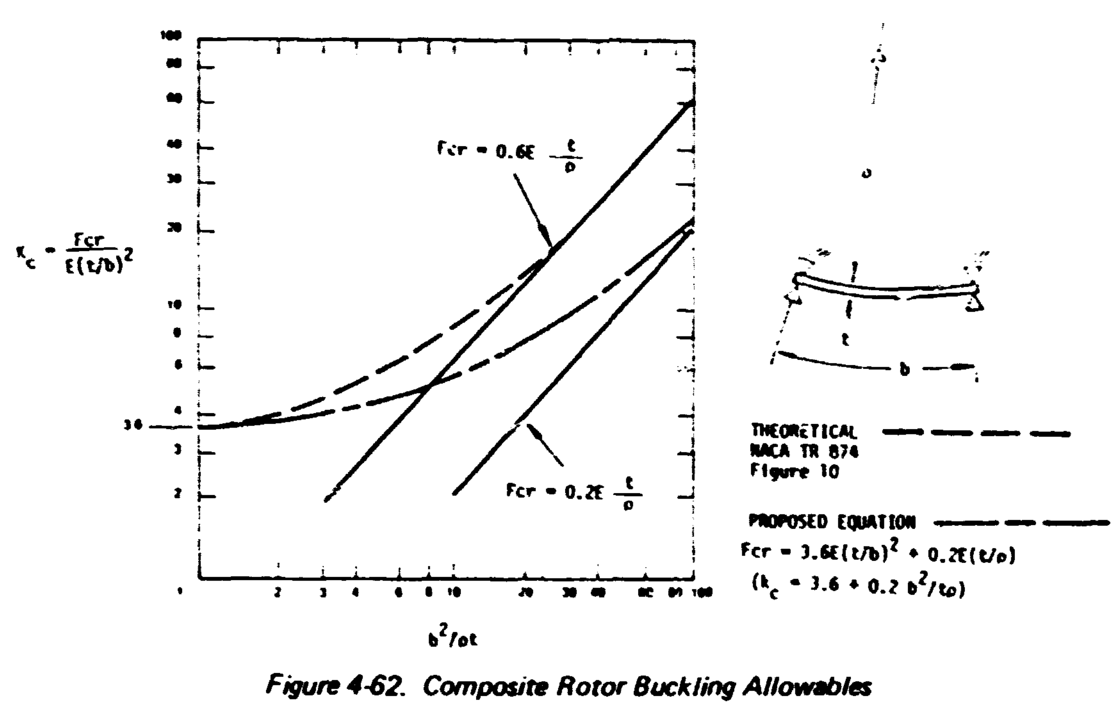


Figure 4-62. Composite Rotor Buckling Allowables

filaments will be increased to permit shear distribution of peak loads. The properties shown in Figure 4-61 apply for the various regions, and have been calculated from the lamina properties shown, considering only forces and deformations in the plane of the web. The 45^o fibers will be near the web surfaces in order to stiffen a web element in torsion and chordwise flexure.

4.3.6.2 Buckling Allowables

The buckling strength of a curved isotropic plate is difficult to predict, and an orthotropic composite material presents even more problems. For preliminary design it is necessary to use simple, conservative equations which may be verified later by tests and more refined analysis. The first equation shown in Figure 4-62 is for an isotropic plate. A flat rectangular plate has the predictable buckling stress F_{cr} of $3.6\ E\ t^2/b^2$. For a plate curved to a radius p, the theoretical value of $0.6\ E\ t/p$ which is found from linear small deflection analysis is optimistic, and has been arbitrarily reduced to one-third that value in Figure 4-62. The effective modulus for calculating buckling stresses in orthotropic plates is approximately $E = \sqrt{E_L E_T}$ where E_L and E_T are equivalent values representing longitudinal and transverse flexural stiffness of plate elements. A conservative value E of 2000 ksi has been used, but a value of 3000 ksi seems feasible with careful design of the lay-up. The allowables, thus derived, are considered to be quite conservative.

4.3.6.3 Fatigue Allowables

In order to derive fatigue allowables for the composite blade, SCI referred to the data from tension - tension fatigue tests run by NASA LERC on small sections cut from an SCI subscale spar (Figure 4-63). The material was representative of the 150 foot blade TFT/Epoxy spar. The maximum number of cycles to failure for any of the specimens tested was 6×10^6. The MOD-2 rotor is designed for 2×10^8 revolutions, which produce 2×10^8 cycles of loading for one per rev loads. The NASA fatigue test data was extrapolated to the higher number of cycles necessary in MOD-2 design by best-fitting a curve to the S-N data using the procedure described in ASTM D-2992, Appendix X1. It was conservatively assumed that all tests were run at a mean stress of 9000 psi, the minimum which was actually applied in the NASA tests. A lower confidence limit for one of the data points on the S-N curve was established by the method given in ASTM D-2992, Appendix X2. A lower confidence curve was drawn through this point and parallel to the nominal curve. The effect on fatigue allowable of a flaw in the material was estimated from tests conducted at NASA LERC by J. Faddoul on notched samples of materials for WTG blade structures. Based on these data, a "knockdown factor" of 15% was applied to the lower confidence limit curve to obtain the fatigue allowable for a flawed specimen. In Figure 4-63, as may be seen, the allowable stress of a flawed specimen for 2×10^8 cycles is 4.08 ksi cyclic stress at a mean stress of 9 ksi. For other mean stresses, a Goodman diagram was constructed as a straight line from the allowable at 2×10^8 cycles to a point representing a reduced static strength of 33.5 ksi at R=1 (R= Smin/Smax). This Goodman diagram and the equation describing it are shown in Figure 4-64.

The composite blade fatigue allowables were substantiated during the MOD-2 P.D. effort by fatigue testing conducted by Boeing on test specimens prepared at SCI using their TFT process. Both a notched and un-notched specimen were

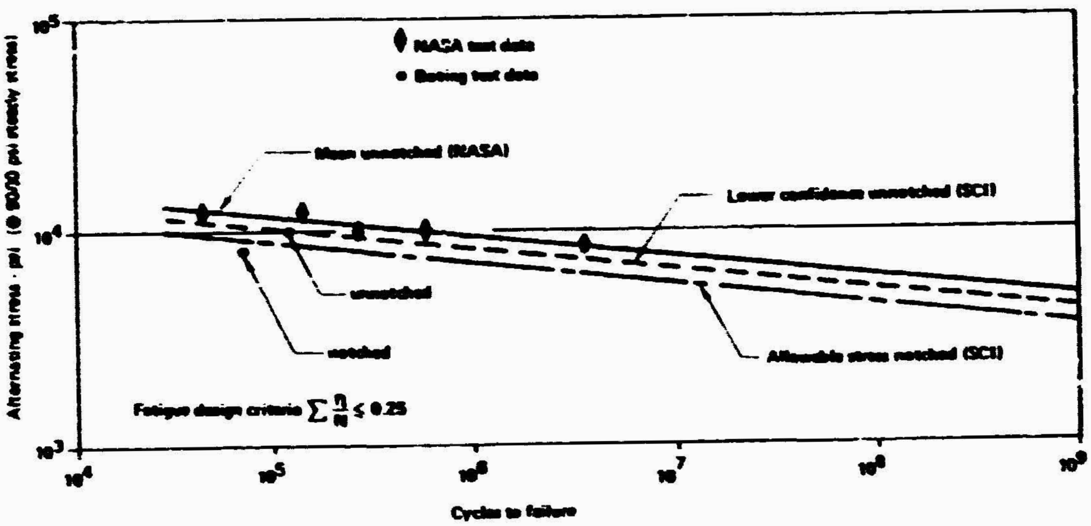


Figure 4-63. Fiberglass Fatigue Allowable Data

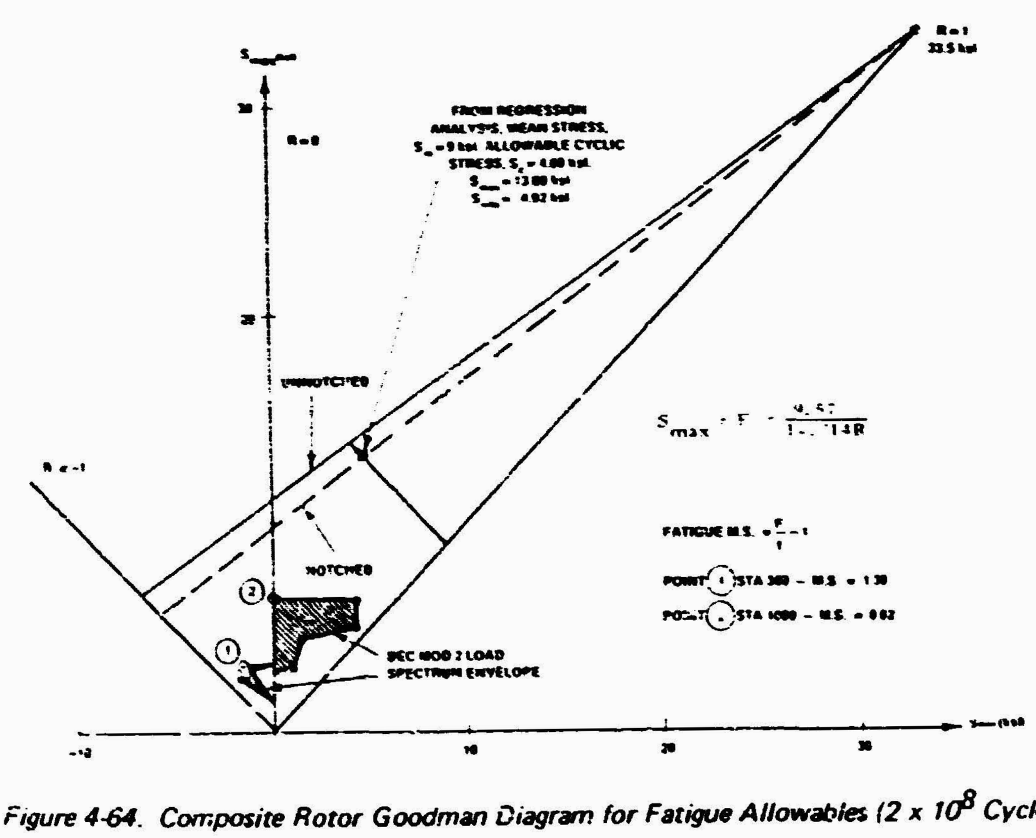


Figure 4-64. Composite Rotor Goodman Diagram for Fatigue Allowables (2 x 10^8 Cycles)

tested. The test section of the specimens were five inches long, two inches wide, and 0.4 inches thick. The flawed specimen had a 3/8 inch diameter hole drilled through the center of the specimen. The specimens were tested to the fatigue spectrum with the amplitude selected to produce a failure at 8×10^7 cycles at loading. Test results were adjusted to provide life equivalent to the 9000 psi mean stress of the NASA test data. These two data points are plotted on Figure 4-63 and substantiate the fatigue allowables. It should be noted that the notched specimen was plotted based on the section gross area stress. The flaw (a 3/8 inch diameter hole) actually results in a net area stress 19% greater than as plotted.

4.3.6.4 Stresses and Margins of Safety

Buckling allowable stresses and the typical distribution of minimum margin for buckling stress is shown in Figure 4-65. All points on the cross-section are occasionally in compression. During feathering, the upwind surface is in compression and the blade bends away from the tower. Points on the leading edge are in tension when the blade is ascending and in compression when the blade descends, while points on the trailing edge have the opposite stresses from dead-weight bending of the blade. In general, webs for the outer half of the span are designed for buckling stress, and have a margin of safety of about 0.5 with a factor of safety of 1.35, so should resist about double the limit loads without buckling.

Typical fatigue stresses are shown in Figure 4-66 for a point near the trailing edge on the upwind face of the blade. Approximately three-fourths the magnitude of the cyclical stresses at this point result from gravity loads, which occur 2×10^8 cycles. The airloads are also assumed to occur this often. Since they represent only a small part of the total stress, this assumption is reasonable. The margin of safety, based on the limit load stress, is about 1.0, or the blade should provide 30 year life at twice the applied stress.

A more detailed analysis of the fatigue stress spectrum was performed for two points on the blade using the postprocessor written for the MOSTAB program. Results are shown by the cross-hatched areas of Figure 4-64. Each of the points on the shaded boundaries represent one loading condition at one point on the blade. If the conservative assumption is made that all fatigue loading conditions occur for 2×10^8 cycles, the condition representing the most serious fatigue damage is the point which is closest to the allowable fatigue stress. In a more refined analysis, different Goodman diagrams would be used for each condition, and cumulative damage calculated by a method similar to Miner's procedure. The preliminary analysis, however, indicates that the composite blade is conservatively designed for fatigue stresses.

In summary, the preliminary design analysis indicated no stress limitations in either buckling or fatigue. The structure was intentionally designed conservatively in order to permit low-cost manufacture. Stiffness to limit airload deflection and to increase vibration frequencies was provided by heavier root structure. The outer blade was kept light in order to reduce gravity bending and centrifugal forces.

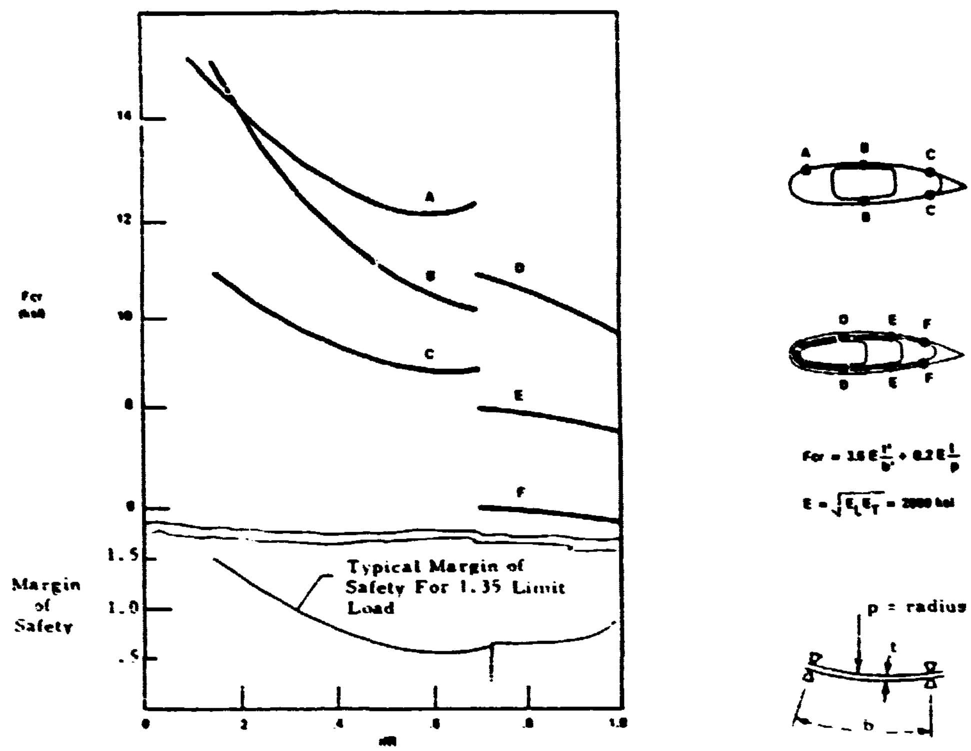


Figure 4-65. Composite Rotor Allowable Buckling Stress

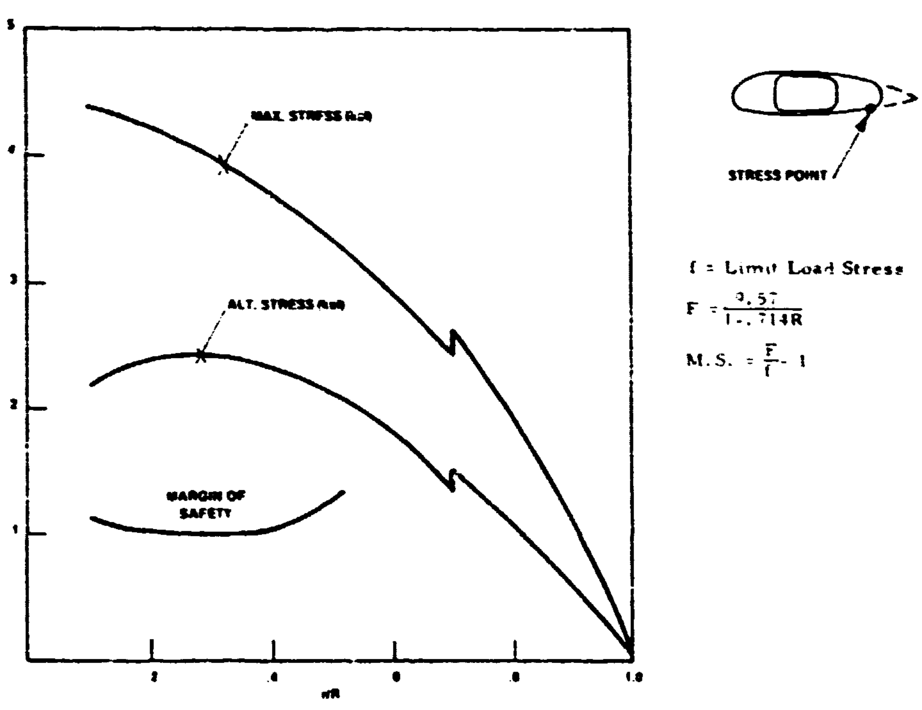


Figure 4-66. Composite Rotor Typical Fatigue Stresses

4.3.6.5 Joint Attaching Blade to Hub

The composite blade must be attached to the steel hub during field erection as shown by the joint detail of Figure 4-58. Because of the severe fatigue environment, it is desirable to transfer loads to the composite through adhesive bonds. Preloaded bolts are also used, with capability of resisting the complete load without the adhesive bond. With each system conservatively designed, there is little possibility of any type of progressive failure. Metal shims are wound in the composite to provide bearing stress capability and to increase the elastic modulus of the composite.

The joint stresses shown in Table 4-55 are first calculated for the bonded joint to resist all loads without bolts. All stresses are low, and stress concentrations are low because of the long (22 inch) scarf joint. The axial strain of the composite is matched to that of the steel, so bond stresses do not peak at the ends as they do for lap joints of untapered members.

Similarly, the bolted joint is analyzed for the entire applied load, neglecting bond strength. All stresses are low, giving large margins of safety.

Table 4-55. Composite Rotor Joint Stress Summary

BONDED JOINT - (NEGLECTING BOLTS)	ALLOWABLE F (KSI)	STRESS f (KSI)	MARGIN OF SAFETY (F/f-1)
MAXIMUM LOADS			
TENSION (HUB)	70	10.8	5.5
BOND SHEAR	3.2	.55	4.8
BOND TENSION (PEEL)	1.0	.077	12.0
ALT. LOADS (FATIGUE)			
TENSION (HUB)	15	3.4	3.4
BOND SHEAR	.8	.175	3.6
BOLTED JOINT (NEGLECTING BOND)			
BEARING (HUB)	120	34	2.5
BOLT SHEAR	75	20.7	2.6
TENSION (HUB)	70	15.9	3.4

CONCLUSIONS: LARGE SAFETY MARGINS AND REDUNDANT LOAD PATHS PERMIT JOINT QUALIFICATION BY ANALYSIS

4.3.7 Manufacturing

Manufacturing methods for the MOD-2 composite blade were chosen for low cost serial production, at the rate of one rotor per day, of a configuration which meets the design requirements and low cost goals of the program. The approach selected makes use of fabrication experience gained in the 150 ft. Composite Blade Program by the commercial composites industry, in addition to aerospace composites fabrication technology.

4.3.7.1 T.F.T.* Winding

The basic TFT process derives from a process used in the commercial filament-wound pipe industry. This process was conceived and reduced to practice by SCI in anticipation of the 150 ft. Composite Blade Program. It was used to fabricate the 142 ft. long, 20,000 lb. D-spar, which was the primary structural element of the blade and comprised 85% of the composite weight.

Figure 4-67 shows the process schematically. Note that the 150 ft. spar used a rotating mandrel with a steady rest, in a lathe-type winding machine, while the proposed MOD-2 process uses a stationary mandrel in a ring winder* machine.

The stationary mandrel eliminates the mandrel deflection and fatigue problems encountered in the 150 ft. Blade Program and two previous NASA composite blade programs (the 60 ft. Blade and MOD-1). Stationary mandrels facilitate fabrication of the multiple-spar blades like the MOD-2, since the chordwise center of gravity of the mandrel stack has no effect on the rotating ring element. In addition, the mandrels are held in place by gravity (except the inner blade leading edge mandrel) and are in their stiffest orientation, with chordlines vertical.

A structural analysis of the inner blade mandrels was performed. A maximum mandrel static stress of 6500 psi and deflection of 1.12 inches are predicted, as compared with 6 inches deflection and 14,000 psi reversing stress for the rotating mandrel used for the 150 ft. blade spar.

4.3.7.2 Inner Blade Fabrication Sequence

Figure 4-68 shows the inner blade fabrication sequence in schematic form. The box spar mandrel is set up in the ring winder, with the hub and spindle box, as in Figure 4-67 and 4-69. Note the bucking ring and jacking ring arrangement in the latter figure, where the winding ring has been removed for clarity.

Box Spar Winding

In the first winding step, this mandrel is TFT wound to form the box spar. The leading and trailing edge mandrels are added and faired to the box spar with splined syntactic foam.

Blade Shell Winding

In the second winding step, the outer blade shell is wound over the uncured box spar and leading and trailing edge mandrels. As shown in Figure 4-70, the spindle box and hub joint area are overwound in this same step. The wire mesh lightning protection and molded trailing edge fairing is then added to the wet wound assembly.

* Patent Pending

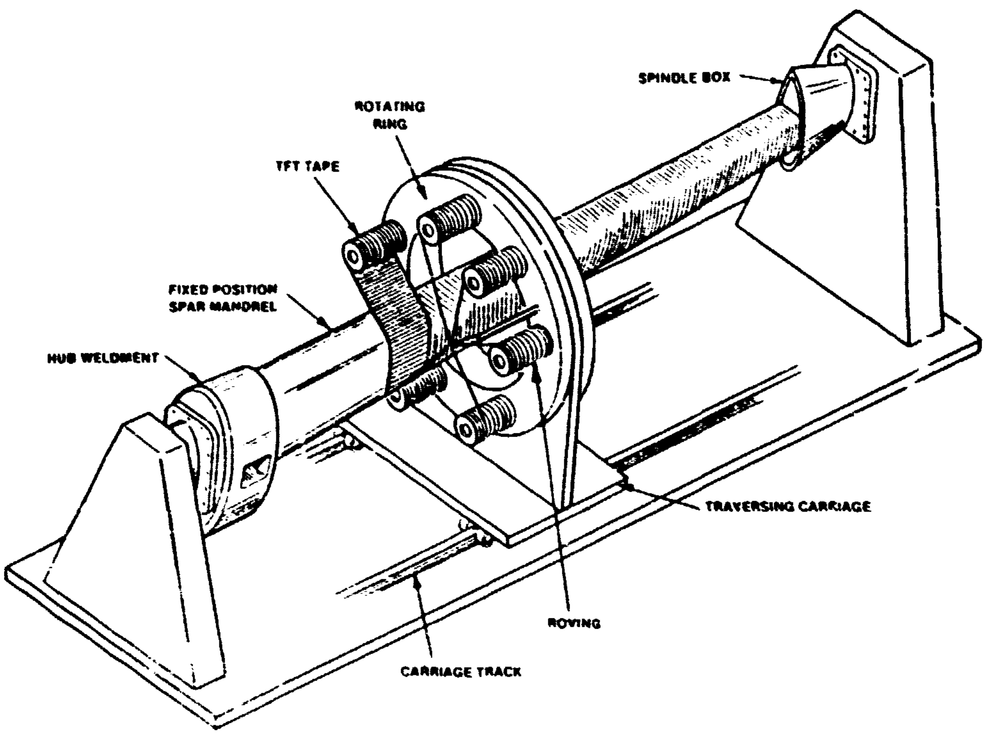


Figure 4-67. Composite Rotor Ring Winder Concept *

* Patent Pending

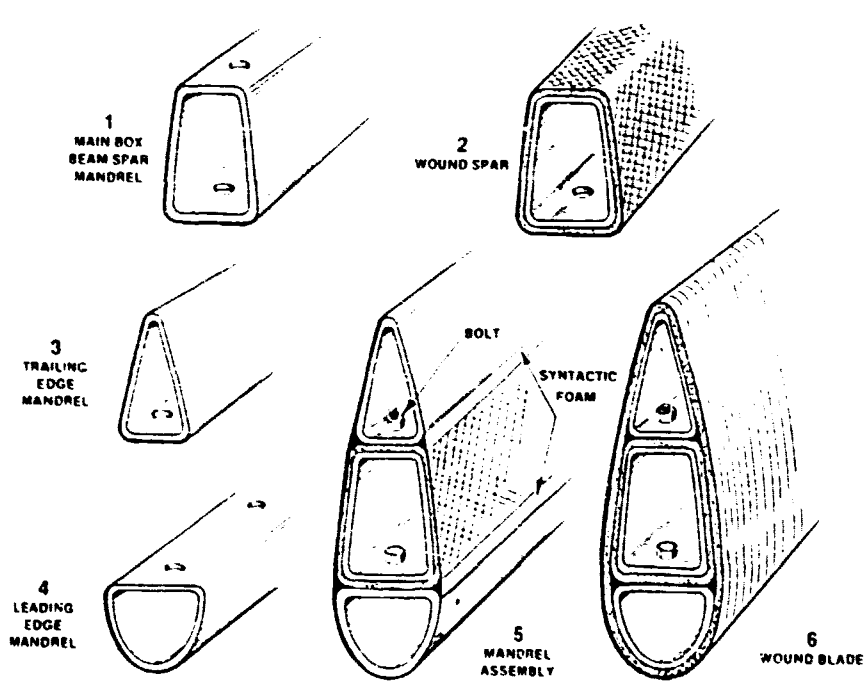


Figure 4-68. Composite Rotor inner Blade Assembly

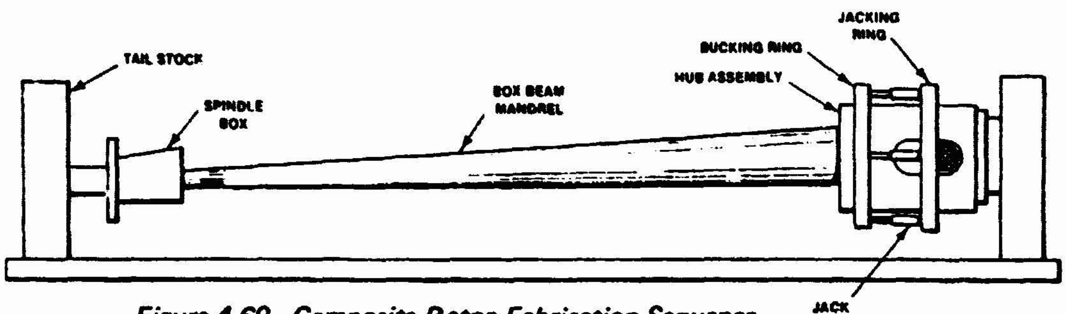


Figure 4-69. Composite Rotor Fabrication Sequence

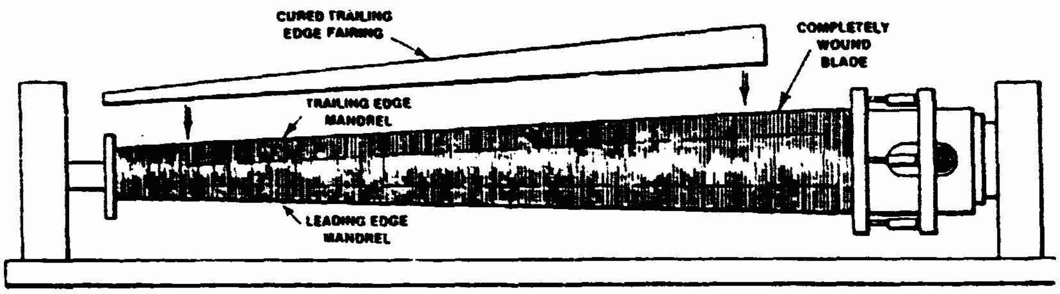
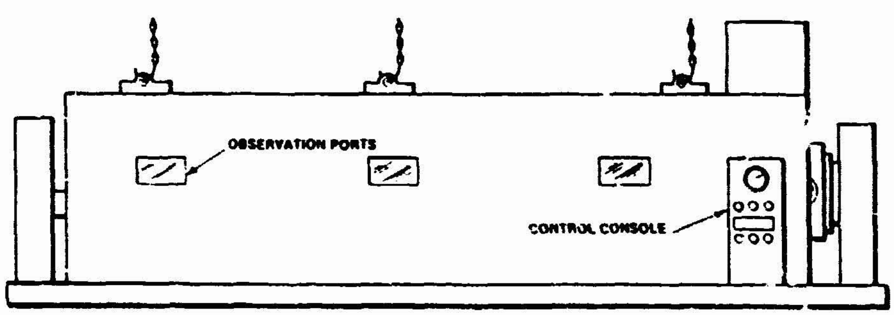
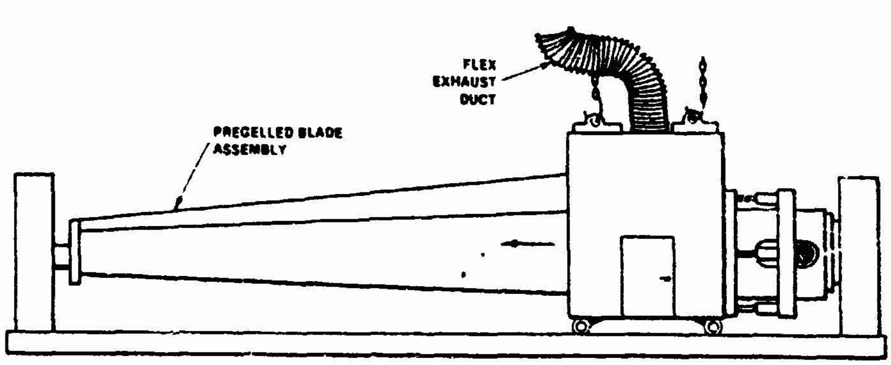


Figure 4-70. Composite Rotor Fabrication Sequence



PREGEL AND CURING OVEN

Figure 4-71. Composite Rotor Curing Oven



FINISH AND PAINT UNIT

Figure 4-72. Composite Rotor Finish and Paint Unit

Painting and Curing

A portable oven is placed over the assembly as in Figure 4-71 to gel the resin. The blade is then painted as in Figure 4-72 and is returned to the oven to cure.

Freeing Mandrels

After cooling, the blade is supported on wheeled dollies and the winding machine tailstock is removed. The fasteners which connect the mandrels to the spindle box are removed, as are those connecting the leading and trailing edge mandrels to the box spar mandrel and hub. As shown in Figure 4-73, the hydraulic jacks are actuated to free the box spar mandrel. Flexible links are installed and the jacks are again actuated, as in Figure 4-74 to free the leading and trailing edge mandrels.

Mandrel Extraction

A winch cable is attached to the spindle box, as shown in Figure 4-75, and the blade is pulled clear of the mandrels. Recessed mandrel supports rise vertically to take the weight of the mandrel as it is exposed by the moving blade.

4.3.7.3 Tip Blade Fabrication Sequence

The tip blade fabrication sequence, shown in Figure 4-76, differs from the inner blade sequence mainly in the way the mandrels are stacked up during winding. The three tip mandrels are too small to allow internal access for bolt installation and removal. However, bolts are not necessary, since gravity holds the two afterbody mandrels in place on top of the leading edge D-spar mandrel. The mandrels are located and aligned by soft shear pins, which merely shear off when the mandrels are extracted.

4.3.7.4 The Hub and Shaft Receptacle Installation

The tip section metal shaft receptacle is used as a winding mandrel in a manner analogous to the way the hub is used on the inner blade. Both are covered with a release film, temporarily bolted to the mandrels and are removed from the finished blade section along with the mandrels. They are then reinstalled by bolting and bonding. However, the hub joint is made up in the field, while the shaft receptacle is installed at the manufacturing plant. Note that the hub section must be returned to the winding machine for winding of the second inner blade section. This is shown in the production flow sequence, Figure 4-77.

4.3.7.5 Trailing Edge Fairing

Figure 4-78 shows the fabrication of the trailing edge fairing by hand layup. Installation on the blade is illustrated in Figure 4-79. (See also Section 4.3.5.6).

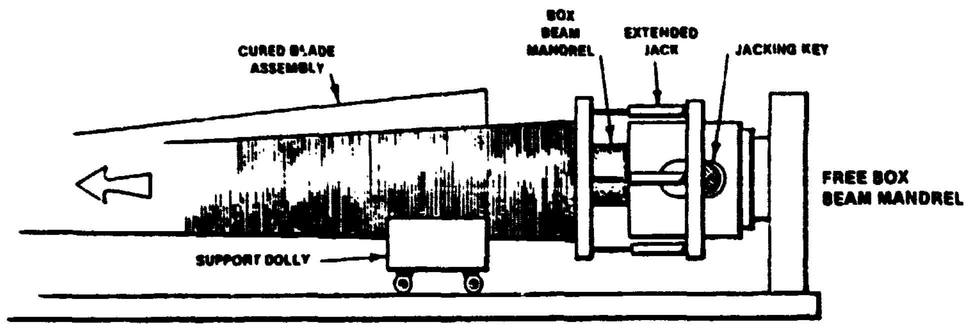


Figure 4-73. Composite Rotor Mandrel Extraction

4.3.7-8

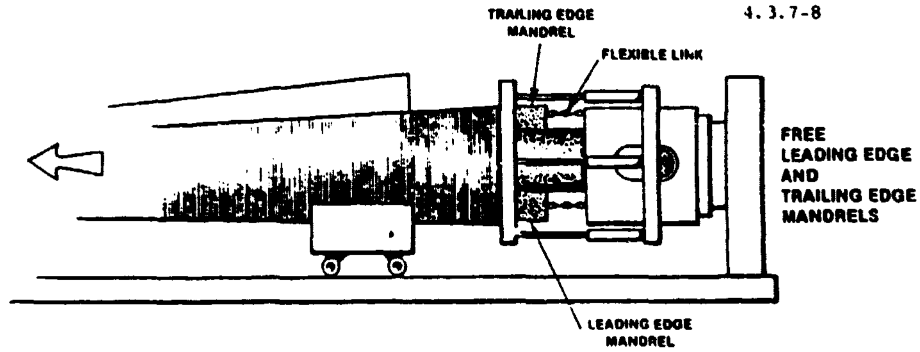
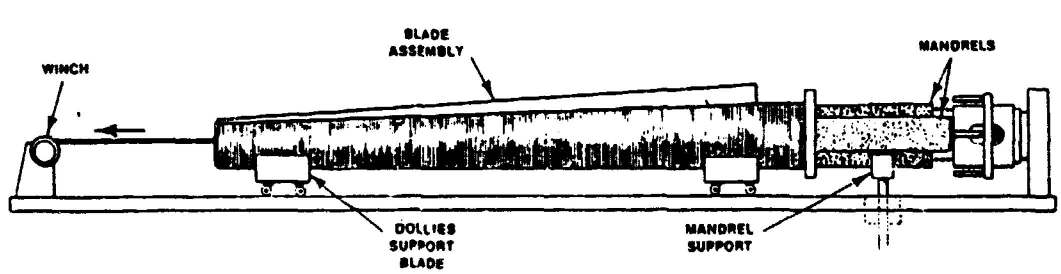


Figure 4-74. Composite Rotor Mandrel Extraction



COMPLETE EXTRACTION WITH WINCH

Figure 4-75. Composite Rotor Mandrel Extraction

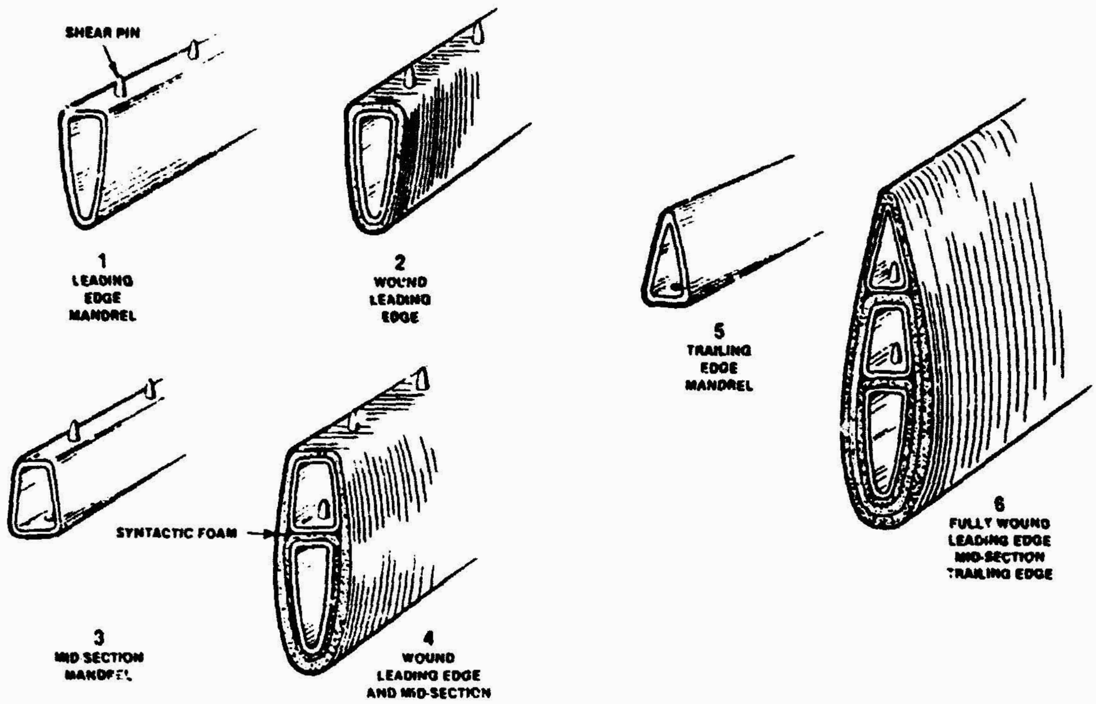


Figure 4-76. Composite Rotor Tip Blade Assembly

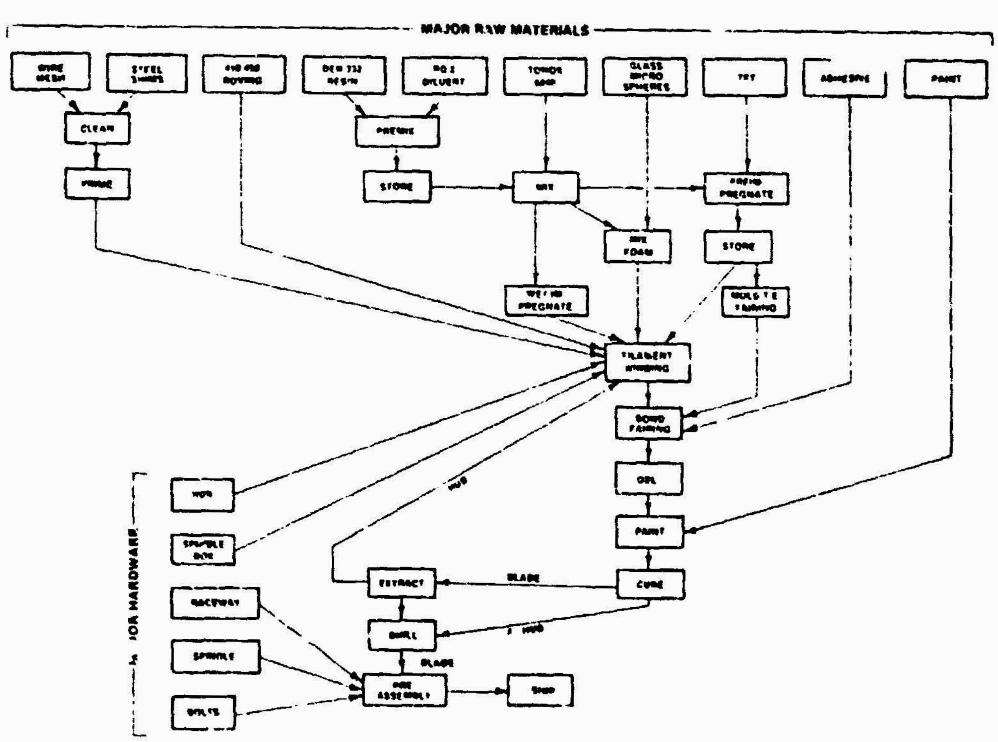


Figure 4-77. Composite Rotor Production Flow Sequence

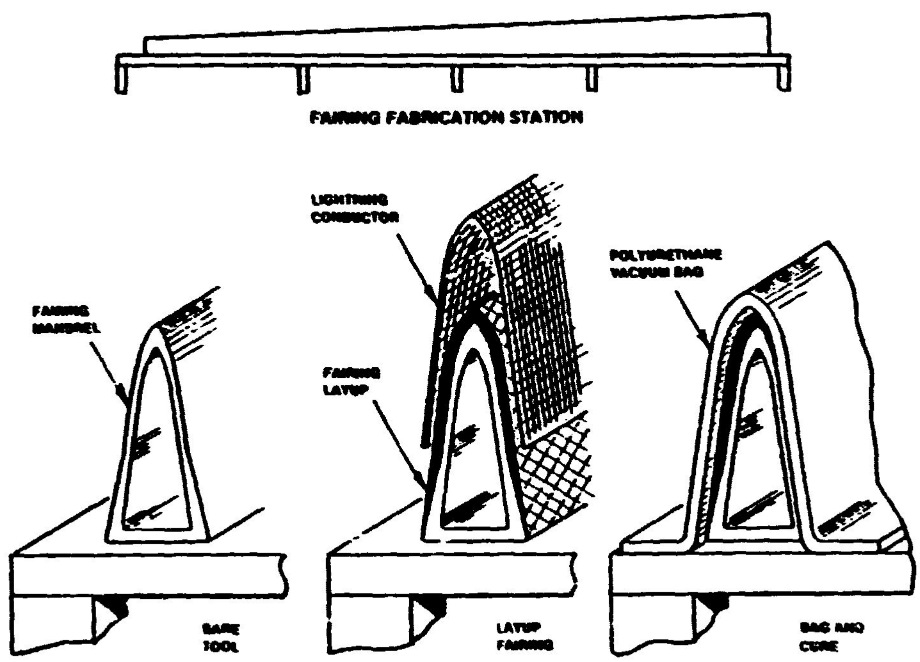


Figure 4-78. Composite Rotor Trailing Edge Fabrication

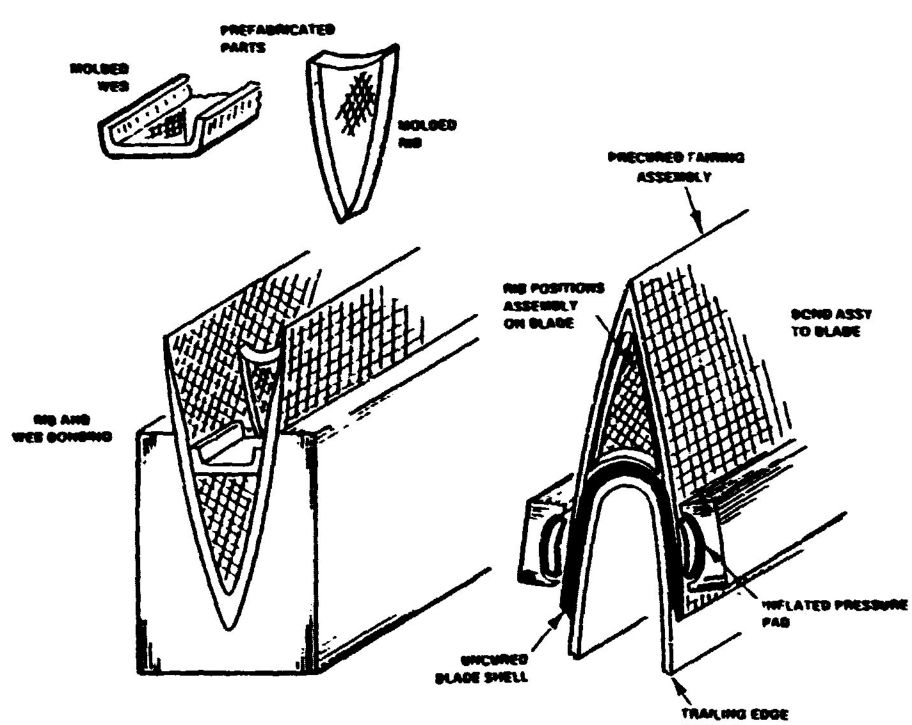


Figure 4-79. Composite Rotor Trailing Edge Assembly

4.3.7.6 Finishing, Final Assembly

Referring to Figure 4-77 the major finishing steps include match drilling of the hub joint, installation of the electrical and hydraulic components, and preassembly of the rotor at the plant for dimensional and functional checks.

4.3.7.7 Production Plant Layout

Figure 4-80 shows a proposed plant layout for composite fabrication and rotor assembly. It is assumed that the metal parts, hardware and electrical and hydraulic components would be outside purchase. This plant would have a capacity of two blades (one rotor) per day. Note that two identical mirror image modules are proposed. For lower production rates, one module could be used. For very low rates (one rotor per week or less), the same winding machine could be used for both inner and tip blade sections. Figure 4-81 shows this production plant layout incorporated into a fully equipped, independent manufacturing plant.

4.3.7.8 Shipping

Figure 4-82 shows the proposed arrangement for shipping an entire rotor on two flatcars. Local movement, from railhead to WTG site, will be accomplished by truck. The 142 ft. spar for the 150 ft. blade was shipped in this manner from Mira Loma, California to Bloomfield, Connecticut.

4.3.7.9 Quality Assurance

Tables 4-56, 4-57, & 4-58 list proposed Quality Assurance provisions for the MOD-2 Composite Blade. The tables list the various tests and inspection steps under the headings of Receiving Inspection, In-Process Inspection and Assembly Inspection.

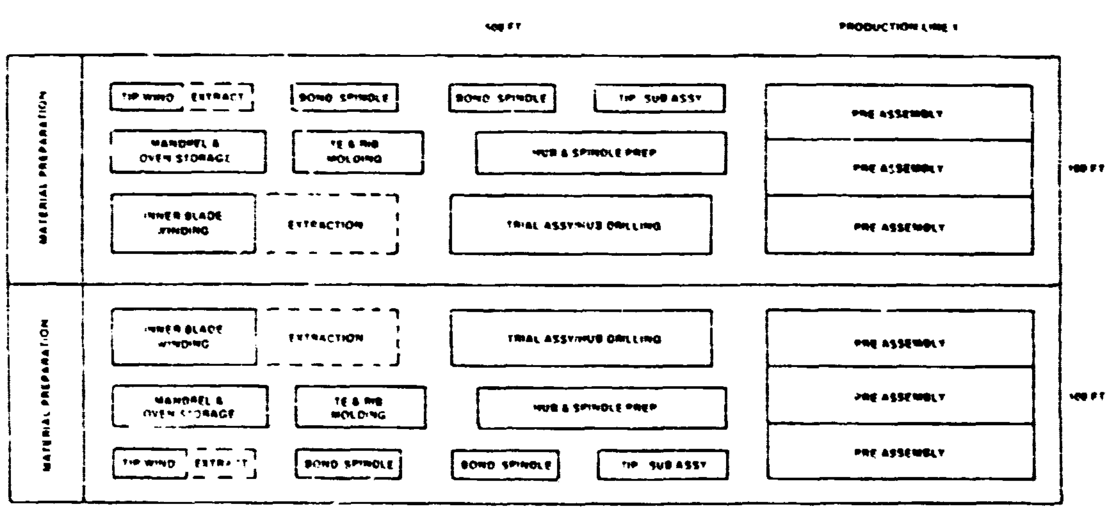


Figure 4-80. Composite Rotor Production Line Arrangement

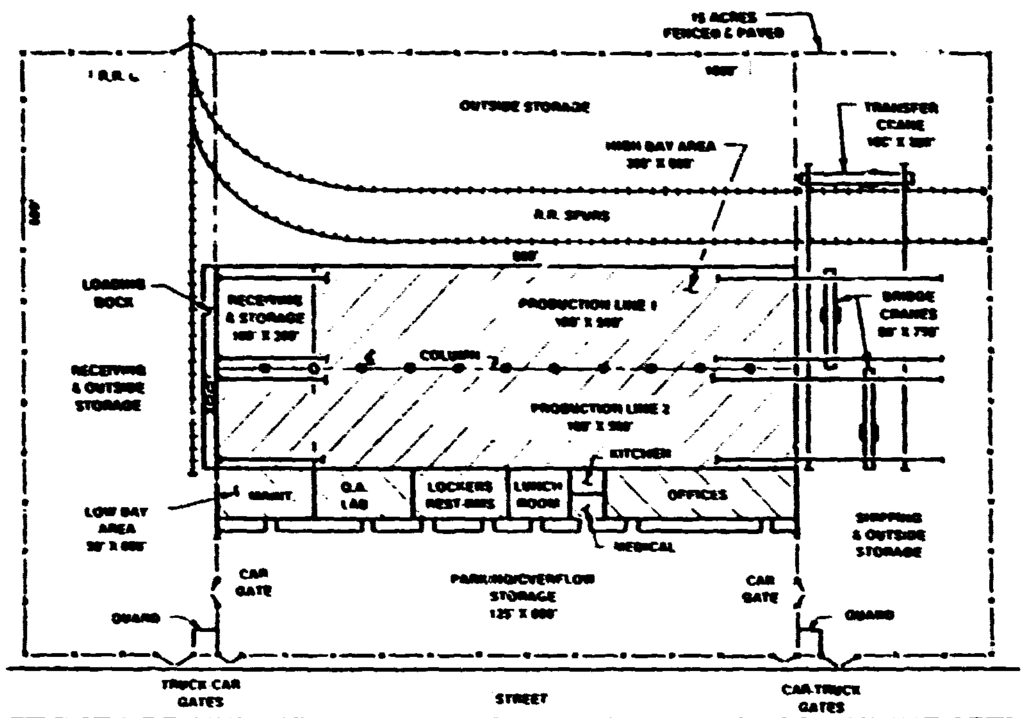


Figure 4-81. Composite Rotor Proposed Production Plant

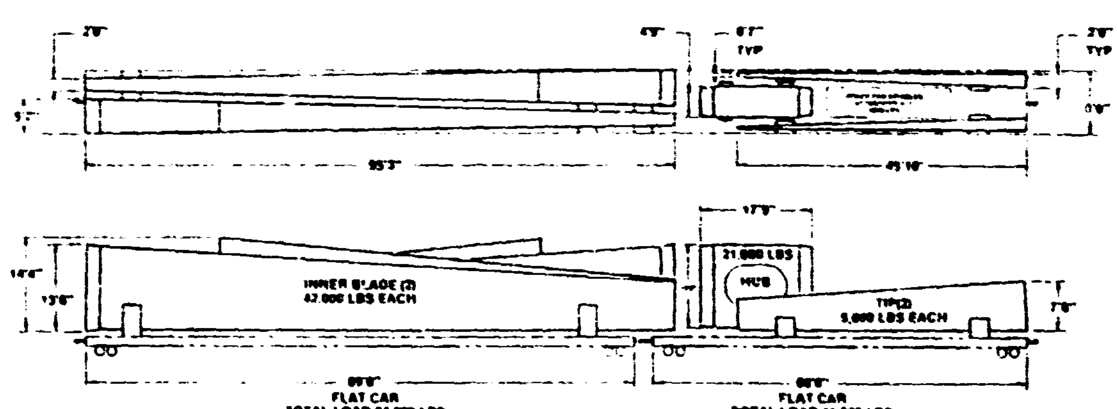


Figure 4-82. Composite Rotor Proposed Shipping Arrangement

Table 4-56. Quality Assurance Receiving Inspection

RESIN SYSTEM	REINFORCEMENTS
VISCOSITY	YIELD - WEIGHT/YD
GEL TIME	TENSILE STRENGTH
SPECIFIC GRAVITY	MOISTURE CONTENT
BARCOL HARDNESS (CASTING)	SIZING CONTENT
CERTIFICATION TO MIL-R-9300	END COUNT
PREIMPREGNATED MATERIAL	DIMENSIONAL CONFORMANCE
RESIN CONTENT	CERTIFICATION TO MIL-R-60346
INTERLAMINAR SHEAR - CURED	**HARDWARE**
BARCOL HARDNESS - CURED	DIMENSIONAL
VISUAL	MATERIAL CERTIFICATIONS
STORAGE CONTROL	WELDING INSPECTION/CERTIFICATION
SUPPORT MATERIALS	
COATINGS - CERTIFICATION	
ADHESIVES - CERTIFICATION	
SHIMS - CERTIFICATION	

Table 4-57. Quality Assurance In-Process Inspection

POINTS AS INDICATED IN MANUFACTURING SHOP ORDER	
TOOLING PREPARATION	**WINDING PATTERN**
MANDREL SURFACE - RELEASE	BAND PLACEMENT
RESIN FORMULATION	WIND ANGLE
METERING DEVICE ACCURACY	LAYER COUNT
PREPRODUCTION GEL TIME	TERMINATION POINTS
CONSUMPTION BY WEIGHT	VISUAL QUALITY
PREIMPREGNATED MATERIALS	**TRAILING EDGE FAIRING ASSEMBLY**
RESIN CONTENT	FABRICATED PART ACCEPTANCE
VISUAL/DIMENSIONAL CONFORMANCE	APPLICATION OF ADHESIVE
	APPLICATION OF PRESSURE JIG
FILAMENT WINDING	**COATING APPLICATION**
PREPREG ACCEPTANCE VERIFICATION	VERIFY PREGEL
ROVING ACCEPTANCE VERIFICATION	COATING ACCEPTANCE VERIFICATION
TENSION	COATING APPLICATION
BAND WIDTH AND UNIFORMITY	**CURE CYCLE**
COMBINED BAND YIELD	
COMBINED BAND RESIN CONTENT	COMPOSITE TIME/TEMPERATURE PLOT
CONSUMPTION BY WEIGHT ASSEMBLY	OVEN TIME/TEMPERATURE PLOT
ASSEMBLY	CURED HARDNESS

Table 4-58. Quality Assurance Assembly Inspection

POINTS AS INDICATED IN MANUFACTURING SHOP ORDER
CURED BLADES
DIMENSIONAL CONFORMANCE
THICKNESS CONFORMANCE
CENTER OF GRAVITY AND TOTAL WEIGHT
SURFACE QUALITY
WITNESS SAMPLES - TEST
COMPONENT ASSEMBLY
HUB ASSEMBLY HOLE PATTERN/BOND
SPINDLE BOX INSTALLATION/BOND
TIP INSTALLATION
RACEWAY INSTALLATION
HYDRAULICS INSTALLATION
ELECTRICAL INSTALLATION
FAIRING CLOSEOUT INSTALLATION
FINAL INSPECTION
LIGHTNING SCREEN CONDUCTIVITY
ELECTRICAL CIRCUIT CHECK
HYDRAULIC CIRCUIT CHECK
ASSEMBLED WEIGHT
ASSEMBLY CENTER OF GRAVITY
SURFACE QUALITY

4.3.8 MOD-2 System Compatibility

The composite rotor can be utilized on the MOD-2 WTS with very minor or no modification to the system as designed with the steel rotor. This is primarily due to the fact that the composite rotor is designed to meet the frequency requirement. As discussed in Section 4.3.5, the blade chord and depth was increased over the steel blade dimensions from the 0.5 span to the 0.05 span to provide the necessary stiffness to achieve the frequency requirement. This stiffness results in blade tip deflections and tower clearance as shown in Table 4-59. As noted, the composite blade has a tip deflection of 23.6 inches greater than the steel blade. This added deflection poses the problem of adequate clearance with the tower. Although the current estimate of the composite rotor tower clearance is 13.8 inches, the study recommends that the clearance be specified as the same as the current estimate of the steel rotor clearance (37.4 inches). This 23.6 inch increase can be accomplished by any of the following three methods.

1. Pre-coning of the composite blade by 0.8 degrees.

   This approach has not been studied, but is estimated to have negligible effect on performance or loads. Steel or composite blades could then be interchangeable without system modification.

2. Tilt of the rotor axis approximately 0.8 degrees.

   This approach has not received detail analysis, but is a minor impact solution which could be used for either the steel or composite rotor to provide for interchangeability of rotors.

3. Extend the low speed shaft.

This approach was analyzed and was found to require a negligible impact on system design and cost as shown in Table 4-60. Because the composite rotor is lighter than the steel rotor, the only system impact is the length of the low speed shaft. This approach is a low cost modification, but would not provide for interchangeability of rotors.

Table 4-59. Rotor Tower Clearance

	DIMENSIONS IN INCHES						
ROTOR	TEETER DEFL.	FLEX. DEFL.	TIP CHORD	0.5^0 SHAFT ANGLE	TOWER MOTION	SHAFT EXTEN.	CLEARANCE
Steel	157.5	55.7	5.5	-15.7	-12.4	0	37.4
Composite	157.5	79.3	5.5	-15.7	-12.4	23.6	37.4

Table 4-60. Composite Rotor System Impact

- Nacelle Structure - No change
- Yaw System - No expected change to hardware cost
  - Bearing - Combined overturning moment reduced 8%
    - Dead weight reduced 8%
  - Drive - Limit torque increased 9%
- Drive Train
  - Low speed shaft extended - wt. increase 1300 lb.
  - Radial bearing loads essentially unchanged
- Tower - no change. Tower structure is strength designed
- Foundation - Difference insignificant

4.4 SPECIFICATION AND CONSTRAINTS SENSITIVITY

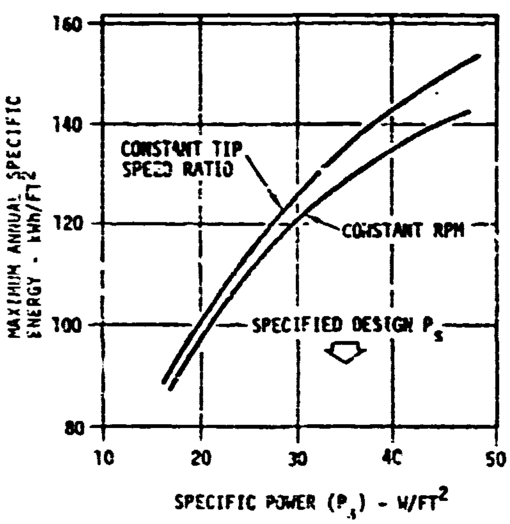
At the time of the start of the MOD-2 contract, some of the specifications and constraints listed in exhibit B of the contract were based on previous experience with smaller wind turbines. Therefore BEC evaluated each specification and constraint in exhibit B to determine if a change in a particular specification or constraint would lead to a lower cost of electricity, a longer service life, or a lesser programmatic risk. This section reports the results of the various analyses.

4.4.1 User Requirements

This section discusses the user requirements listed in exhibit B of the contract that are closely associated with operation of the wind turbines.

4.4.1.1 Control Strategy

Specifications required determination of fixed or variable speed rotor. The results were as follows (Figure 4-83):

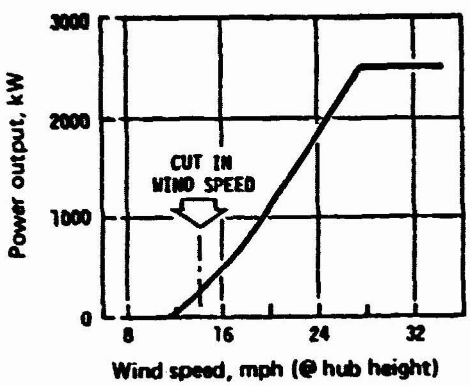


- VARIABLE SPEED ROTOR CAPTURES 6% MORE ENERGY BUT OFFERS HIGHER RISK DUE TO WIDE ROTOR SPEED RANGE AND MORE EQUIPMENT REQUIRED
- FIXED SPEED ROTOR WIND TURBINE LESS COSTLY
- VARIABLE SPEED ROTOR ADDS .57¢/kWh TO COST OF ENERGY
- FIXED SPEED ROTOR CONTROL STRATEGY SPECIFIED

Figure 4-83. Effect of Control Strategy on Energy Output

4.4.1.2 Cut-In Wind Speed

Specification required operation at reduced power between cut-in and rated wind speed. The objective of this analysis was to determine the most economical value of cut-in wind speed. The results are as follows (Figure 4-84):

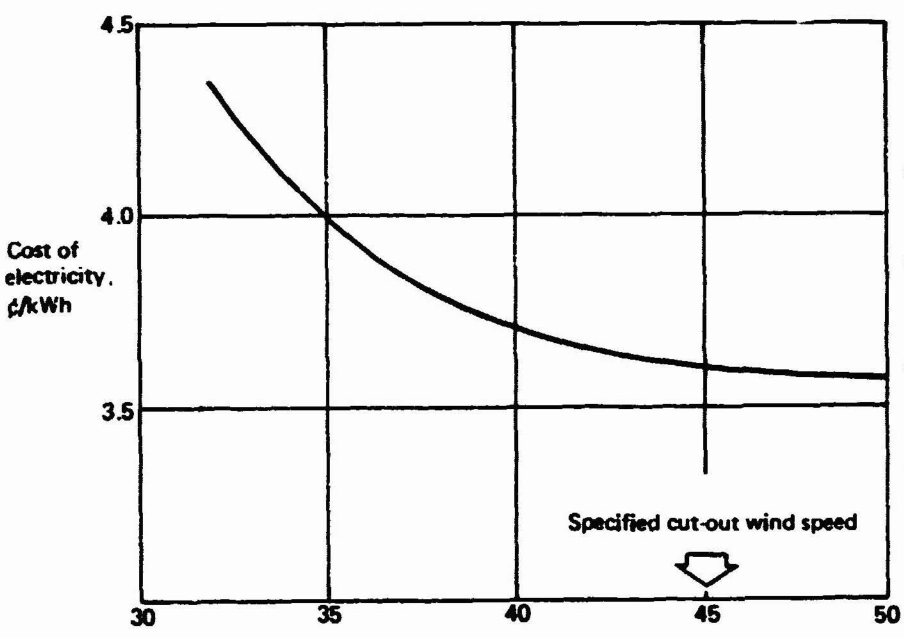


- 11 MPH IS WIND SPEED REQ'D TO SUSTAIN RATED ROTATIONAL RPM
- CUT-IN SPEED IS SET AT 2-3 MPH ABOVE 11 MPH TO AVOID FREQUENT START UP/SHUT DOWN CYCLES
- LOSS OF ENERGY DUE TO HIGHER (14 MPH) CUT-IN SPEED = 50,000 kWhr/YR
- REDUCTION IN OPERATING HOURS DUE TO 14 MPH CUT-IN SPEED = 1000 HR
- IMPROVED AVAILABILITY FROM REDUCED OPERATING HOURS OFFSETS LOSS OF ENERGY

Figure 4-84. Wind Speed Versus Power Output

4.4.1.3 Cut-out Wind Speed

The contract specification required rated power between rated and cut-out wind speed. The objective of this study was to determine the most economical value of cut-out wind speed. The results are as follows (Figure 4-85):

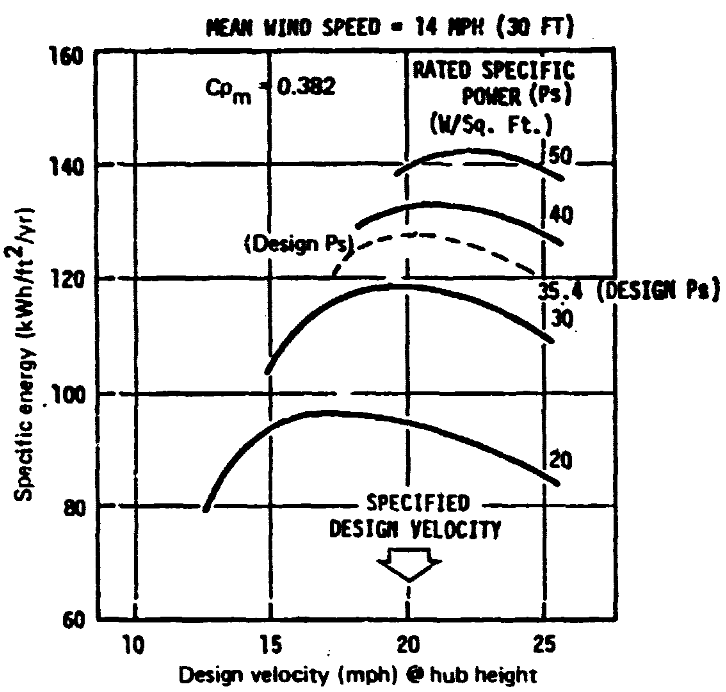


- Blade weight and cost increase small between 40-50 mph cut-out (< 1%)
- Energy output decrease between infinite cut-out speed and 45 mph speed is only .75%
- Load transients at higher cut-out windspeeds due to emergency shutdown are greater
- Cost of elctricity decrease above 45 mph is small
- 45 mph selected as cut-out wind speed

Figure 4-85. Cut-out Wind Speed Versus Cost of Electricity

4.4.1.4 Design and Rated Wind Speed

Specification required the highest efficiency between cut-in and rated wind speed. In order to satisfy this requirement, the design wind velocity was selected as that wind speed where the specific energy output (kWh/sq. ft/yr) is maximum for the design specific power rating (35.4 watts/sq. ft). Design wind velocity is that wind velocity at which the rotor produces maximum efficiency and is the basic parameter for a given rotor, which determines the rotor rpm. Separate studies showed that the cost of electricity was minimum where the specific energy was maximum. The results are summarized as follows (Figure 4-86):

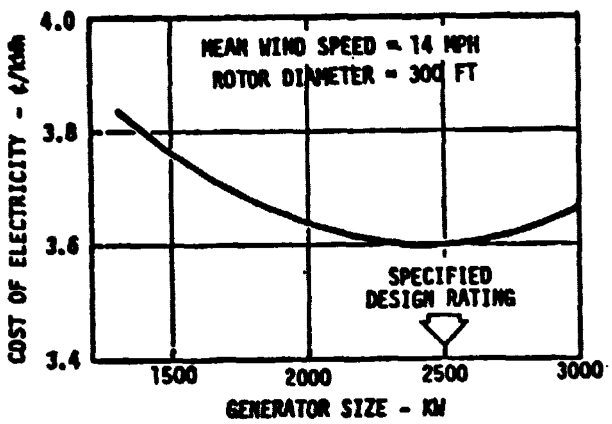


- Cost of electricity is minimum at wind speed where specific power is maximum
- The specific power rating of MOD-2 is 35.4 watts/sq/ft of rotor area and energy output peaks at wind speed of 20 mph
- Design wind speed selected at 20 mph
- Rated wind speed that is compatible with the design wind speed is 27.5 mph

Figure 4-86. Selection of Wind Speed

4.4.1.5 60 HZ Power in Megawatt Range

This specification required that MOD-2 generate three-phase, 60 HZ power in the megawatt range. Studies showed no penalty in the generation of three-phase, 60 HZ power since a majority of the available generation equipment has these characteristics as standard. The trade studies (4.2.7) showed that the most economical rating for MOD-2 in the 14 mph, mean wind spectra is 2500 kW as shown in Figure 4-87.

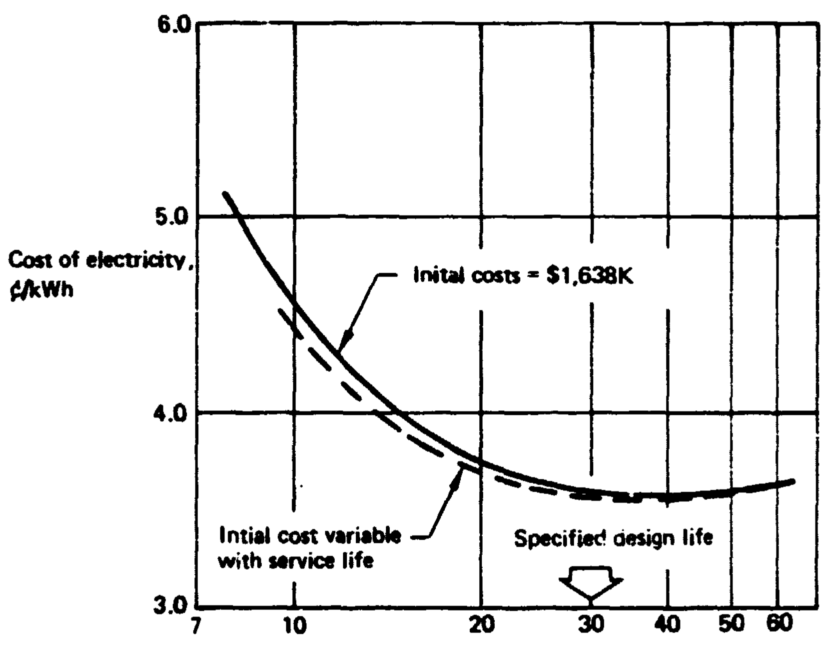


- 2500 kW is best rating
- 35.4 watts/sq. ft. is best specific power rating for rotor in 14 mph spectra
- 3 phase, 60 cycle power is most economical and adaptable

Figure 4-87. Selection of Design Power Rating

4.4.1.6 30 Years Useful Utility Service Life

This specification required that the wind turbine, including all components, be designed for a useful utility service life of 30 years. These studies showed that cost of electricity increases with decrease in service life less than 30 years and that 30 years is the optimum life. In these analyses side trade studies were made as to the effect of periodic replacement of some parts. Periodic replacement was used in all cases where the use of replaced, but lower cost parts, lowered the cost of electricity. This study is summarized below (Figure 4-88):



- Levelized fixed charge rate established by standard economics $FCR_{30\ years}$ = 18%
- Major life sensitive components are the rotor and the gear box
- Periodic replacement considered in cost assessment
- 30 year service life selected

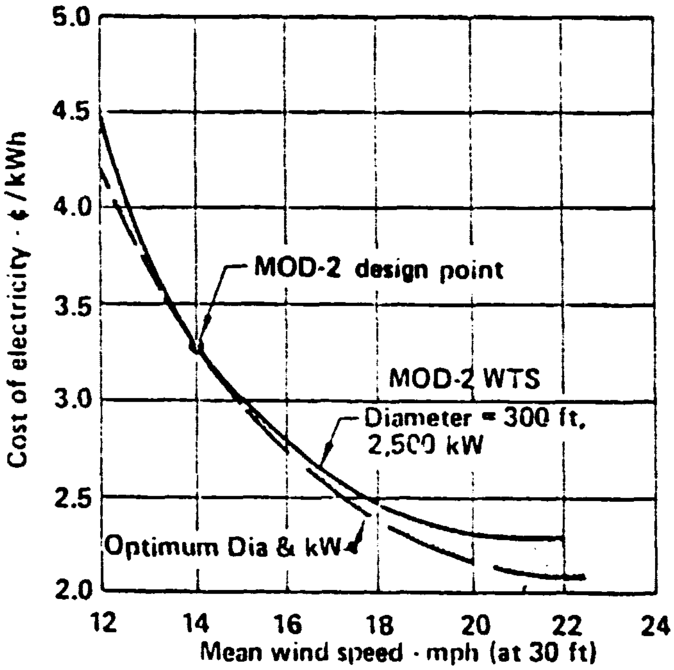
Figure 4-88. Effect of Service Life on Cost of Electricity

4.4.2 System Design Requirements

This section documents the sensitivity studies, as applicable, that are listed in section 2.0 of exhibit B of the contract (System Design).

4.4.2.1 Mean Wind Speed

Specifications required that the design yearly mean wind speed be 14 mph at 30 ft. reference height. Studies were made at other annual mean wind speeds for a rating of (a) 2500 kW and 300 ft. rotor diameter and (b) for optimum (lowest cost of electricity) diameter and power rating. These results are shown as follows (Figure 4-89):



- Mean wind speed has a significant effect on energy cost
- MOD-2 design can be used over a wide range of mean wind speeds
- MOD-2 has good siting versitility when compared to wind turbine designed for higher wind sites

Figure 4-89. Effect of Mean Wind Speed on Economic Performance

4.4.2.2 Extreme Wind Speed

The Specification required that the extreme wind loads be computed for a 120 mph wind at the 30 ft. reference height. When the boundary layer is considered, this design maximum wind becomes 127 mph at the rotor hub. Recurrence interval vs wind velocity was investigated for Central Wyoming and the Southeast coast of the United States and are plotted in Figure 4-90. Specification winds are shown to occur once every 500 years in Wyoming and approximately every 60 years on the Southeast Coast. These data are from pages 26 and 30 of W. Frost and B. H. Long, "Engineering Handbook on the Atmosphere Guidelines for use in Wind Turbine Development", Final Report NASA/MSFC Contract NAS8-32118, University of Tennessee Space Institute, November 1977.

Loads due to extreme winds comprise critical design conditions for only two structural components: the reinforced concrete foundation and the tip of the steel rotor blade. This sensitivity study compares the cost of designing to an extreme wind speed value versus the cost of failure due to the extreme winds. The cost of failure data has been developed from a rigorous probability of failure analysis.

These studies were based on the following assumptions:

1. Applied structural loads (L) may be obtained from a deterministic function of the fastest-mile wind; that is, fastest-mile wind speed is the only random variable involved in the determination of structural loads.

2. Component internal structural loads (S) may be expressed in terms of wind speed.

3. Both applied and internal loads as well as strength variations may be adequately represented by the lognormal probability law.

4. The reliabilities of a few individual structural components may be considered independently.

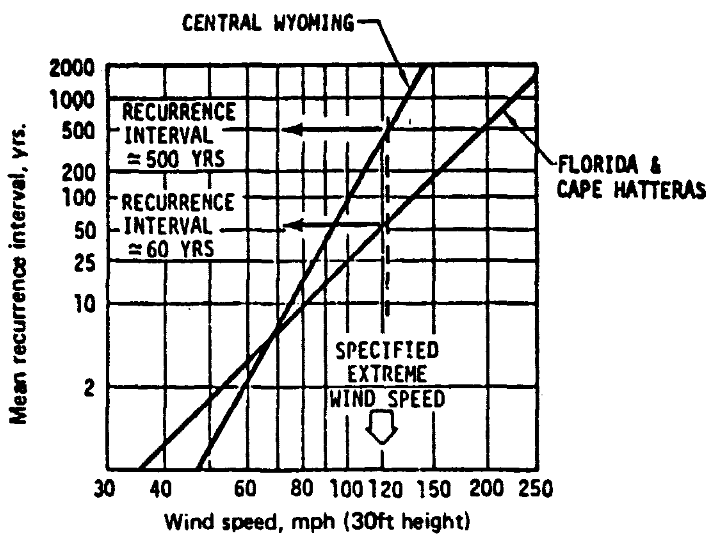
With these assumptions, the probability of failure of an individual structural component over the 30-year life of the MOD-2 wind turbine was estimated using allowable strength probabilities corresponding to MIL-HDBK-5 allowables with an assumed coefficient of variation based on a corresponding factor of safety.

These values were:

	FOUNDATION	ROTOR TIP
Factor of safety	1.50	1.25
Allowables (P_A)	"B" (0.90)	"A" (0.99)
Strength Coefficient of Variation	0.25	0.15

The reciprocal of the calculated failure rate (mean-time-to-failure) is shown in Figure 4-91.

With the above calculated probabilities of failure, the MTBF is applied to determine the percentage of a failure cost per year. The cost of a failure of the foundation is the complete loss of the WTS ($1,942,000). The cost associated with loss of the tip blade panel ($120,000) is the material cost for repair, plus the installation cost for repair, plus the cost of lost electricity during the forced outage time. Since the cost of failure is assumed as a uniform annual cost over the life of the machine, a levelizing factor of 2.0 is applied to account for capital recovery, return on investment, and rate of inflation over the system lifetime of 30 years. The annual cost of failure is then divided by the fixed charge rate (FCR=0.18) to derive the equivalent initial cost.



- RECURRENCE INTERVAL SITE SPECIFIC
- CAPE HATTERAS AND FLORIDA - TIP WORST CASE
- CENTRAL WYOMING REPRESENTATIVE OF LARGE FAVORABLE WIND AREAS IN CENTRAL CONUS

Figure 4-90. Occurrence of Maximum Winds

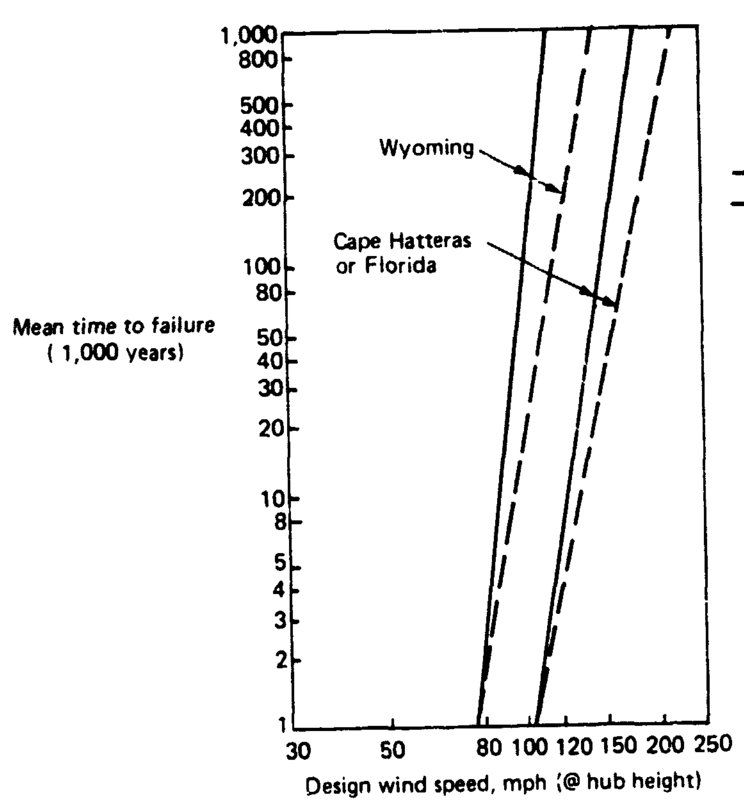


Figure 4-91. Structural Failure Probability Due to Extreme Winds

The results of the sensitivity studies are shown in Figure 4-91.1. The effect of extreme wind criteria on the initial systems cost is shown by the dashed lines. As noted, below 100 mph the tip blade panel is fatigue critical rather than designed by the extreme wind. In Figure 4-91.1, the equivalent initial costs of failures are plotted as a delta cost from the specified design extreme wind speed. Examination of these data show that the new cost optimum for Wyoming would be an extreme wind speed of approximately 100 mph at hub height for both the foundation and tip blade. However, at the higher wind frequency of Cape Hatteras or Florida, the 127 mph wind speed is the least cost design point. Since the savings which could be realized for Wyoming are very small, it is appropriate to reduce risk by designing for the sites with higher extreme winds.

4.4.2.3 90% Minimum Availability

The MOD-2 availability requirement of at least .90 was examined to determine if it was a cost effective level. A preliminary reliability analysis based on the use of commercially available components was conducted to form a baseline for further trade studies. Major components were then examined to determine if additional expenditures would be justified to increase reliability. Sensitivity studies that traded availability vs increased cost were conducted for the rotor, gearbox, generator, and electrical/electronics components. It was determined that expenditure of additional funds was not cost effective for any of these items with the exception of the electrical/electronic components where selective redundancy was applied.

Additionally, various maintenance concepts were cost traded to arrive at the least cost logistics support elements. It was determined that a two man shift, two shifts per day was optimum for a 25 unit WTS farm and that it was cost effective to have a full complement of spares on site.

Based on the above studies it was determined that the .90 availability requirement was cost effective and that a .96 availability goal was appropriate. See section 5.4 for a more thorough discussion of the reliability/maintainability analyses.

4.4.3 Environmental Constraints

This section documents the applicable sensitivity studies that were applied to the specifications in section 3.0 of exhibit B.

4.4.3.1 Wind Speed Duration Profile

Sensitivity studies of the wind speed duration profile to energy output were conducted and are shown in Figure 4-92. Wind duration profiles were also examined for several D. O. E. sites which are now under investigation and it was concluded that a Weibull 'K' constant equal to 2.27 gave a distribution of wind duration that most closely approximated a majority of the sites. A constant of 2.27 has been used widely in the MOD-2 performance analyses and produces near maximum energy output at the design specific power loading of 35.4 watts/sq. ft. These data are summarized in Figure 4-92.

4.4.3.2 Atmospheric Density

Sensitivity studies were conducted to determine the effect of atmospheric density on the energy output of the MOD-2 wind turbine. These studies were made on MOD-2-103 which used a constant power law wind profile. This study is summarized as follows for density changes due both to temperature and altitude (Figure 4-93).

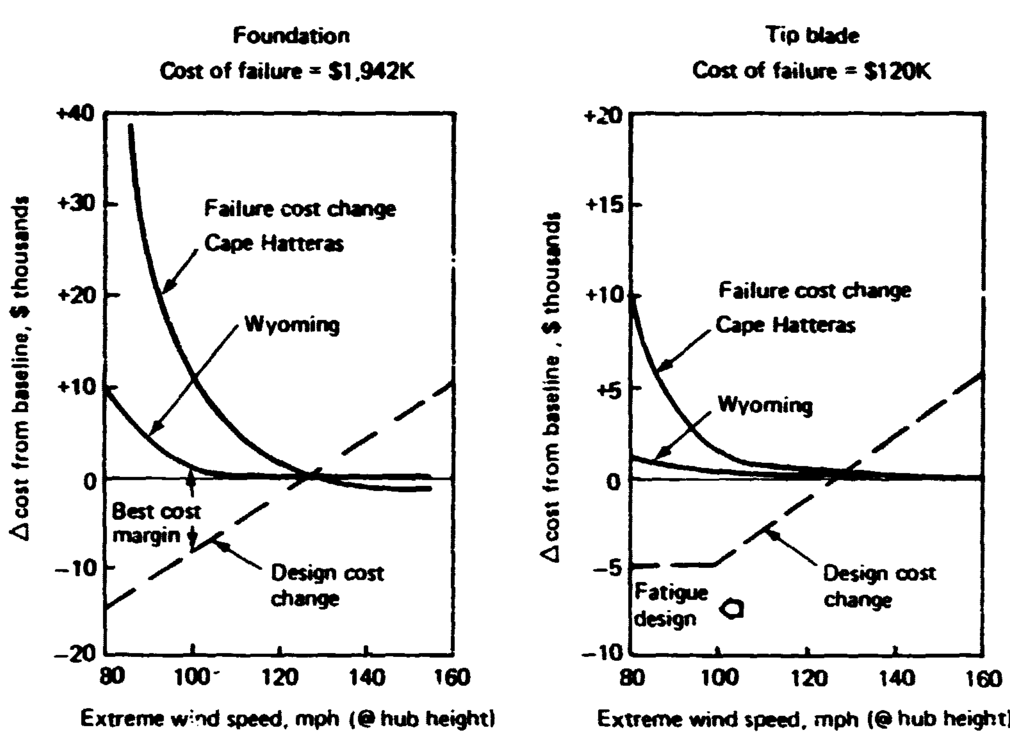
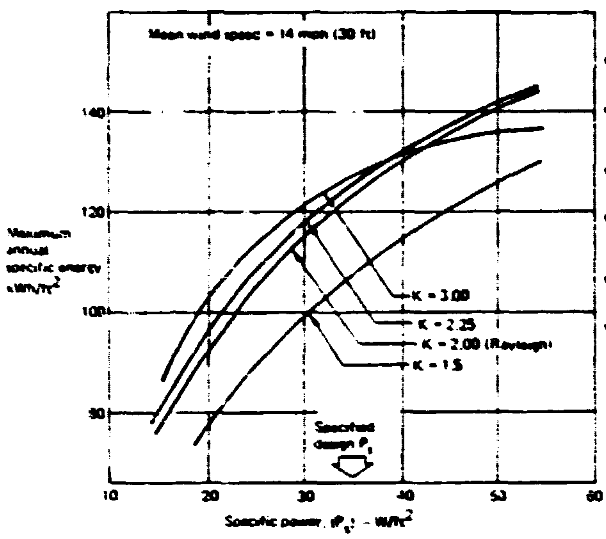
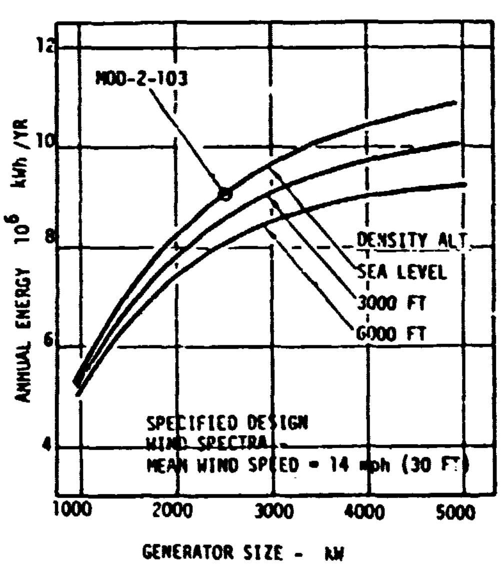


Figure 4-91.1. Effect of Extreme Wind on Failure Cost Margin



- Wind speed duration profile is site specific
- Fatigue load spectra is a function of wind speed distribution
- Energy output is a function of distribution of wind speed
- Weibull constant that best defines a majority of sites under measurement is K = 2.25
- K = 2.27 is specified for MOD 2 103
- Sites with K < 2.27 will reduce fatigue life

Figure 4-92. Effect of Weibull Constant (K) on Energy Output



- OPERATION AT 3000 FT (STD) REDUCES ENERGY OUTPUT BY 5%
- OPERATION AT 6000 FT (STD) REDUCES ENERGY OUTPUT BY 11%
- OPERATION AT 90° F AT SEA LEVEL REDUCES ENERGY OUTPUT BY 3.5%
- COLD DAY OPERATION (20°F) AT SEA LEVEL INCREASES ANNUAL ENERGY OUTPUT BY 4.6%

Figure 4-93. Effect of Air Density on Energy Output

4.4.3.3 Effect of Ground Surface Roughness

The effect of the surface roughness was studied to determine the effect on cost of electricity. Surface roughness causes a change in wind profile and the rougher the surface the greater the average wind changes with increase in height above the ground. Therefore, a rougher surface will produce higher fatigue loads in the components, especially the rotor. If the mean average wind is considered constant at the rotor hub, the cost of electricity increases with increase in surface roughness. This study is summarized in Figure 4-94.

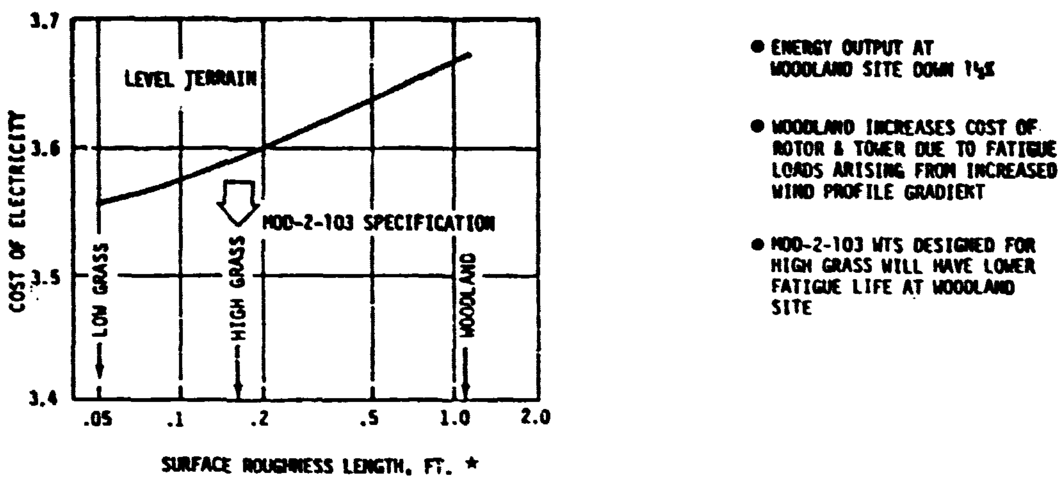


Figure 4-94. Surface Roughness Effect on Cost of Electricity

* Reference: NASA PIR 70, revised wind shear power law model, D. Spera and T. Richard, dated 9/11/78

4.5 FAILURE MODE AND EFFECTS ANALYSIS

4.5.1 Introduction

The failure mode and effects analyses (FMEA) were performed during preliminary design in order to identify all MOD-2 failure modes and to ensure that the resulting effects are (1) acceptable from a personnel safety standpoint and (2) fail safe or, if not fail safe, represent least cost solution (i.e., result in the lowest cost of electricity). The complete FMEA is contained in a document titled "MOD-2 Failure Mode and Effect Analysis", dated November 2, 1978 and submitted to NASA as part of the PDR data package.

The FMEA's were completed by the cognizant designers and reviewed by system engineers and a reliability specialist. In general, fail safe design is employed wherever cost effective and a safe life design is employed for single thread structural items such as the rotor. The failure severity code used in the analysis is described in Table 4-61 and an example of a completed FMEA worksheet is shown in Figure 4-95. Completion of the FMEA's by the designers as an in-line part of the design process resulted in numerous design changes to either prevent serious failure modes or reduce their impact.

Whenever applicable, failure frequency data has been included in the analyses. This data was used to quantify the probability of occurrences and hence the impact on the cost of electricity. The major sources for the failure frequency data is as follows:

1. Non-electronic Reliability Notebook, Rome Air Development Center, January 1975, AD/A-005 657
2. Reliability of Electrical Equipment in Industrial Plants, IEEG Survey
3. Component Removal Data - 727, 737, 747 Commercial Aircraft, Compiled by the Boeing Company
4. Component Removal Data - 107 Helicopters
5. RADC Reliability Notebook

The "k" factors used to account for environmental differences were based on the following groundrules:

1. Rotor - equivalent to helicopter and "unmanned aircraft" environment
2. Nacelle equipment - equivalent to "manned aircraft" environment
3. Ground equipment - equivalent to "ground, stationary" environment

4.5.2 Summary of Results

Over 750 failure modes were analyzed and numerous corrective actions were implemented to preclude costly failures. Special attention was directed at all potentially catastrophic failure modes; the results of this effort are summarized in Table 4-62.

Wherever practical, redundancy is used to preclude catastrophic failures. Safe life design is employed for items whose failure could cause serious damage, but could not be made redundant (e.g., blade fatigue cracks). The ability to control each rotor pitch control surface independent of the other precludes several potentially serious failure modes, such as control linkage binding or bearing failures (the MOD-2 design provides for an orderly shutdown with just one control tip operative). The most probable, potentially catastrophic failure mode is the rapid progression of a fatigue crack in the blade. Boeing has developed a crack detection system which is designed to initiate an orderly WTS shutdown prior to suffering significant rotor damage.

Table 4-61. FMEA Safety and Failure Severity Categories

Hazard category	Impact			
	Function	Repair cost	Time to repair	Personnel injury
Minimal	None [1]	and Under $1,000	and Under 2 days	and None
Marginal	None critical [2]	and Under $1,000	and Under 2 days	and First aid
Critical	Loss of function [3]	or Up to $10,000	or Up to 10 Days	or Hospital
Catastrophic	Loss of system [4]	or Over $10,000	or Over 10 days	or Fatal or permanent disable

[1] Minor items that can be repaired with convenience.

[2] No loss of generating capability, but repair must be accomplished within 2 weeks to avoid shutdown.

[3] Causes WTS shutdown.

[4] Destruction of major element such as rotor or gear box.

MOD-2 Failure Mode and Effects Analysis

SUBSYSTEM	COMPONENT	PAGE / FMEA NO
NACELLE - EPS	POWER CABLE TO THE YAW DRIVE SYS	PAGE 79; FMEA NO 6. 6. 2.

FUNCTION OF COMPONENT

PROVIDE A PATH FOR THE FLOW OF ACCESSORY POWER FOR OPERATION OF THE YAW DRIVE SYSTEM. CABLE RUNS FROM THE CIRCUIT BREAKER PANEL IN THE GENERATOR ACCESSORY UNIT TO THE YAW DRIVE SYSTEM.

FAILURE MODES & EFFECTS	APPLICABLE OPERATING MODES
1 Cable is open circuited. WTS shuts down	F, G, H
2 Cable is shorted to ground, protective circuitry clears, WTS shuts down	F, G, H
3	
4	

FAILURE FREQUENCY				FAILURE SEVERITY			
FAILURE MODE NO	FAILURE MODE FREQ %	COMPONENT FAILURE RATE x 10^6 PER HOUR	MEAN TIME BETWEEN FAILURE (YEARS)	MINIMAL I	MARGINAL II	CRITICAL III	CATASTROPHIC IV
1	75	.15	>1000			X	
2	25	.15	>3000			X	

FAILURE DETECTION METHODS

1 & 2. Power output from WTS indicates zero.

2

3

DISCUSSION AND CORRECTIVE ACTION (IF APPLICABLE)

Yaw drive hydraulic pump motor does not run or develop sufficient power to maintain yaw accumulator pressure. Excessive difference between nacelle position indicator and wind sensors initiates shutdown.

NAME G. TRUSK/ H. ROTH

DATE 8/17/78 Rev. 9/26/78 Rev. 10/25/78

OPERATING MODES

A SHUTDOWN
B TRANSITION TO WARM-UP
C WARM UP
D TRANSITION TO STANDBY
E STANDBY
F TRANSITION TO OPERATE
G OPERATE
H TRANSITION TO FEATHER
I FEATHER

78 K-6162-001

Figure 4-95. Example of FMEA Worksheet

Table 4-62. Summary of Major Modes and Effects

Failure mode	Effect	Potential corrective action
Structural failures		
Rotor		
• Blade fatigue cracks • Spar buckling • Fatigue crack at control tip spindle and thread or at tip-blade interface • Inboard joint rotor to hub bolt or flange weld failure • Broken teeter trunnion or flange cracks • Buckling inboard sections or hub compression chine	Loss of part of rotor and possible secondary damage if allowed to progress	• Safe life design • Fatigue tests • Inspection schedule • Possible crack detection system • Mid-blade assembly buckling test
Drive		
• Broken low speed shaft, • Broken quill shaft bulkhead joint	Loss of load emergency shutdown effected prior to reaching damaging overspeed (see memo K-6286-SS-487, DTD, 18 Oct. 1978)	• Safe life design
• Teeter shaft or flange cracks	Possible rotor loss	• Strain gage correlation • Safe life design
Tower		
• Failure of structure or foundation	Could initiate collapse	• Safe life design
Control system failures		
• Signal to one tip incorrectly drives control surface to zero pitch • Control linkage to one tip jams	Emergency shutdown triggered by differential of tip position signals. Shutdown occurs prior to damaging overspeed	None required, analysis verifies that one tip operative can safely stop rotor (see memo K-6286-SS-487, DTD 18 Oct. 1978)
• Control system signal to both pitch actuators incorrectly drives control surfaces to zero pitch	Emergency shutdown triggered by generator output power sensor. Shutdown occurs prior to damaging overspeed	None required
• Power output sensor fails, calling for power increase when system is already at full power output	Damaging overspeed possible if load drops off prior to initiating shutdown	System changed to command shutdown prior to load dropping off. Also, back-up power sensor signal sent to controller
Electrical power failures		
• Synchronizer provides signal to close line tie contactor too soon or too late (WTS not proper phase relationship or voltage to mate with bus)	High current transient causing high torque load on the generator that could cause mechanical damage to generator or drive train	Synchronizer is fully redundant and fail safe
• Loss of commercial power while WTS is at rated power	Shutdown occurs prior to damaging overspeed	None required, see memo K-6286-SS-487, DTD 18 Oct. 1978

4.6 UTILITY INTERFACE ANALYSIS

Technical information exchange meetings were held on three occasions with representatives of utility companies to determine the suitability of the utility interface. Participation in each of the meetings is noted in Table 4-63.

Table 4-63. Utility Interface Meeting Participation

Meeting Date	Participants
October 27, 1977	Bonneville Power Administration Pacific Gas and Electric Co. Portland General Electric NASA Lewis Boeing
February 8, 1978	Bonneville Power Administration Pacific Gas and Electric Co. Portland General Electric NASA Lewis Boeing
July 25, 1978	Bonneville Power Administration Bureau of Reclamation Pacific Gas and Electric Co. Portland General Electric Southern California Edison NASA Lewis Boeing

As part of the agenda for each meeting the design and operation of the MOD-2 WTS, as it was defined at the time, was described. The informal presentations were interrupted by questions that reflected the special interests and concerns of the utilities. The discussions, centered around these interruptions, focused attention on specific design areas. Later assessment by Boeing resulted in informal trade studies, analyses, practical alternatives and further refinement of the conceptual and preliminary designs. The meetings significantly impacted the configuration of both the control and electrical power systems in the areas noted in Table 4-64.

Table 4-64. Utility Suggested Design Criteria

1. Interface with Utility Transmission Line
2. Simplify Utility Manual Controls
3. Operate Generator at High Power Factor
4. Eliminate Diesel Driven Generator
5. Accept Power Swings of 0 to 100%

An early concern of the utilities was associated with power surges due to wind gusts and their effect on the quality of power delivered to customers near the wind turbine. Of specific interest was the method to be used to meet voltage flicker standards. Based on a Boeing analysis, it was concluded that supplying local customers was incompatible with the design concept. The MOD-2 WTS was being designed to generate large blocks of power and deliver them to a utility to supplement existing generating capacity. Because of the non-continuous availability of wind power and the need for complex load management, direct delivery to customers appeared uneconomical. It was concluded that MOD-2 utility interface should be constrained to require the delivery of power to a utility transmission system where it could be combined with power from continuously available sources, thus providing load reduction on these other sources when wind power was available. Control of power quality would be maintained by the continuous source if it provided an order of magnitude more power than the Wind Turbine System. Using this concept, customers would be supplied from a distribution system that was in turn supplied from the transmission system, and the quality of power delivered to a customer would be unaffected by the action of the Wind Turbine System. This concept is used by the utilities for their smaller power sources and it permits the output from the smaller sources to swing from 0 to 100% of their rating without creating significant power line disturbances.

Through our interfacing with the power utilities it was established that the remote, manned utility station, has only limited control of the Wind Turbine System (WTS). This control should include the following capabilities: to enable and disable power production from the WTS; to display the following operational parameters at the WTS: wind data, blade pitch position, rotor rpm, power output, and status of the WTS. Also to be included was a form of alerting the remote attendant of an occurence of a problem at the WTS; and to display operational historical data upon request to aid a "probable cause" troubleshooting of the identified problem.

The generator proposed for use in the Wind Turbine is a standard machine in regular use throughout the country. It is rated to deliver power efficiently at load power factors as low as 0.80. In the utility meeting it was pointed out that the utilities continuously attempt to optimize the efficiency of their systems by transmitting power at near unity power factor. In recognition of this practice it was concluded that Boeing performance analysis of the Wind Turbine System should use a load power factor greater than that of the generator rating. A value of 0.95 was selected for use in the analysis to recognize that optimum operation of this system could not be continuously maintained.

As a result of a maintenance and inspection concern expressed during one of the meetings, the use of a diesel driven generator for standby power was reassessed. Use of this device would require regular start-up by maintenance personnel to assure its availability and require periodic inspection of its fuel storage facility by a safety inspector. The reassessment identified the standby power loads and their function. It was concluded that the unit could be deleted from the design. Certain critical control system loads will be supplied by the station battery. It contains sufficient capacity to maintain the system for an adequate period. Other standby loads were the heaters required for cold weather operation. Their loss is acceptable. Further discussion of the deletion of the diesel driven generator is provided in this report in Section 4.2.4.2.

5.0 SUPPORTING ANALYSES AND VERIFICATION

Development of the MOD-2 design relied heavily upon a number of analytical relationships. Many of these analysis tools had been substantiated on previous programs, but others had to be verified by tests. This section describes the methodology used to ensure that the analyses used to support the selection of each design feature was accurate and gave the best results possible.

The general approach used to achieve accurate analysis tools was as follows. If a recognized analytical technique, formula or computer program was already in general industry use, it was applied to the MOD-2 trade studies. If such tools did not exist they were developed and verified by comparing their predicted results with test data. In some cases it was necessary to conduct tests (e.g. structural fatigue, buckling, wind tunnel tests); the analytical tools were then changed as appropriate to reflect the empirical results from the tests.

The following sections describe the analyses programs, their development, and sample results.

5.1 STRUCTURAL ANALYSES AND VERIFICATION

5.1.1 Code Verification

REQUIREMENT: It was required that the contractor verify (wherever practical) all computer codes used in the analysis of rotor blades and coupled dynamic loads.

APPROACH: The approach to code verification taken by Boeing for MOD-2 was as follows:

(1) Identify existing codes used by industry for each design condition
(2) Obtain a verified computer code available in industry as the prime analytical code
(3) Assure that the version of the code obtained is traceable to previous verification work
(4) Identify additional required future code modification
(5) Perform tests required for further verification

COMPUTER CODES: The primary computer code obtained by Boeing for rotor loads analysis is MOSTAB. This code has been verified for full-span fixed rotor analysis by correlation with MOD-2 test data. In addition to MOSTAB, other computer codes are required for structural analysis of various aspects of the WTS system. These computer codes have all been verified by many years of usage in previous hardware programs and correlation with test data. A list of computer codes used in the MOD-2 program is shown in Table 5-1.

MOSTAB/MOSTAS VERIFICATION: Test programs were conducted to verify the MOSTAB code for a teetered, partial span rotor, and to verify the MOSTAS (system) code for loads analysis of rotors on a soft tower wind turbine system. Testing was done on a 1/20 Mach scale wind tunnel model of the MOD-2 WTS. Test results were as follows:

Performance: Good correlation of power vs collective pitch was achieved with MOSTAB as illustrated in Figure 5-1.

Table 5-1. MOD-2 Computer Codes

NAME OF PROGRAM	TYPE	APPLICATION
MOSTAB	Rotor loads analysis	MOD-2 rotor loads
NASTRAN	Finite element dynamic & stress analysis	Rotor & tower natural freq. & stress analysis
BOSOR-4	Analysis for shells of revolution	Tower stress analysis
STAGS	Shell analysis	Blade buckling
GEM-1 *	Aerodynamics	Aero performance
SECTIONS	Section properties	MOD-2 blade stations
LSD *	Structural dynamics simulation	Teeter stop analysis
C-60	Rotor dynamics prog.	Rotor loads
L-01	Blade loads & flutter	Flutter analysis
EASY *	Control simulation	Pitch control analysis

* Boeing Proprietary

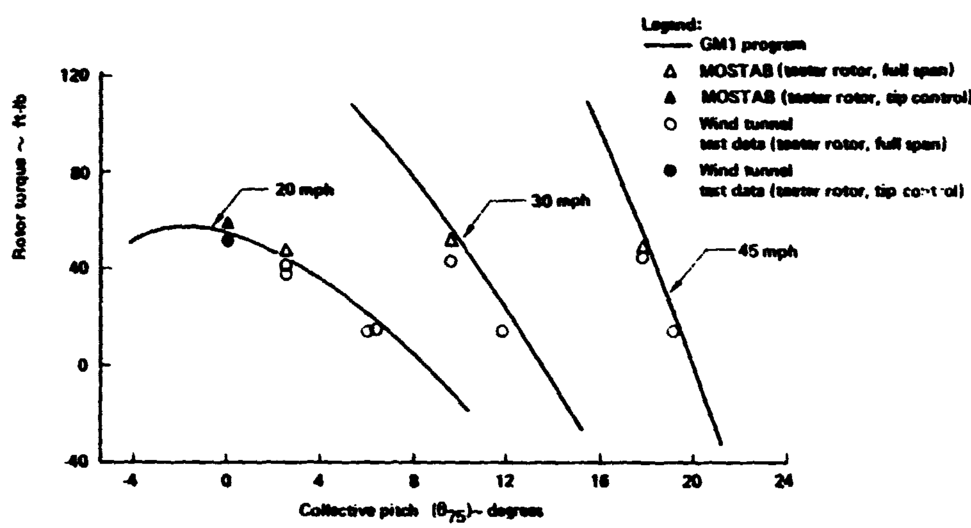


Figure 5-1. Performance Correlation

Blade Loads: Correlation of MOSTAB predictions with measured blade loads for the teetering, tip control configuration is shown in Figure 5-2 and 5-3. Good correlation was achieved for steady flapwise and chordwise moments. Correlation of alternating flapwise moments could only be achieved by using a nonlinear wind gradient in MOSTAB. A factor of 1.65 was applied to the measured wind gradient velocity decrement to improve correlation of alternating flapwise moments with test data as shown in Figure 5-2.

MOSTAB provided a good prediction of the gravity (one per rev) content of the chordwise moments.

An attempt to improve the blade loads correlation by analyzing the coupled system with MOSTAS-A was disappointing. For the full span-fixed rotor the blade moment perturbations were small and did not improve the correlation. For the teetering rotor, a program error prevented recovery of blade moments for the coupled analysis so that no correlation could be made. Work is continuing on the MOSTAS correlation.

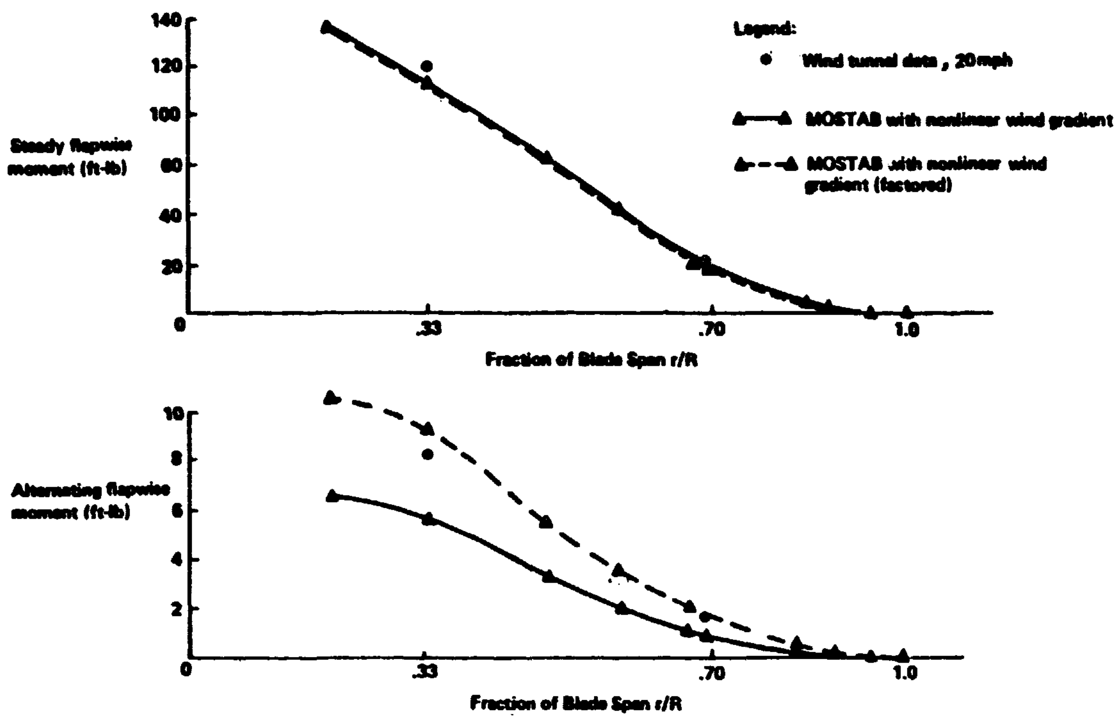


Figure 5-2. Correlation of MOSTAB with Measured Flapwise Moments

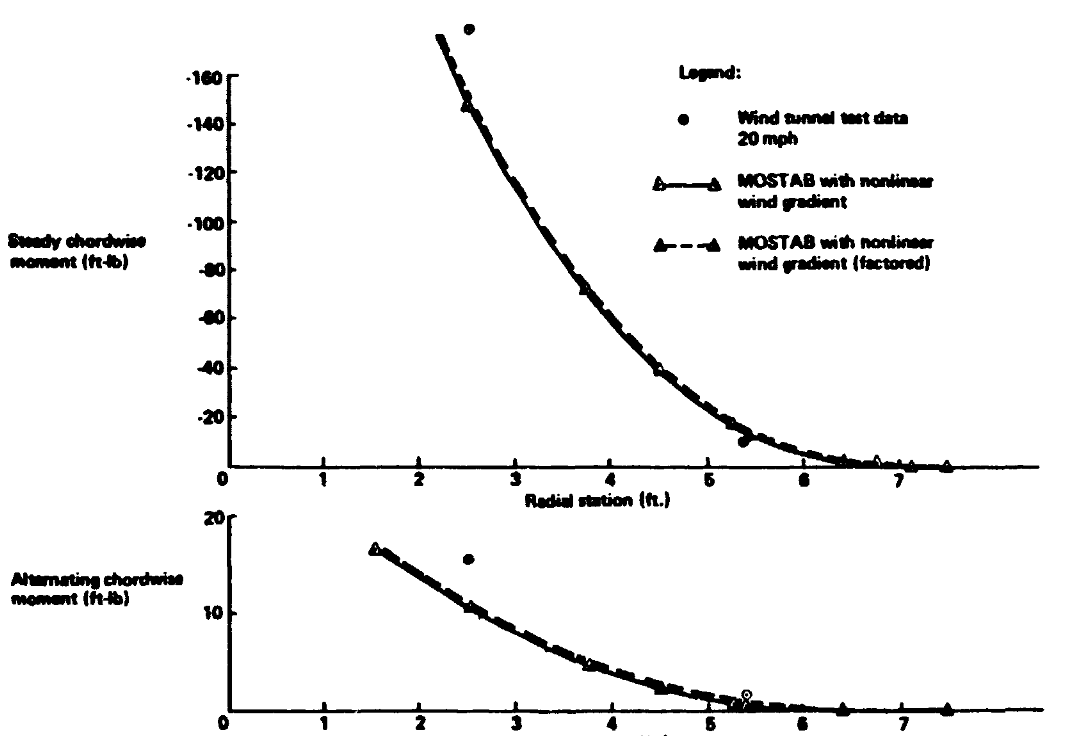


Figure 5-3. Correlation of MOSTAB with Measured Chordwise Moments

Tower Motions: MOSTAS-A predictions were correlated with motions measured at the top of the tower. Correlation of the even harmonic content of tower motion was surprisingly good, lending some credence to the coupled analysis. The measured motions were so low in comparison to instrumentation noise levels, however, that generalization about the accuracy of MOSTAS-A cannot be made. MOSTAB-B was not evaluated.

Teeter Angle: Correlation of teeter angles predicted by MOSTAB with measured data was good as shown in Figure 5-4.

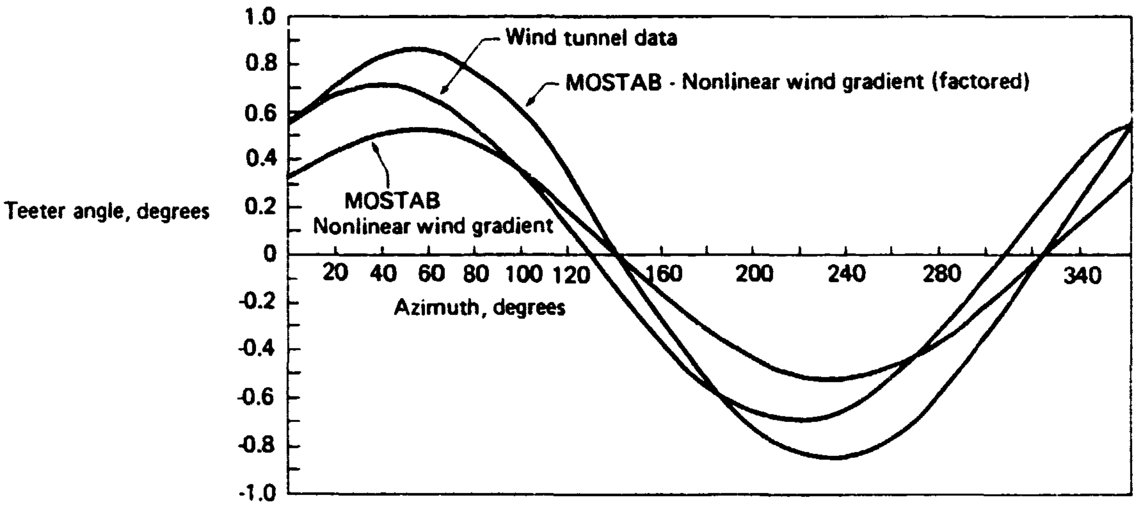


Figure 5-4. Correlation of MOSTAB with Measured Teeter Angle

5.1.2 Development of Fatigue Life

Determination of fatigue life and the development of fatigue allowables for a 30 year life is being accomplished using the "pre-existing crack" damage tolerance approach. This approach assumes the statistically determined worst possible defect, which could escape detection during fabrication and inspection, to exist when the system is put into service. The growth of the initial crack is described by a crack growth model which employs the stress intensity concept for characterizing the crack growth. Verification of the crack growth model was accomplished by testing preflawed specimens under representative spectrum load conditions and correlating these results with the crack growth model. The pre-existing crack concept was selected for analysis and test rather than a conventional fatigue analysis because it is more representative of real life and verification of a conventional fatigue analysis by test would be prohibitive in terms of cost and schedule because of the large number of tests required.

Originally three different crack growth models were considered:

- Retardation Model-Accounts for load interaction effects and considered all cycles to produce crack growth

$$da/dn = C(1-R)^n (Kmax)^m (K/Kol)^L$$

- Threshold Model-Considers only those cycles above the threshold to produce damage

$$da/dn = 0 \text{ for } K \le K_{th}$$
$$da/dn = C(1-R)^n (Kmax)^m \text{ for } K > K_{th}$$

- Combined Model-Combines the above to account for both threshold and load interaction effects

$$da/dn = 0 \text{ for } K \le K_{th}$$
$$da/dn = C(1-R)^n (Kmax)^m (K/K_{ol})^L \text{ for } K > K_{th}$$

Verification of crack growth model validity was accomplished by analyzing the test data using the above models and comparing actual with predicted results. The combined model provided excellent correlation between actual and predicted results whereas the retardation underestimated the lives of the long term tests and the threshold model underestimated the lives of the short term tests. The final combined model used for determining fatigue life for all of the steel assemblies (all of the steels are covered by either the ASTM A-6 or A-20 specifications) is as follows:

$$da/dn = 0 \text{ for } K \le K_{th}$$
$$da/dn = 3\times10^{-10}(1-R)^{2.4}(Kmax)^3(Kmax/Kol)^2 \text{ for } K > K_{th}$$

Where:	da/dn	=	crack growth rate in inch/cycle
	R	=	$\frac{\text{minimum stress}}{\text{maximum stress}}$
	K	=	stress intensity
	Kmax	=	maximum stress intensity for each block of cycles
	Kol	=	maximum stress intensity in each spectrum
	K_{th}	=	maximum stress intensity for which $da/dn=0$ (see Figure 5-5)

The above combined model was used to analyze the results from spectrum load tests. A comparison of the actual test stress level and the predicted stress level to produce the test life is given in Figure 5-5.1. A detailed description of each load spectrum is presented in Table 5.1-1. Further substantiation of the crack growth model was obtained through tests of a simulated field joint. The specimen configuration and test results are presented in Figure 5-6.

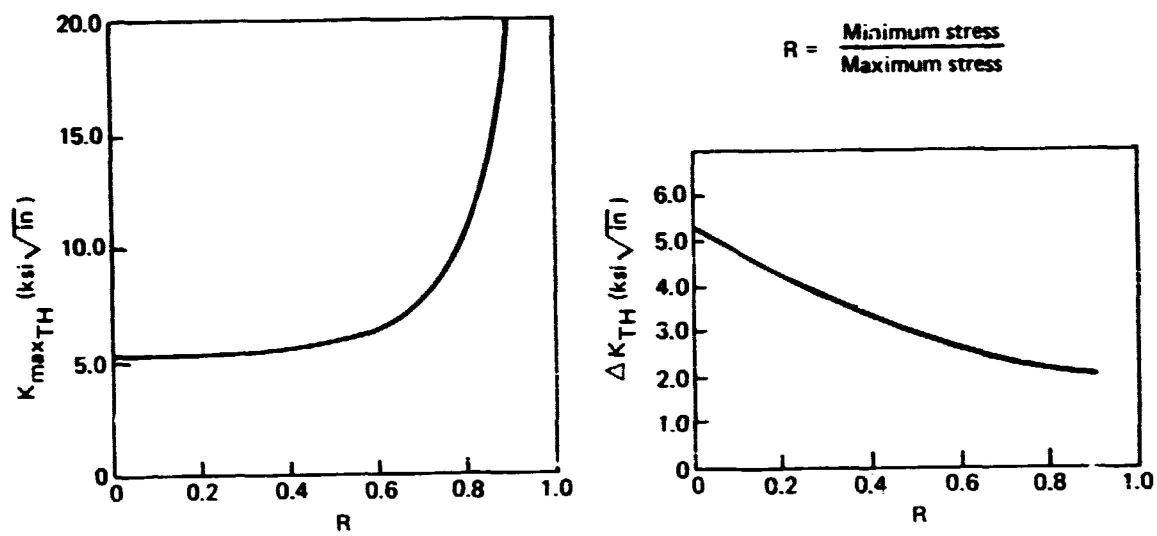


Figure 5-5. Relationship Between Threshold Stress Intensity and R Ratio

Test results and data correlation have been accomplished exclusively with base metal specimens or weld metal specimens which were free of detrimental residual stresses. To verify the applicability of the data and crack growth model for stress relieved weldments a specimen was produced with an area of residual stresses and then stress relieved. The ratio of actual to predicted stress level to produce the life for the specimen was 1.06. The good correlation of actual with predicted behavior for this specimen provide additional verification of the procedure for determining fatigue allowable.

The fatigue allowables were calculated using the combined model with the initial flaw size defined in Table 5-2 The variation in assumed initial flaw size reflects the initial quality and the ability to inspect the material. The lower the initial quality or inspectability, the larger is the assumed initial flaw to increase the chance of finding, and decrease the possibility of having, a flaw of the assumed size. The flaw sizes used in the determination of allowables are much greater than the detectable size.

- Each bar represents a test data point
- Except as noted the test material was A533

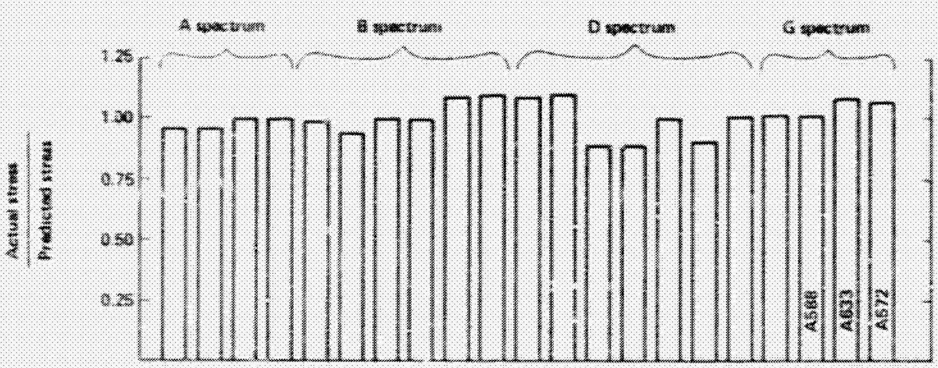
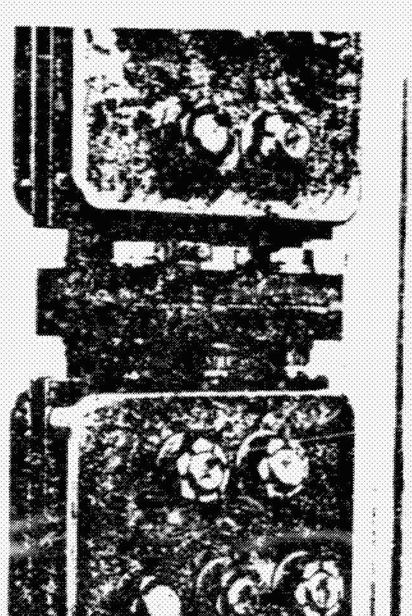
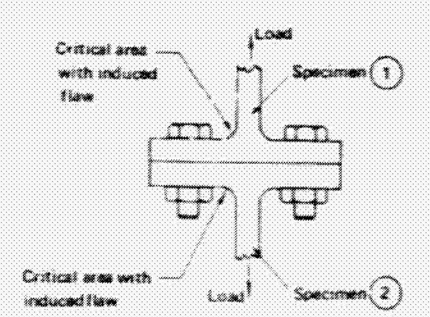


Figure 5-5.1. Correlation of Test and Predicted Results





TEST DATA

	1	2
• Fillet area stress (psi)	18,000	18,200
• Initial flaw length (in.)	0.248	0.234
• Predicted fatigue life	17,000,000 Cycles	17,000,000 Cycles
• Test Specimen Life	18,953,754 Cycles	21,151,277 Cycles

Figure 5-6. Field Splice Fatigue Test

Table 5.1-1. Load Spectra

LOAD SPECTRUM A

LOAD BLOCK	CYCLES	LOAD (% OF LIMIT) MAX.	MIN.
1	9076	43.4	6.2
2	3328	49.7	-0.2
3	6050	38.4	11.2
4	10500	46.4	3.1
5	908	52.6	-3.0
6	303	55.0	-5.5
7	6133	88.1	44.7
8	558	92.4	40.5
9	19513	85.5	47.4
10	11150	78.8	54.1
11	16725	82.9	49.9
12	1673	90.4	42.4
13	2373	64.5	19.3
14	4153	79.9	3.9
15	3560	74.1	9.8
16	356	91.9	-8.1
17	119	96.9	-13.1
18	1305	86.4	-2.5
19	34	83.9	-47.2
20	370	71.1	-34.4
21	673	44.8	-8.8
22	1178	63.3	-26.5
23	1010	56.2	-19.5
24	101	77.9	-41.2
25	13050	89.0	52.1
26	1305	97.7	43.4
27	8700	84.2	56.9
28	15225	91.9	49.2
29	435	100.0	41.1
30	4785	95.0	46.1
31	9265	52.8	20.7
32	1390	70.3	3.2
33	16217	62.7	10.9
34	13900	58.9	14.6
35	463	73.3	0.2
36	5097	66.7	6.8

NOTE: Load Spectrum A consists of Load Blocks 1 thru 36 (Total cycles per Spectrum 191,069)
1000 Spectra equals 1 lifetime

LOAD SPECTRUM B

LOAD BLOCK	CYCLES	LOAD (% OF LIMIT) MAX.	MIN.
1	3328	49.7	-0.2
2	908	52.6	-3.0
3	303	55.0	-5.5
4	6133	88.1	44.7
5	558	92.4	40.5
6	1673	90.4	42.4
7	4153	79.9	3.9
8	3560	74.1	9.8
9	356	91.9	-8.1
10	119	96.9	-13.1
11	1305	86.4	-2.5
12	34	83.9	-47.2
13	370	71.1	-34.4
14	673	44.8	-8.0
15	1178	63.3	-26.5
16	1010	56.2	-19.5
17	101	77.9	-41.2
18	13050	89.0	52.1
19	1305	97.7	43.4
20	15225	91.9	49.2
21	435	100.0	41.1
22	4785	95.0	46.0
23	1390	70.3	3.2
24	463	73.3	0.2
25	5097	66.7	5.8

NOTE: Load Spectrum B consists of Load Blocks 1 thru 25 (Total Cycles per Spectrum 67,512)
1000 Spectra equals 1 lifetime

LOAD SPECTRUM D

LOAD BLOCK	CYCLES	LOAD (% OF LIMIT) MAX.	MIN.
1	3328	67.2	34.5
2	708	69.0	32.7
3	303	70.6	31.1
4	6133	92.2	63.9
5	558	95.0	61.1
6	1673	93.7	62.4
7	4153	86.9	37.2
8	3560	83.0	41.0
9	356	94.7	29.4
10	119	78.0	26.1
11	1305	91.1	33.0
12	34	89.5	3.8
13	370	81.1	12.2
14	673	63.9	29.4
15	1178	76.0	17.3
16	1010	71.4	21.9
17	101	85.6	7.7
18	13050	92.[illegible]	68.7
19	1305	96.5	63.0
20	15225	91.7	66.8
21	435	100.0	61.5
22	4785	96.7	64.8
23	1390	80.6	36.8
24	463	82.6	34.8
25	5097	78.3	39.1

NOTE: Load Spectrum D consists of Load Blocks 1 thru 25 (Total Cycles Per Spectrum 67,512)
1000 Spectra equals one lifetime

LOAD SPECTRUM G

LOAD BLOCK	CYCLES	LOAD (% OF LIMIT) MAX.	MIN.
1	Start at 0.0 and Load to 70.1		
2	64	59.1	11.0
3	13	70.1	11.0
4	2	70.5	35.0
5	64	83.1	35.0
6	60	90.6	35.0
7	10	97.2	45.2
8	17	84.6	28.0
9	64	59.1	11.0
10	14	90.6	36.2
11	10	73.6	25.6
12	64	83.1	35.0
13	21	99.6	51.6
14	13	93.5	22.0
15	55	84.3	36.2
16	2	97.2	36.6
17	2	100.0	22.0
18	4	84.3	22.0
19	55	84.3	36.2
20	60	78.3	30.3
21	200	90.0	41.5
22	30	85.0	31.5
23	60	89.8	36.2
24	200	90.0	41.5
25	10	85.0	41.7
26	30	91.2	37.2
27	1	93.7	0

NOTE: Load Blocks 1 thru 20 are repeated 9 times followed by Load Blocks 1 thru 27 to produce Load Spectra G (Total Cycles Per Spectra 6,481)
10,000 Spectra equals 1 lifetime

Table 5-2. MOD-2 Inspection Matrix

(Defects oriented parallel to thickness direction)

Weld category	Crack growth design allowable flaw size [2]		Inspection method & flaw detection capability		Flaw size acceptance criteria
	Surface	Internal			
B	.05 deep x .25 long	.10 deep x .25 long	VT	.005 wide x .06 long	.06 long (linear) .125 long (rounded)
			PT	.005 wide x .03 long	
			RT	2% of t deep x .04 long	
			UT	.03 deep x .09 long	
C	.10 deep x .50 long	.20 deep x .50 long	VT	.005 wide x .06 long	.06 long (linear indications)
			PT	.005 wide x .03 long	
E	.40 deep x 2.00 long	.80 deep x 2.00 long	VT	.005 wide x .06 long	.125 long (rounded indications)
			PT	.005 wide x .03 long	
Base metal	.022 deep x .11 long	.044 deep x .11 long [1]	VT	.005 wide x .06 long	.06 long (linear indications) .125 long (rounded indications)
			PT	.005 wide x .03 long	
	.030 deep x .15 long	.060 deep x .15 long [1]	RT	Not practical	.06 long x .04 deep
			UT	.03 deep x .09 long	

[1] Acceptance sampling only [2] Flaw size for constant stress intensity.

Note: All base material to satisfy ASTM A-6 requirement.

The crack growth model is used in conjunction with the assumed initial flaw size and load spectrum to determine the allowable stress for thirty years of operation. The allowable stress, for each load spectrum, is the maximum stress for which the predicted time to failure is 30 years. Failure is considered to occur when the crack growth model predicts that the assumed initial defect has grown to a size such that the stress intensity at maximum stress is 125 ksi $\sqrt{in.}$ The 125 ksi $\sqrt{in.}$ value is an estimate of the minimum fracture toughness of the material at -40°F (minimum design temp.) The conservatism in the fatigue analysis is in the assumed initial flaw size and the fact that the flaws are assumed to exist at the worst possible location in the worst possible orientation. The allowable design stresses for Class B welds are presented in Figure 5-7. The determination of actual fatigue life will be based on inspection of actual material quality (flaw size) at critical locations.

5.1.3 Stress Analyses

The MOD-2 Wind Turbine System has been designed to operate trouble free for a period equal to or exceeding thirty years. During this period of time, the dynamically loaded structures of the WTS will experience at least 2.0×10^8 cycles of loading - the primary source of oscillatory loading in both the rotating and fixed systems (examples - blades and drive shaft are in

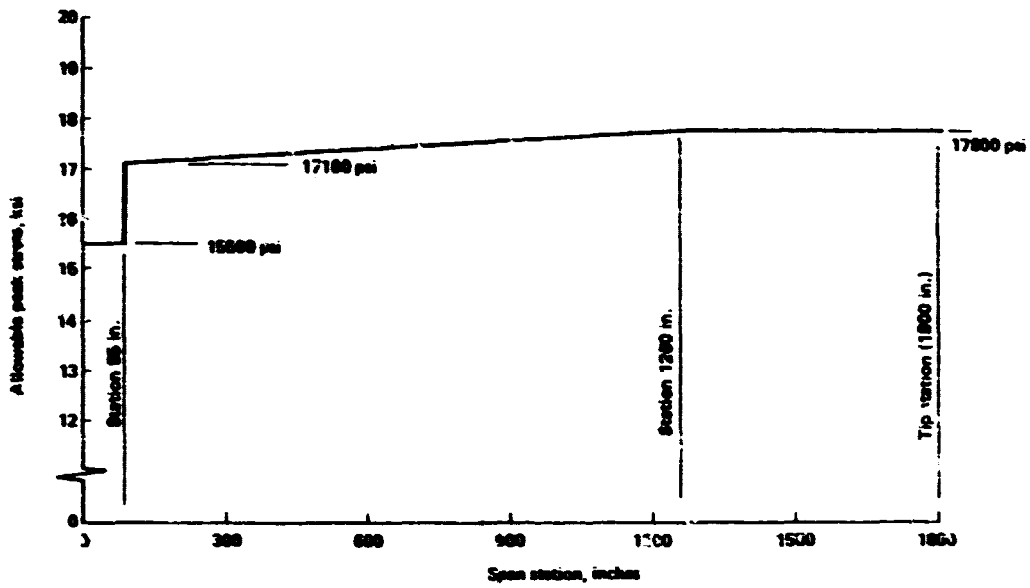


Figure 5-7. Rotor Blade Fatigue Al!owab!e (Category "B" Weids)

the rotating systems while the nacelle and tower are in the fixed systems). The majority of these load cycles are of very low magnitude, producing stresses well below the crack propagation threshold. However, the effects of low wind speed and high wind speed startups and shutdowns are to produce peak stresses of sufficient magnitude and frequency to limit the fatigue life of the affected structures to thirty years. The development of the design fatigue allowable stresses for the anticipated stress spectrum is presented in section 5.1.2.

There are infrequently applied loads which produce stresses that are permitted to exceed those of the fatigue allowables. These stresses are not permitted to exceed yield or buckling stresses, nor are they permitted to produce permanent deformation (see section 4.1.4).

The following subsections address the stress analyses performed on the various structural elements of the MOD-2 WTS and summarize their margins of safety.

5.1.3.1 Design Allowable Stresses

The following paragraphs outline the basis for determination of design allowable stresses.

Basic Material (Element) Allowable -

Materials allowables used for commercial standard components shall be based on specifications for the American Institute of Steel Construction (AISC), American Society of Mechanical Engineers (ASME) and American Society for Testing Materials (ASTM). Material allowables for special designed hardware, e.g. rotor blade, shall be Boeing Design and Material Allowables presented in this document.

Component Allowables -

Structural component allowables shall be based on applicable component test data cr analyses. Where required component allowables data are insufficient or unavailable in the industry, the Contractor shall conduct component testing or analyses as necessary to define WTS structural sizing at the component level. An example of this is the fatigue allowables developed as shown in section 5.1.2.

5.1.3.2 Blade Stress Analysis

The stress analyses of the MOD-2 all metal rotor blade is accomplished to a great measure, as a sub-routine in the MOSTAB computer program. This is the case for all operating loading conditions. A discussion of methodology is presented in this section along with a summary of fatigue and limit loading margins of safety presented in Figure 5-8. The blade mass distribution and section properties are shown in Figures 5-8.1 through 5-8.3.

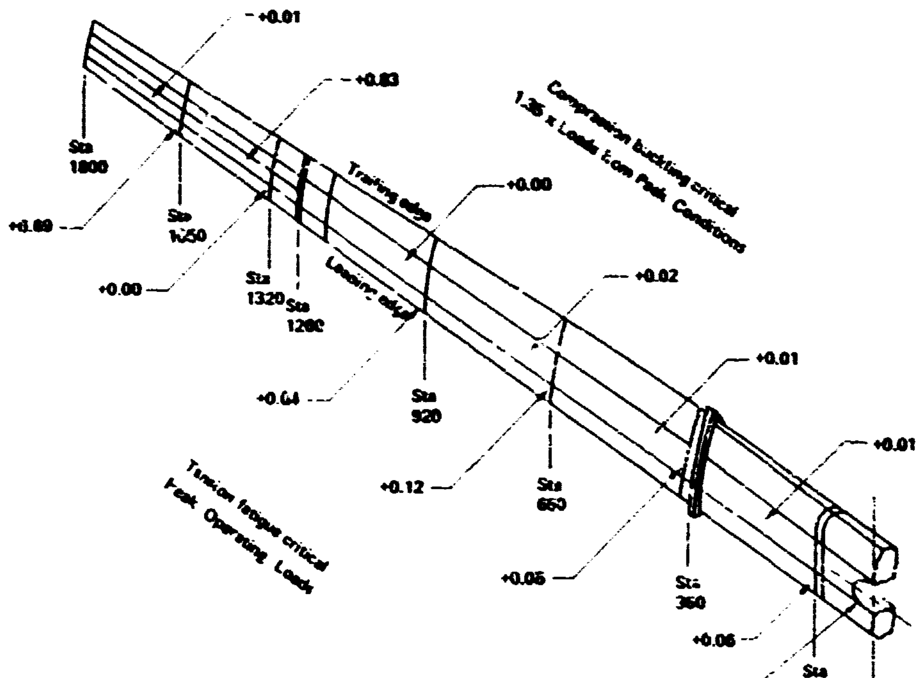


Figure 5-8. Rotor Blade Skin Gages-Margins of Safety

All operating and non-operating loading conditions have been defined by thorough examination of such conditions. Once having defined the operating condition mean wind and gust velocity, the blade pitch control setting and rotor torque are determined in the Boeing EASY computer program as rotor trim boundary conditions. The rotor blade loads are determined at 13 span-wise stations, with internal stresses calculated at 9 locations around the periphery of each span station. Once the stresses are defined for the mean wind and gust conditions, they are extrapolated lognormally to 99.9% and 99.99% probabilities which are the design fatigue and design limit operating conditions respectively. The 99.9% probability stresses for each wind speed are input into the fatigue crack growth model, along with the associated number of stress cycles for each defined condition, for the determination of the design fatigue allowable stresses (section 5.1.2).

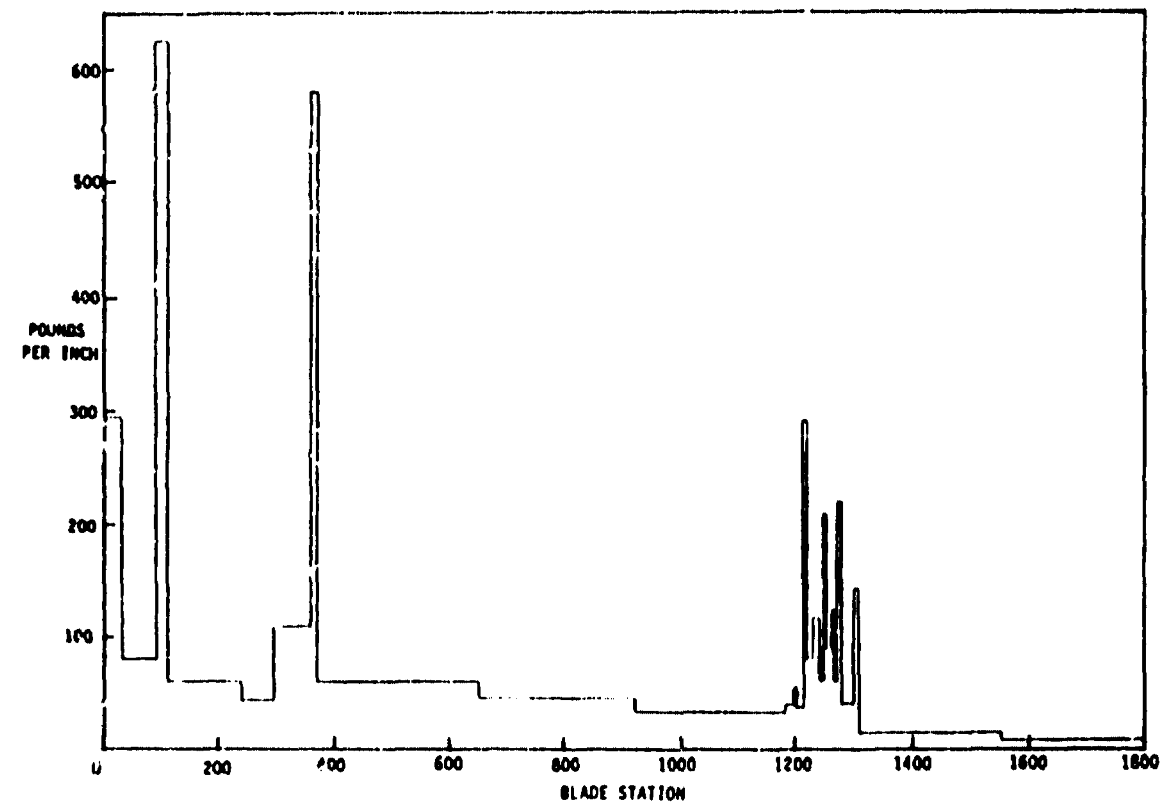


Figure 5-8.1. Blade Mass Distribution

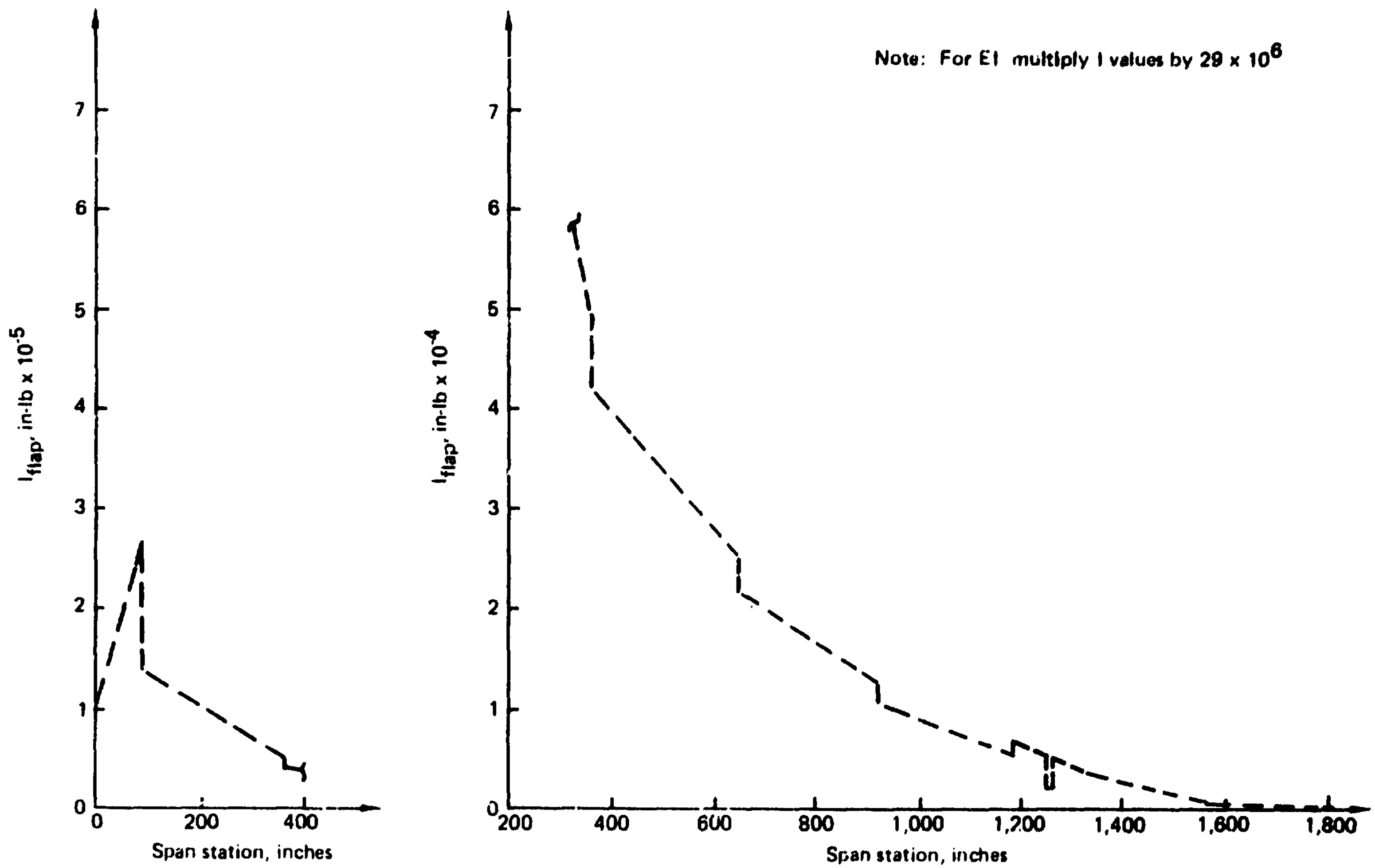


Figure 5-8.2. Blade Section Properties

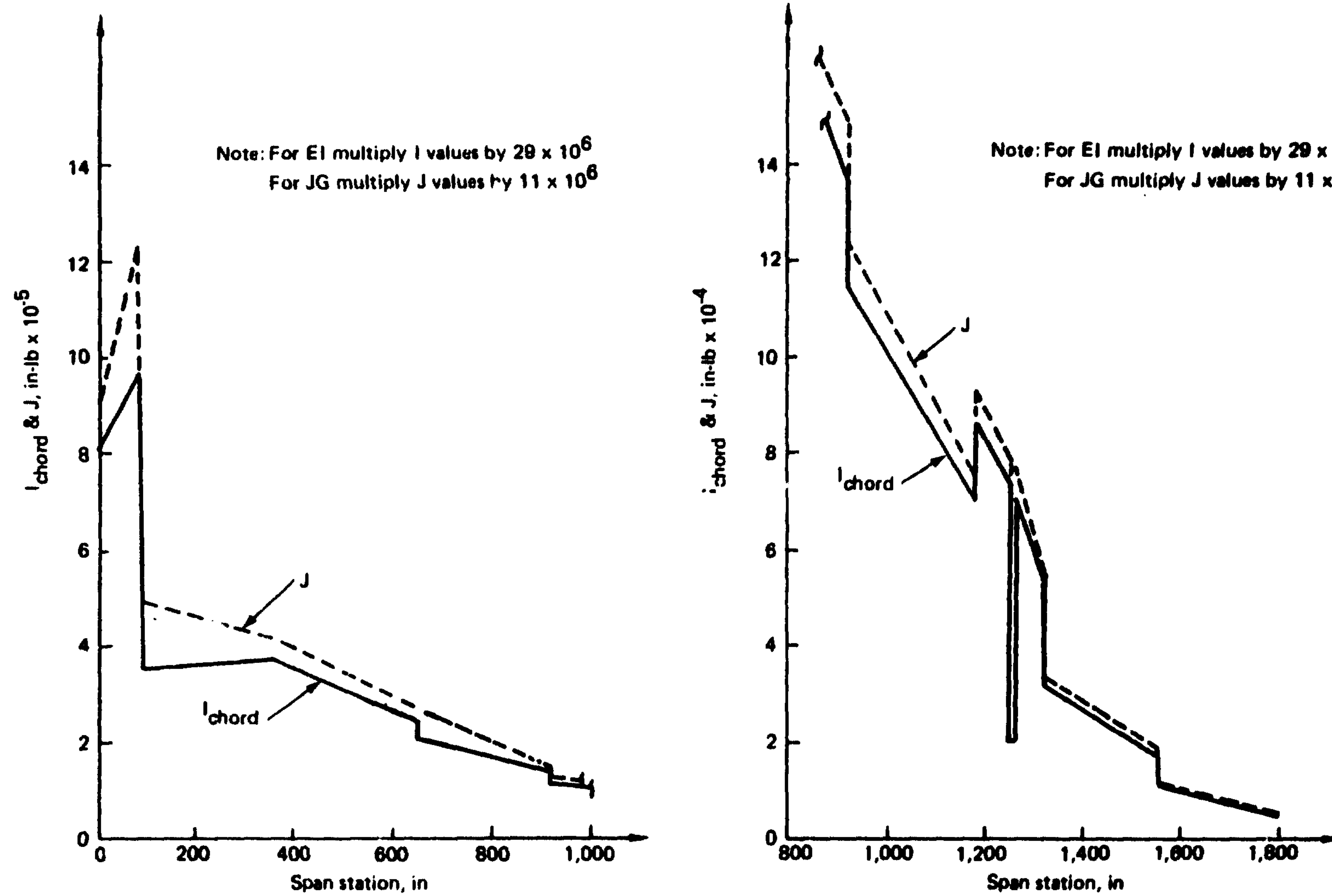


Figure 5-8.3. Blade Section Properties

Fatigue analyses have shown that approximately 70% of the fatigue damage produced by all conditions considered in the analyses (steady state, gusts, as well as start ups and shutdowns at cut-in and cut-out wind speeds) is attributed to the so called "ground-air-ground" cycle. The ground-air-ground condition involves the stress extremes experienced throughout the start-up, operate, and shut-down period. In all the preliminary design work, the maximum stress range predicted for all operating conditions (up to 99.9% probability stress levels) were kept below the design fatigue allowable stress level. The ratio of the allowable fatigue stresses for different weld configurations are presented in Table 5-2.1. The definition of these weld categories may be found in AWS D1.1-79, Structural Welding Code - Steel, Third Edition, pages 132, 133, and 134.

Table 5-2.1 COMPARISON OF FATIGUE ALLOWABLES

FOR DIFFERENT WELD CATEGORIES

WELD CATEGORY	ALLOWABLE FATIGUE STRESS / ALLOWABLE FATIGUE STRESS (FOR "B" WELD)
A'	1.52
A	1.29
B	1.00
C	0.68
D	0.44
E	0.31

Note: 1) A' is for machine base metal with RMS of 125 or less

2) All other weld categories as defined in AWS D1.1-79, pages 132, 133 and 134.

3) Typically, ribs, spanwise and all welds inboard of station 90 are "C" welds and cordwise butt welds (outboard of station 90) are "B" welds.

All operating and non-operating design limit stresses were kept to levels which precluded elastic buckling, material yielding, and permanent deformation. Additionally, the deformations encountered during operation were limited to ensure adequate blade-tower clearance (See Table 4-59). The following section is a discussion of the panel buckling analyses methods used, and the results of a full scale blade test which substantiated the conservative nature of the analyses.

5.1.3.2.1 Panel Buckling Analysis Methods

Those panel structures loaded in compression have been analyzed in the classical way; that is, using the general equation for initial buckling stress (in the elastic range) of $\sigma_{cr} = KE\ (t/b)^2$. The panel geometry and edge fixity are used in defining the value of the constant K. It is conservatively assumed in all analyses, whether the panel be curved or flat, that the edge fixity is "simply supported" (edges are free to rotate, but are restrained from translating). Buckling constants, k, for flat and curved panels are shown in Figures 5-9 and 5-10.

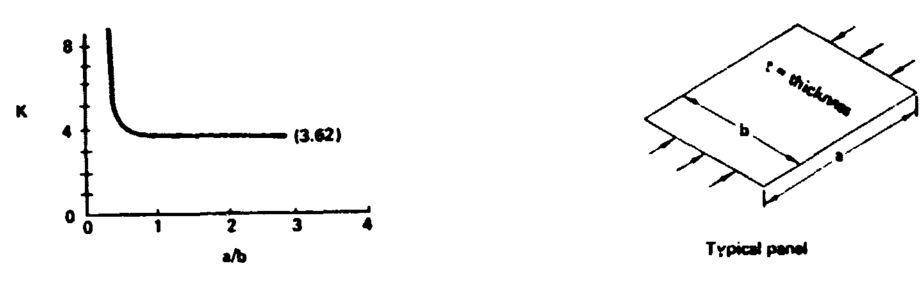


Figure 5-9. Buckling Constant for Flat Panels (Simply Supported)

The radius of curvature of the blade leading edge box skins increases continually with increasing chord station. To analyze the skins for buckling as simple curved panels it is necessary to determine an equivalent constant radius, r . To use the maximum radius of curvature at the intersection with the front spar would be too conservative. A reasonably conservative value of r was determined in the following manner: Using an approximate radius for the panel region just forward of the front spar, and the known panel thickness, an r/t value was determined. Entering the curves of Figure 5-10 with r/t, the value of the curvature parameter, $b^2/rt\sqrt{1-\mu^2}$, was determined at the point where the curve departs from the horizontal (where K begins to increase above the flat panel value).

A value b* obtained from the value of the curvature parameter so determined and the previous values of r and t corresponds to the minimum panel width at which curvature begins to affect buckling strength. A distance of b*/2 was measured forward of the front spar and the equivalent radius of curvature, r, was determined as the radius of curvature at this point from the known section geometry. If the value of r so determined differed significantly from the approximate value used initially, the process was repeated. The point at which the equivalent radius of curvature was taken, according to this procedure, was consistently at the approximate 30% chord. To complete the panel buckling analysis the equivalent panel width b was assumed to be equal to the distance from the front spar to the leading edge. This width invariably gives a curvature parameter well up on the inclined portion of the curves of Figure 5-10, where the critical stress is independent of panel width and is a function of the panel r/t only.

In accordance with system requirements the initial buckling allowable stresses shall be 1.35 times the design limit compressive stresses. The blade structure has two critical limit conditions for compressive stresses; (a) the emergency shutdown condition for the outboard portion, and (b) below rated wind with a gust for the inboard portion. The tower structure also has two potentially critical compression load conditions: (a) seismic zone 3 requirements for the upper 25%, and (b) extreme wind (120 mph @ 30 ft.) for the next 40%.

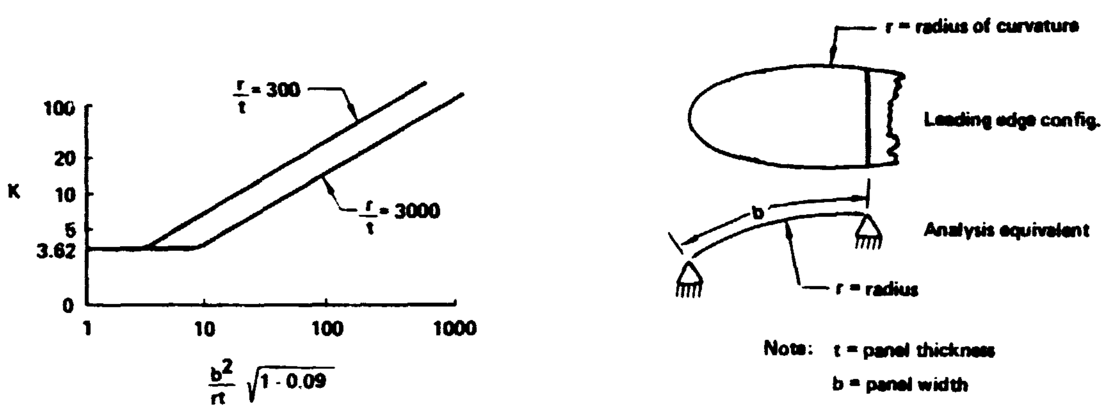
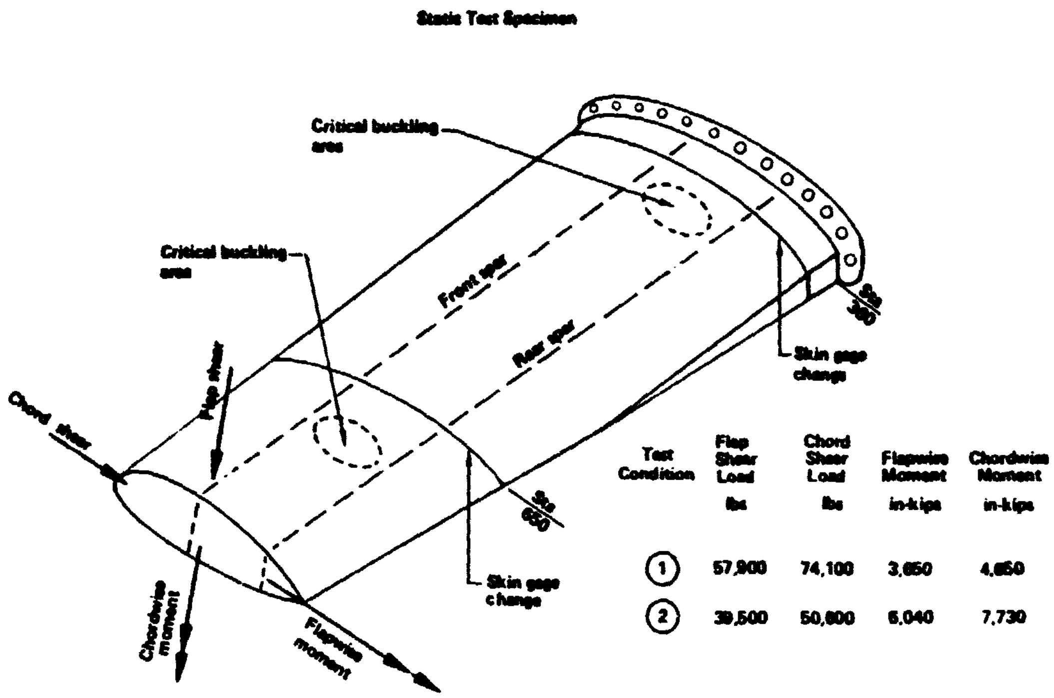


Figure 5-10. Buckling Constant for Curved Panels (Simply Supported)

5.1.3.2.2 Conservative Design Buckling Allowables Confirmed By Test

A bending test was conducted on a mid span representative blade structure, which was to identify fabrication problems, as well as to be used as a means to evaluate the buckling analysis previously discussed. The blade section included the field splice joint (spanwise station 360 inches) and all structure outboard to station 780 inches. The specimen had a special bulkhead at station 780 to accommodate the load-application fixture. The inboard end was attached to a strongback, with the blade's chord plane parallel to the ground. The applied loads at station 780 were in such a combination of transverse shear and couple loads as to make compression panels at both stations, 400 and 670, buckling critical at the same time. (See Figure 5-11.)



Test Condition	Flap Shear Load lbs	Chord Shear Load lbs	Flapwise Moment in-kips	Chordwise Moment in-kips
1	57,900	74,100	3,650	4,650
2	39,500	50,800	6,040	7,730

Figure 5-11. Buckling Test Conditions

The test results have validated the conservative nature of the analytical model, since the specimen was tested to 148% of the predicted ultimate strength of one of the critical panels, without buckling. These results are summarized in Table 5-3.

Table 5-3. Buckling Test Results

	ANALYSIS	TEST
Blade Station	Calculated Ultimate Stress	Maximum Measured Test Stress
400	12,480 psi	18,500 psi
670	8,350 psi	11,220 psi

- Tests have validated the analytical model as conservative
  - Tested to 148% of ultimate Test Load - no buckling
- Local panel limit load deflections were significantly smaller than the allowable aerodynamic requirements.
  - Measured .06 inches <<< allowable of 0.25 inches

5.1.3.3 Drive Train Analyses

5.1.3.3.1 Teeter Trunnion/Low Speed Shaft Analysis

The MOD-2 WTS has a teetering rotor blade which pivots about, and is connected to the trunnion of the low speed shaft by two pairs of elastomeric radial and thrust bearings. The low speed shaft is essentially simply supported within the nacelle by two large spherical roller bearings. The quill shaft, which connects the low speed shaft to the gearbox, provides little if any bending rotation restraint because of its relatively low bending stiffness.

The primary operating loads on the trunnion/low speed shaft are rotor thrust, rotor weight, and rotor torque. The teetering motion of the rotor blade does impose periodic bending loads in the shaft because of the torsional spring rate of the radial elastomeric trunnion bearings. However, teetering angles are ordinarily low and increase with wind cross-flow angle. Also, the phasing of the one-per-rev teetering loads are 90° out of phase with the one-per-rev weight (rotor) moment affects, since teetering moments are a maximum about a vertical axis while the weight moment is maximum about a horizontal axis. Wind cross-flow will alter the teeter moment phase angle relative to the weight moment. Rotor thrust is a function of wind speed and under steady wind conditions is a maximum at rated wind speed.
The design non-operating loads are dead weight loads and extreme wind, while the design limit operating condition loads are deadweight, over-torque, and corresponding rotor thrust.

The stress analyses of the low speed shaft, in general, is based on the ordinary assumptions of plane sections remaining plane. However, the trunnion shaft and that portion of the low speed shaft through which the trunnion shaft passes, are rather complex, and a finite element analysis using NASTRAN was performed on these structures to better define the state of stress in them. The results of these analyses are shown in Figures 5-12 and 5-13.

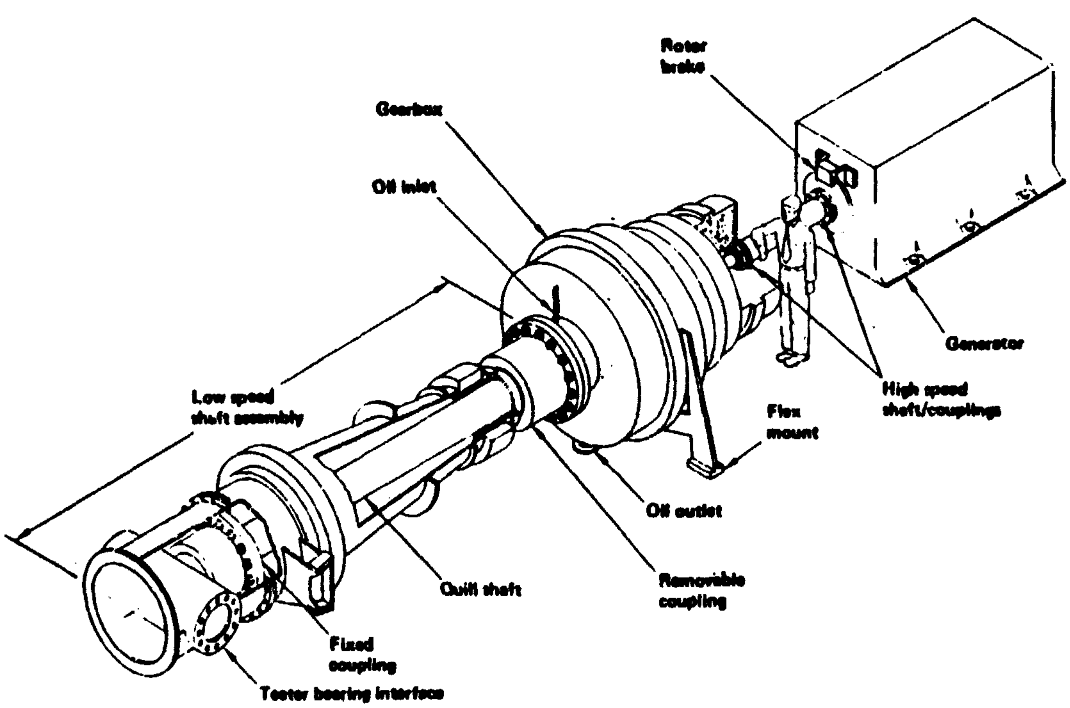


Figure 5-12: Drive Train

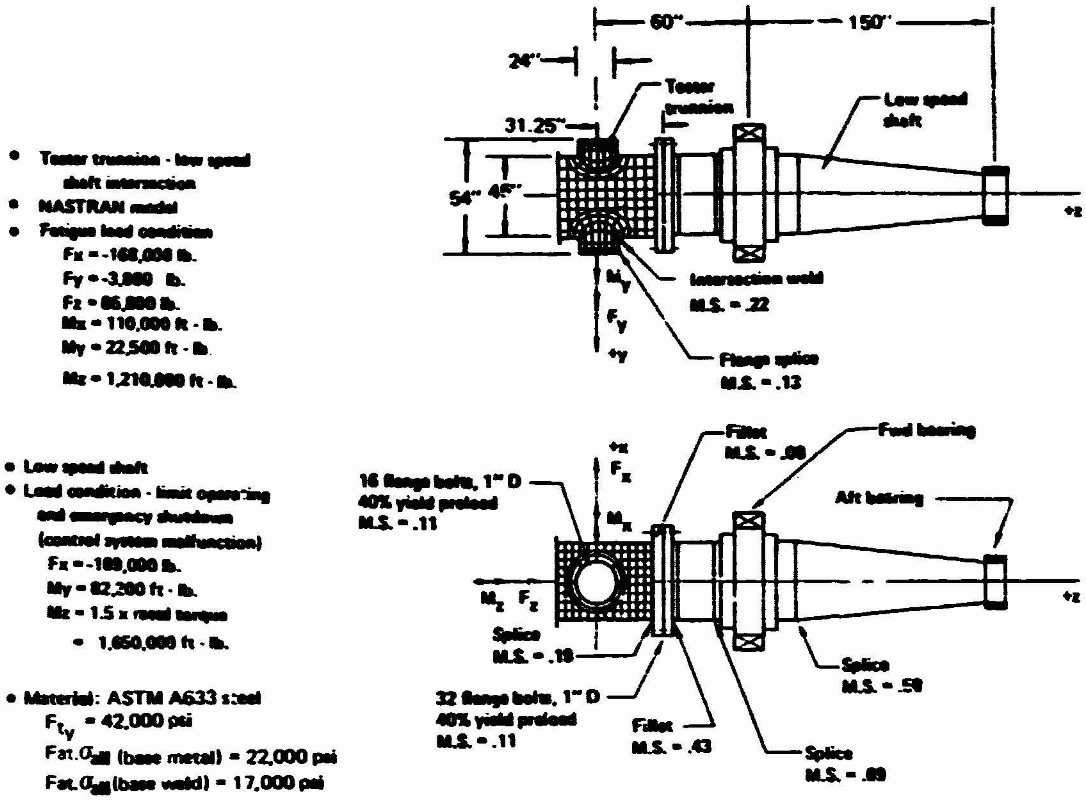


Figure 5-13. Teeter Trunnion-Low Speed Shaft Analysis

5.1.3.3.2 Quill Shaft Analysis

The quill shaft is basically a torque transmitting device, but by the nature of its torsional spring rate, it is also used to minimize the vibratory torques which go into the gearbox/generator systems. Thus most of the two per rev coriolis torques arising in the rotor due to the one per rev teeter motions, as well as the two per rev aerodynamic rotor torques, are reacted as inertia loads in the rotor.

Bending loads do exist in the quill shaft due to the bending of the low speed shaft under the rotor weight, as well as the quill shaft bending under its own weight. The critical design loads and the associated strength margins of safety are presented in Figure 5-14.

5.1.3.3.3 Gearbox/Generator

The generator and gearbox stress analyses have been performed by the sub-contractors supplying them, and will be procured and tested to BEC specifications.

5.1.3.4 Nacelle

The nacelle structural configuration is shown in Figure 5-15. Nacelle design loads arise primarily from the dead weight of the rotor and drive system components, as well as its own dead weight and are shown as Figure 5-16. The rotor thrust load remains fairly local to the aft support bearing, feeding almost directly

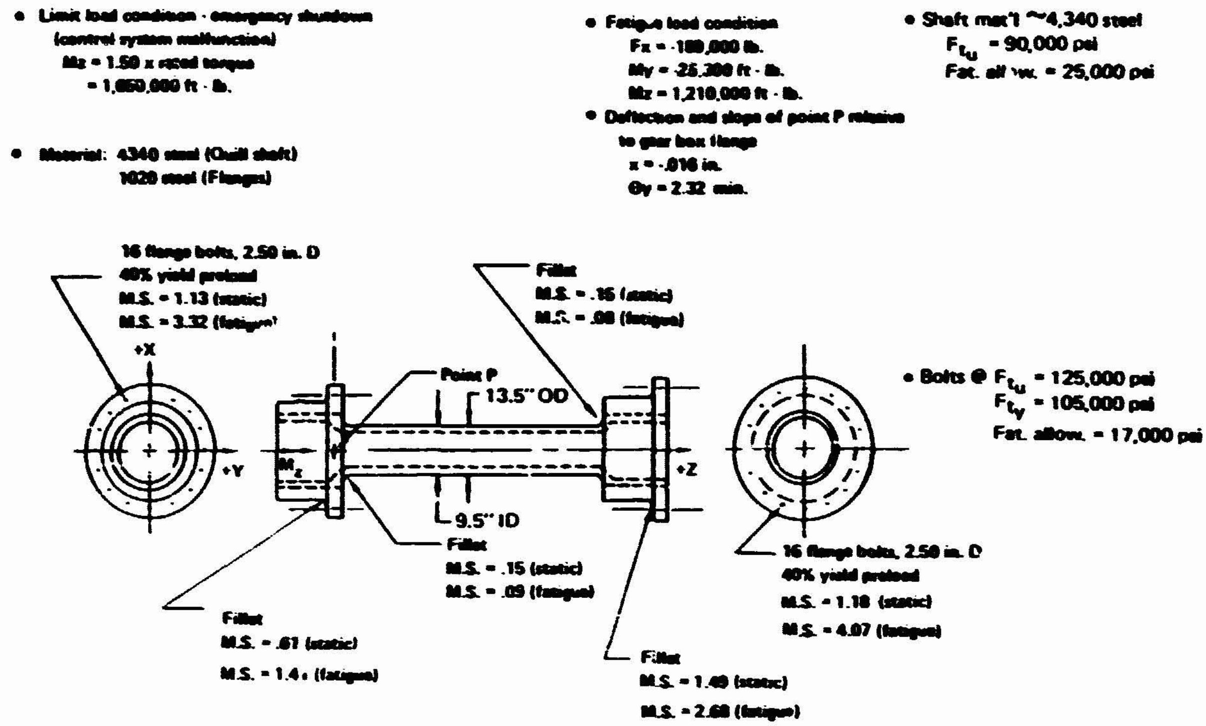


Figure 5-14. Quill Shaft Analysis

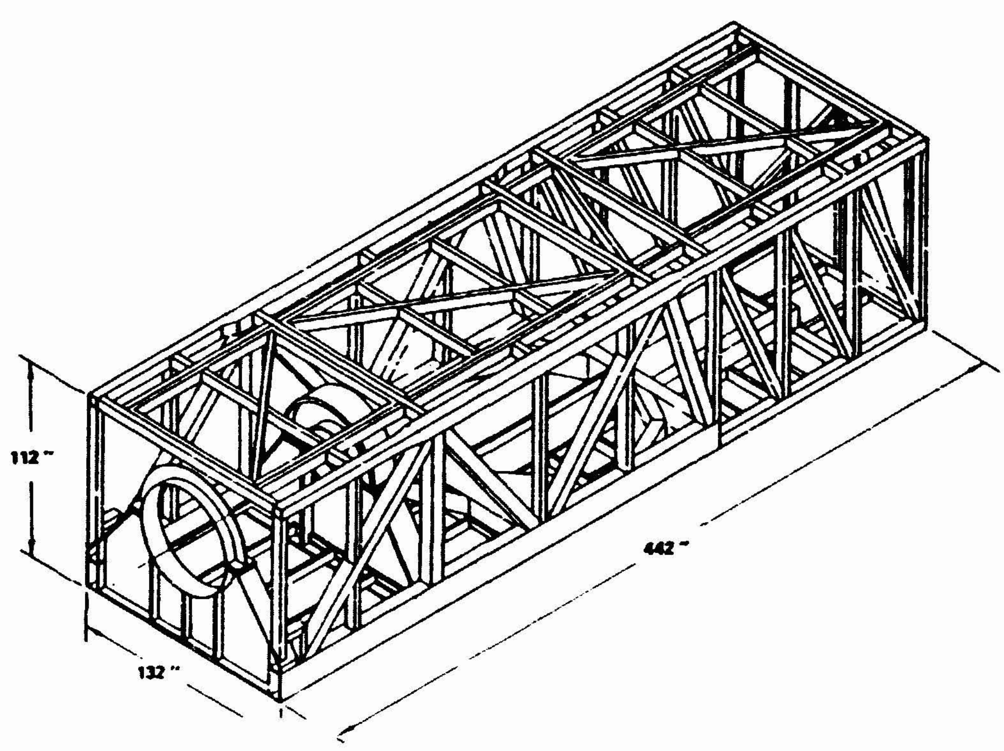


Figure 5-15. Nacelle Structure

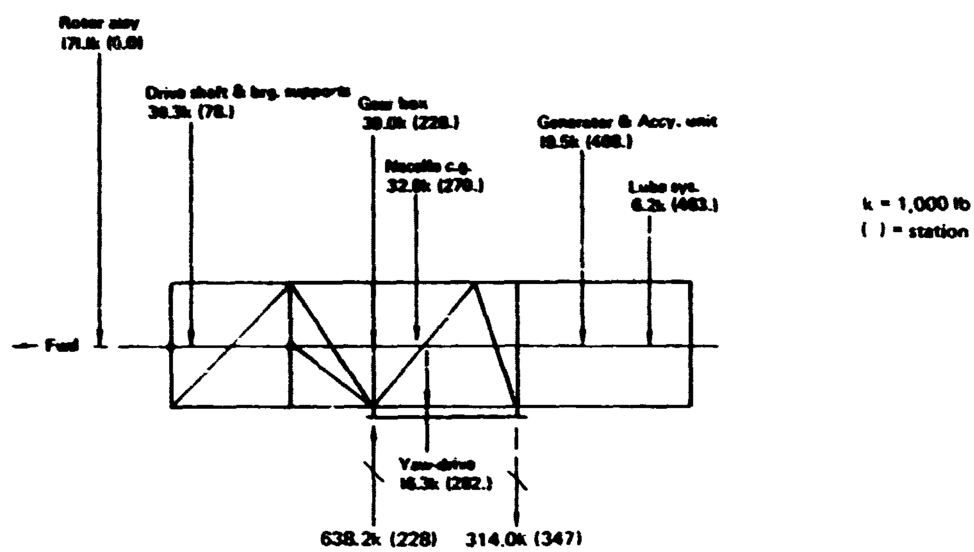


Figure 5-16. Nacelle Component Dead Weight Loading

into the nacelle structure around the yaw bearing. The drive shaft rotor torque is reacted local to the gearbox and it too feeds directly into the yaw bearing. Lateral loads (yaw direction) on the low speed shaft, arising from the rotor during operating as well as non-operating conditions do impose lateral loads into the low speed shaft support bearing, thus, producing lateral bending as well as twisting of nacelle, forward of the yaw bearing. This lateral bending is reacted by the yaw drive structure, feeding through into the tower.

The nacelle structure has been modeled for analysis using NASTRAN finite element procedures. Rotor loads are fed through the shaft support bearings in the nacelle structure, and seismic zone 3 inertia loads are imposed on the nacelle local to each of the drive system components. A summary of element margins of safety are presented in Figure 5-17, based on the NASTRAN analyses and strength allowables from AISC and BEC specifications.

5.1.3.5 Tower/Foundation

The tower structure consists of a 120-inch O.D. cylinder joined to a conical base by a short hyperbolic transition section. Tower wall thickness varies from 1 1/8 inches in the transition region to 1/2 inch near the top in 1/8 inch increments. Due to handling considerations a maximum D/t of 300 was permitted, thus establishing minimum gages of 1/2 inch at the top and 7/8 inch at the base.

The critical load conditions affecting tower design are the extreme wind condition and fatigue under startup/shutdown load cycles. Fatigue governs from the transition region, near elevation 500 inches, to approximately 900 inches elevation. Above 900 inches to approximately 1600 inches, the extreme wind condition is the critical design condition. Above 1600 inches the minimum gage is 1/2 inch. Wall thickness in the conical base, where stresses decrease rapidly due to increasing diameter, is stepped down to the minimum gage of 7/8 inch. Stresses due to zone 3 seismic loads were also examined but did not control any part of the tower design.

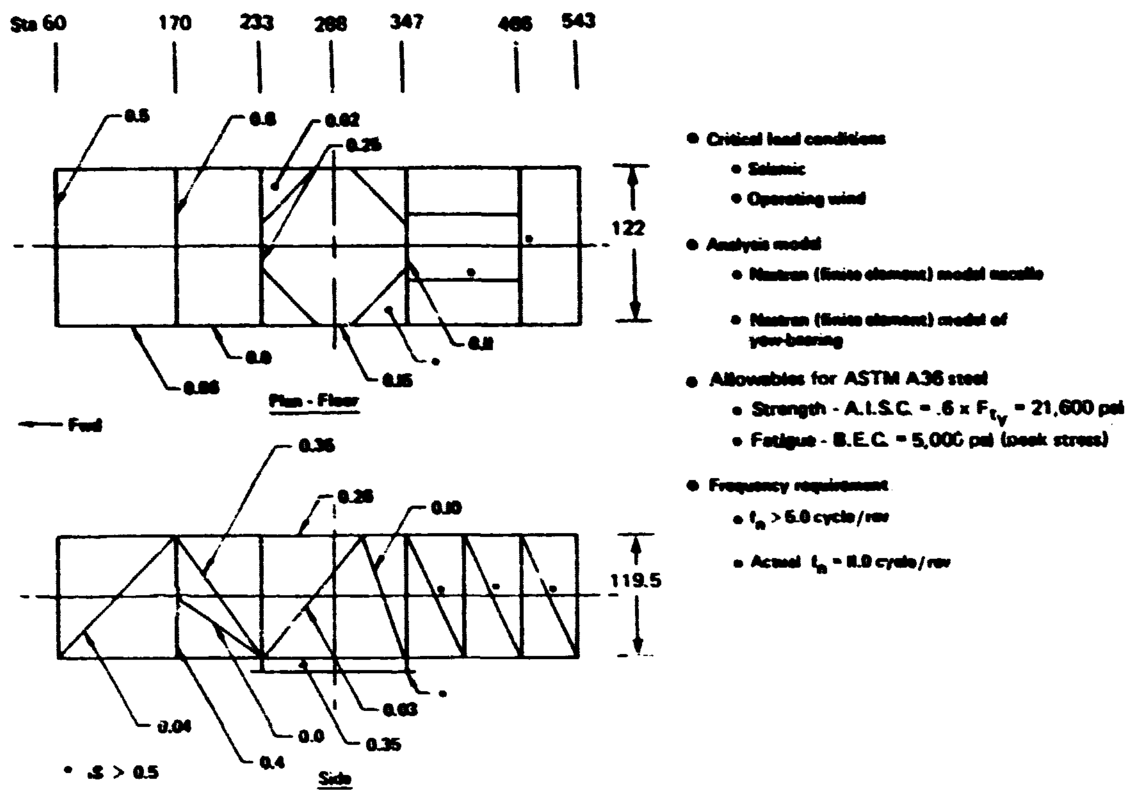


Figure 5-17. Nacelle Primary Structure Margins of Safety

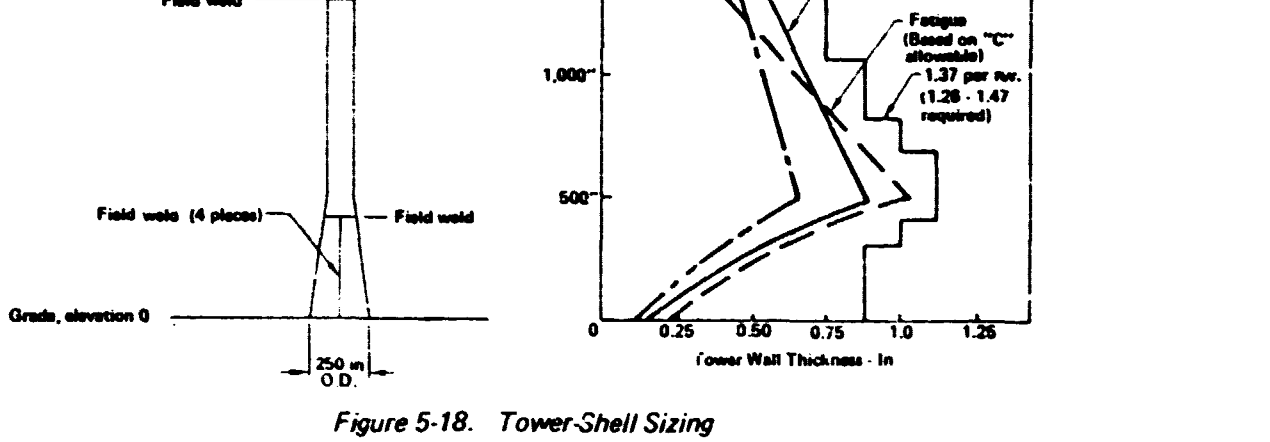


Figure 5-18. Tower-Shell Sizing

Figure 5-18 shows the tower wall thicknesses required vs elevation for the three load conditions mentioned compared to the actual tower design wall thickness. Step changes in wall thickness were located so as to maintain a minimum safety margin of approximately 15 percent for the critical load condition, with minimum gages of 1/2 inch at the top and 7/8 inch at the bottom.

Tower wall stresses were determined by elementary methods from tower bending moments and axial loads except for details near the top where a NASTRAN analysis was made, and for the transition region where a BOSOR4 analysis was made to determine local bending and discontinuity stresses. Static allowable stresses used in establishing required wall thicknesses were determined according to the AISC specifications for members subject to combined axial compression and bending. Fatigue allowable stresses were determined in using established fracture mechanics procedures. Local buckling of the tower wall was evaluated using buckling allowables from Boeing design experience.

Tower system frequency of 1.37 per rev was determined using a NASTRAN beam model with rigid body nacelle and rotor mass representation. This value is well within the permissible range of 1.26 to 1.47 per rev established by acceptable dynamic response limits to 1 per rev and 2 per rev excitation forces.

5.1.4 Load and Stability Analyses

All load and stability conditions that could impose critical structural loading in the MOD-2 WTS have been investigated and are listed in Table 5-4. Each study was carried to the depth necessary to provide adequate structural strength within the operating envelope. The critical load conditions for each component of the WTS are summarized in Table 5-5.

Table 5-4. Load and Stability Conditions

Function		Environment							
		Normal (-40° 120° F)	Gust	Extreme wind	Ice	Snow	Hail	Projectile impact	Seismic
Normal Operating	Startup	1							
	Operating	2	3				4	5	6
	Shutdown	7							
	Parked			8	9	10	11	12	13
Operating Fault	Loss of electrical load	14							
	Control sys. malfunction (1.5 x rated power)		15						
	One tip jammed	16							
	One tip control lost	17							
	Inadvertent braking	18							
Transportation and Handling	Shipping	19							
	Handling	20							
	Erection	21							
Stability	Classical blade flutter & divergence			Flap/lag/torsion					
	Stall flutter			Rotor/tower					
	Rotor/generator			Pitch control feedback					
	Tower vortex shedding			Yaw drive control					

Table 5-5. Component Critical Design Loads

Component	Critical load condition
Blade	Normal operating - fatigue and limit emergency shutdown
Drive train	
Trunnion/low speed shaft intersection	Normal operating - fatigue
Low speed shaft	Normal operating - limit, emergency shutdown
Quill shaft	Stiffness designed
Nacelle	
Nacelle structure	Normal operating - limit, seismic
Yaw bearing	Normal operating - limit
Yaw brake	Parked/extreme wind
Yaw drive motor	Normal operating - limit
Tower	Stiffness designed
Foundation	Parked/extreme wind

5.1.4.1 Normal Operating Loads

The structural loads during normal operation were obtained using the MOSTAB computer code as modified by Boeing. These results provide a trimmed solution that reflects teetering, coupled blade vibration modes, actuator stiffness and a non-linear wind gradient. This gradient was specified by NASA (see Appendix B) and further modified to correlate with wind tunnel test data. A typical MOSTAB generated blade flapwise bending moment distribution is shown in Figure 5-19.

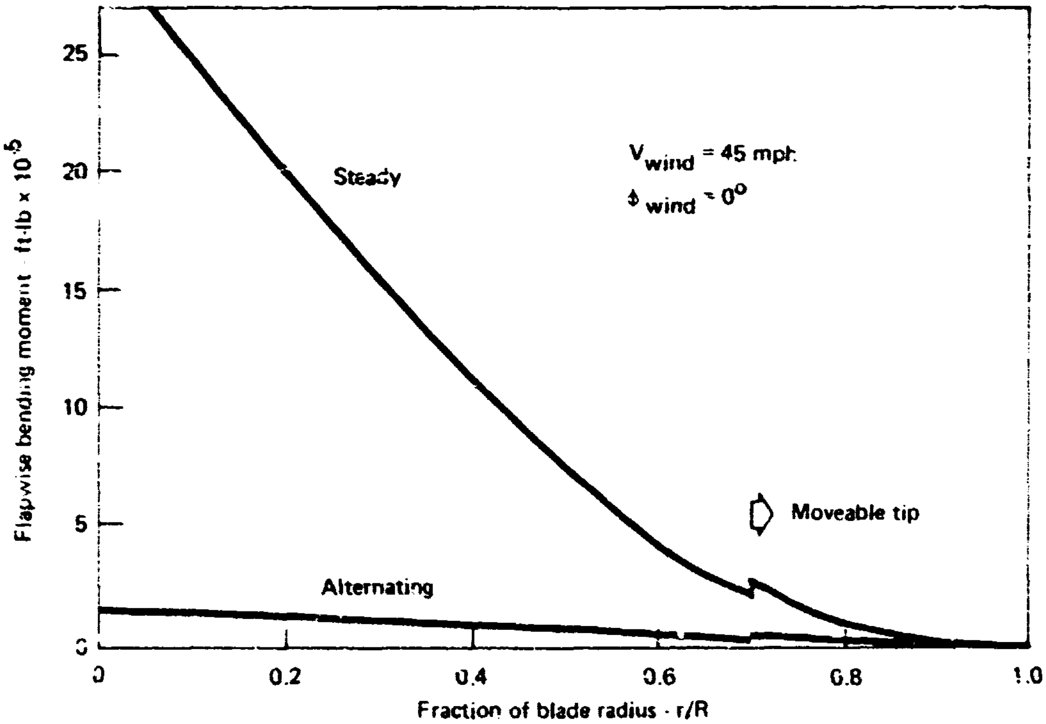


Figure 5-19. Typical Blade Load Distribution

A post-processor addition to MOSTAB was developed to include teeter bearing effects, calculate properly time-phased combined loads and give internal blade stresses at all desired blade locations. Additional computer output provides teeter shaft, drive train, yaw mechanism, bearing and tower loads. Based on MOD-0 experience, design level loads are obtained from gust criteria that calculates loads for both the no gust (steady wind) and 1σ gust (84 percentile) and then extrapolates them lognormally to higher gust levels. Specifically, fatigue loads and limit loads are taken at the 99.9 and 99.99 percentile gust levels respectively. Illustrative gust loads at a particular blade station are shown in Fig. 5-20. First, the variation of blade load with steady wind speed is given; then, loads resulting from impositions of both positive and negative gusts are superimposed on the steady wind curve at selected wind speeds. Below rated wind speed (28 mph) the blade pitch is fixed and large increases in loads due to gust occur. When the effective wind speed exceeds the rated value the blade tip pitch control system is activated reducing the gust loads. The most severe loads occur at 16.5 mph with a limit load gust producing the maximum blade loading before relief is provided by control system actuation.

Thorough load analyses of the rotor in the parked condition were made to ensure that all possible combinations of extreme wind direction and rotor orientation were considered.

5.1.4.2 Operating Fault Loads

A number of operating fault conditions were analyzed based on the FMEA studies. The rationale for those conditions was that they resulted from a single unique electrical or mechanical fault, initiating an emergency shutdown.

Fault Conditions: The following fault conditions were considered:

1. Loss of electrical load
2. Control system malfunction (1.5 x rated power)
3. One tip jammed
4. One tip control lost
5. Inadvertant braking

Loss of electrical load was considered the critical fault because of the possibility of a rotor runaway condition. The other fault conditions are sensed by the control system and an emergency feather is initiated with the generator still on line. The electrical load is dropped only when the aerodynamic rotor torque is reduced to an acceptable low level equivalent to 125 kW.

Shut down analyses were performed to determine feather rates to limit overspeeds without introducing excessive blade loads. It was found that feather rates in the range of 4-6 degrees per second were suitable.

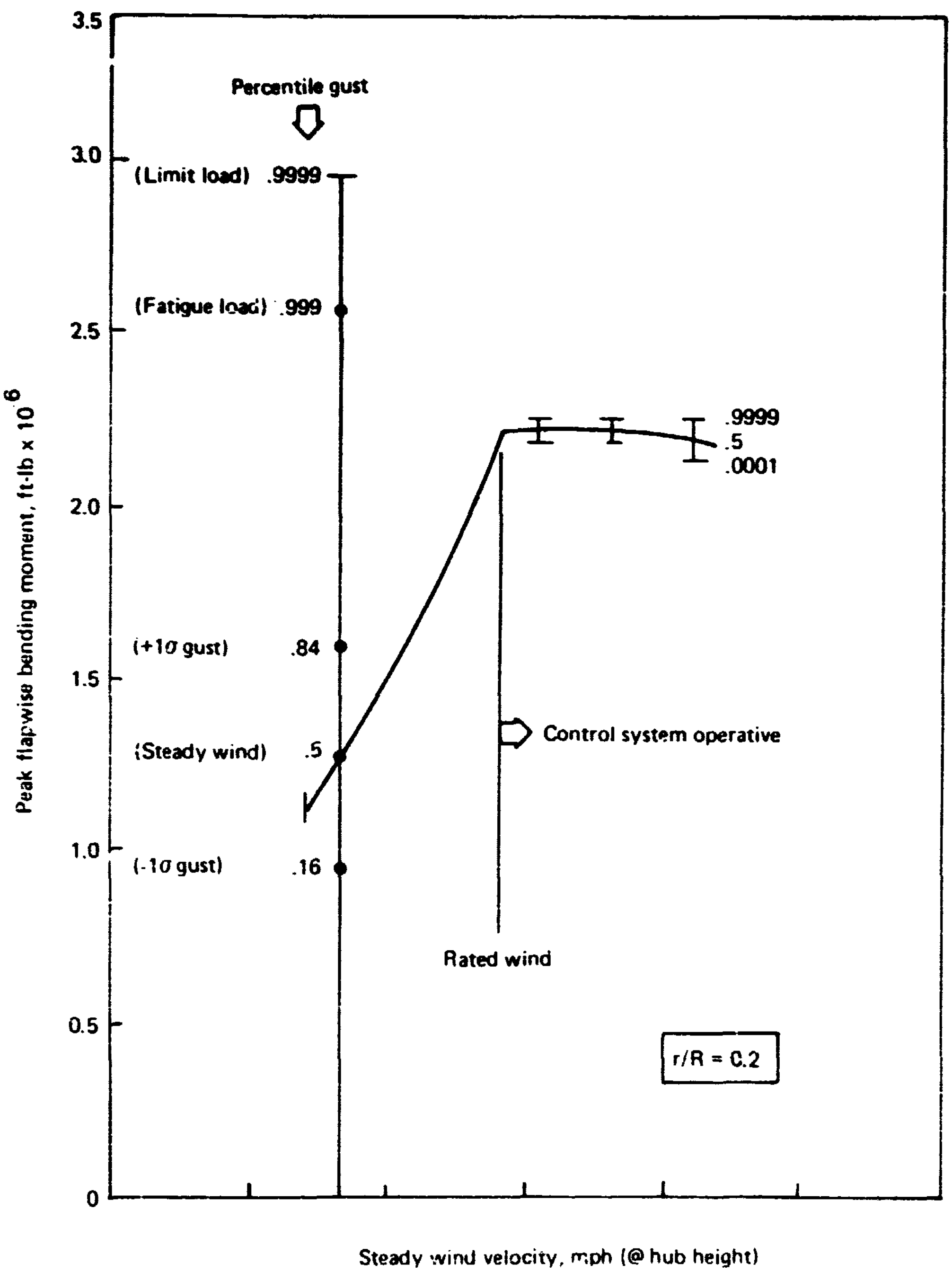


Figure 5-20. Blade Loads Due to Gusts

RESULTS:

Overspeed - Rotor overspeed resulting from loss of electrical load was less than 15 percent. A typical overspeed history vs collective pitch angle is shown in Figure 5-21. Dropping the electrical load at 125 kW did not result in an overspeed condition.

Blade Loads - Aerodynamic braking during emergency feathering resulted in critical reverse flap bending blade loads, particularly in the tip section. The loads produced critical compression buckling stresses in blade sections normally under tension. Ultimate and fatigue stresses due to emergency shutdown have been considered in the blade design.

Control system malfunction allows the power (and blade loads) to increase before a backup power sensor triggers an emergency shutdown. In order to limit blade loads, an emergency shutdown is triggered when the power reaches 150 percent of rated. Blade loads during this condition are on the order of normal operating loads including gusts.

Pitch Actuator Loads - Pitch actuator torque during emergency shutdown is illustrated in Figure 5-22. The actuator torque was found to reverse sign during shutdown, peaking approximately at the time of maximum aerodynamic braking. The pitch actuator torques were found to be sensitive to centrifugal effects induced by blade bending, blade center of gravity location with respect to the pitch axis and collective pitch position. Adequate torque capability was designed into the pitch actuator over its operating range.

Nacelle and Tower Loads - Not Critical

CONCLUSIONS: Critical blade loading conditions result at emergency feathering in response to an operating fault condition associated with loss of electrical load. Feather rates of 4-6 degrees per second limit overspeed and blade loads to acceptable levels.

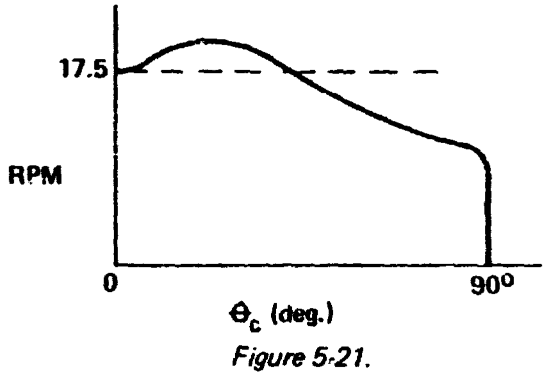


Figure 5-21.
Rotor Overspeed During Emergency Shutdown

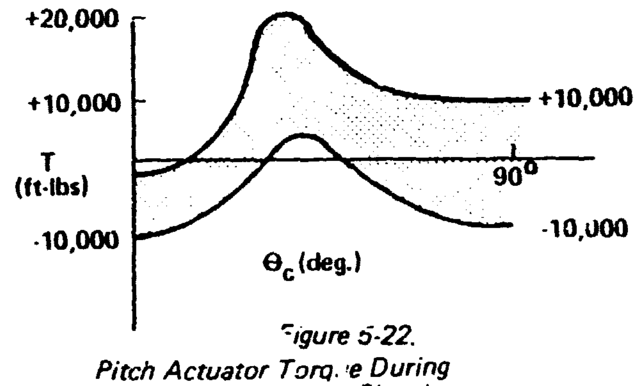


Figure 5-22.
Pitch Actuator Torque During Emergency Shutdown

5.1.4.3 Handling and Transportation Loads

No loads imposed by handling and transportation will design any portion of the MOD-2-WTS.

5.1.4.4 System Frequencies

MOD-2 system natural frequencies are separated from forcing frequencies and other system natural frequencies in order to avoid resonances, which could cause corresponding high forces and deflections. Forcing frequencies for the two-bladed, teetered MOD-2 design occur primarily at one per rev. and at even multiples of rotational frequency, which is nominally at 17.5 rpm. The tower and foundation is designed so that the fundamental bending frequency of the coupled rotor/nacelle/tower/foundation system is greater than 1.2 per rev and less than 1.4 per rev. The lower limit is chosen to avoid possible coincidence with 1P excitation and the upper limit to keep 2P dynamic amplification under unity. The torsional frequency must not be coincident with fundamental system bending frequencies or even multiples of the rotor speed. The tower meets these requirements by having a first coupled system mode of approximately 1.3P (on firm soil) and a first torsional frequency of 5P.

The frequency requirement for the blade is that all natural frequencies be separated by 0.25 cycles/rev from all integer multiples of rotating frequency and by 0.5 cycles/rev from all even multiples of rotating frequency. The steel blade meets these requirements and avoids other system frequencies by having a first flapwise frequency of 2.7 cycles/rev, and a first chordwise frequency of 7.5 cycles/rev.

The drive train avoids forcing frequencies and other system natural frequencies by having an on-line first torsion frequency at 0.5 cycles/rev.

5.1.4.5 Flutter and Divergence

Flutter and divergence studies were made to ensure that both blade tip spindle and pitch control actuator stiffnesses were sufficient for trouble free WTS operation. Results indicate that static divergence is not a problem and that there is at least a 50% overspeed margin for flutter.

5.1.5 Wind Characteristics

The wind environment used for structural analysis is comprised of the following:

1) a steady wind model for a reference elevation
2) a wind shear model
3) a turbulence model

5.1.5.1 Steady Wind Model

The annual distribution of steady wind at the reference height of 9.1 m (30 feet) is given by the following Weibull distribution:

$$P_r(V \geq V_p)Z_r = \exp\left[-\left(\frac{V_p}{C}\right)^k\right]_{Z_r}$$

where $P(V \geq V_p)$ = probability that $V \geq V_p$

V = steady wind speed, m/s

V_p = prescribed value of V

C = 7.06 m/s

K = 2.27

Z_r = reference height = 9.1 m (30 ft.)

5.1.5.2 Wind Shear Model

Wind turbines operate in the atmospheric boundary layer where wind shears are caused by the tendency of the wind to be slowed down near the ground due to surface roughness. The tendency to slow down is increased as the obstructions, or ground characteristics are increased in size.

The wind shear may be expressed in an exponential form. It has been common to characterize sites with one exponent regardless of wind speed. However, recent meteorological studies show that the exponent, α , is increased with decreasing wind speed as shown for several sites in Figure 5-23. The increasing exponent increases the wind velocity more rapidly with elevation. Wind gradients at several wind velocities are shown in Figure 5-24. The use of a variable exponent was recommended in NASA PIR's 70 and 73.

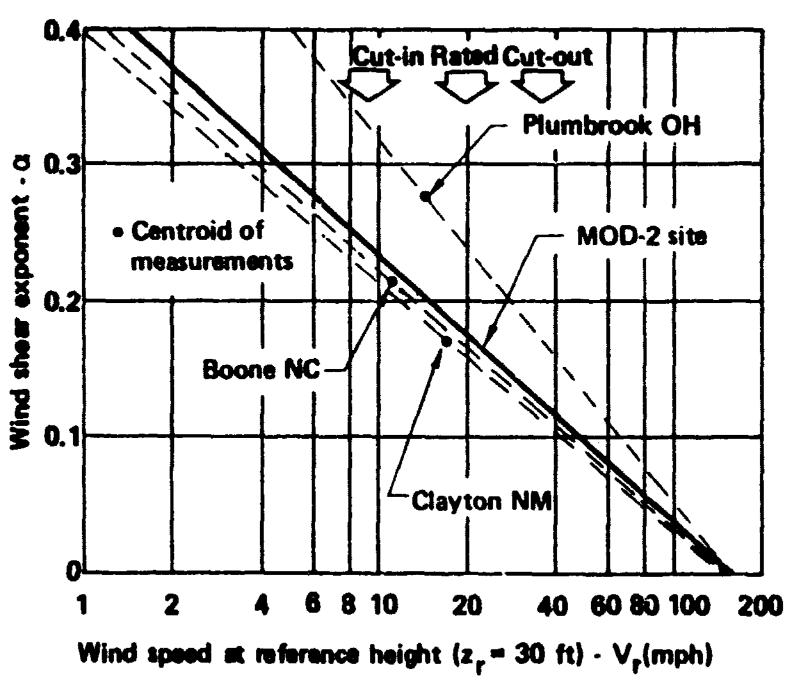


Figure 5-23. Wind Speed Versus Exponent

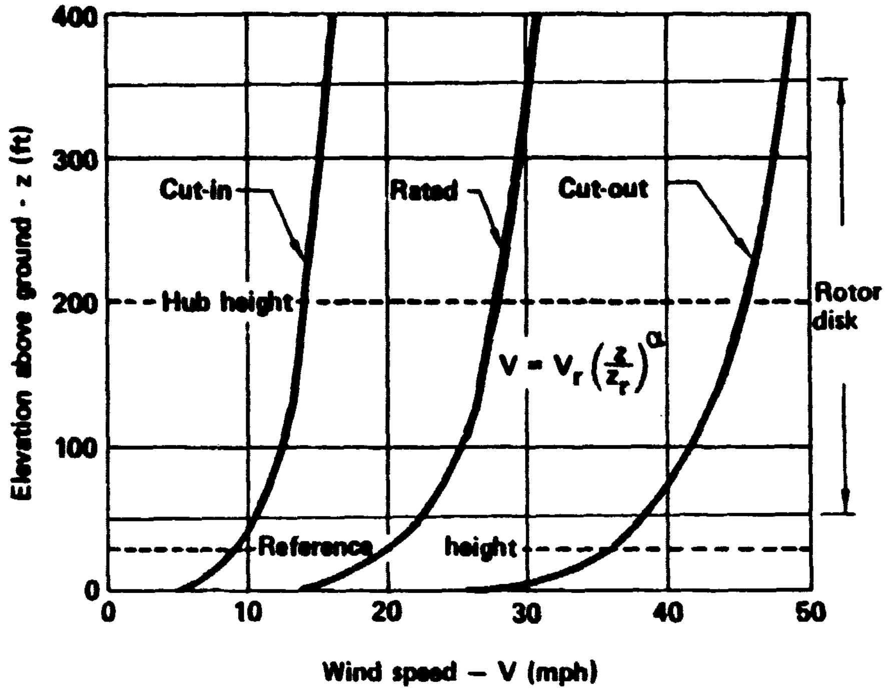


Figure 5-24. Wind Gradients

Steady wind speeds at elevations other than the reference height of 9.1 m (30 ft.) are given by the power law:

$$V = V_{Z_r}\left(\frac{Z}{Z_r}\right)^{\alpha}$$

where V = steady wind speed at elevation

V_{Z_r} = steady wind speed at reference height Z_r = 9.1 m (30 ft.)

and $\alpha = \alpha_0\left(1 - \dfrac{\log V_{Z_r}}{\log V_0}\right)$

$$\alpha_0 = \left(\frac{Z_0}{Z_r}\right)^{0.20}$$

where Z_0 = surface roughness length = 0.05 m (0.16 ft)

V_0 = empirical homogeneous wind speed = 67.1 m/s (150 mph)

5.1.5.3 Turbulence Model

Turbulent wind velocities are functions of steady wind speed, elevation, frequency, and direction. At the reference elevation and a given wind speed, the turbulence is a Gaussian random process with turbulence intensity (i.e., standard deviation of turbulent velocity) given by:

$$\sigma_{z_r,x} = V_{z_r}\left[\frac{1.04}{\ln\left(\frac{z_r}{z_0}+1\right)}\right]$$

$$\sigma_{z_r,y} = 0.8\,\sigma_{z_r,x}$$

$$\sigma_{z_r,z} = 0.5\,\sigma_{z_r,x}$$

where σ_{z_r} = turbulence intensity at reference height, m/s

and subscripts x, y and z denote the longitudinal, lateral and vertical directions, respectively.

The longitudinal turbulence spectrum at an arbitrary elevation is given in terms of the turbulence intensity at the reference elevation by:

$$\phi_x(n,\eta,v) = \frac{\sigma_{z_r,x}^2}{n}\left[\frac{0.164\,(\eta/\eta_{o,x})}{1+0.164\,(\eta/\eta_{o,x})^{5/3}}\right]$$

where ϕ_x = longitudinal component of turbulence spectrum, m^2/sec

n = circular frequency, Hz

η = reduced frequency = $n \cdot z/v$

$\eta_{o,x}$ = 0.0144

Simular equations apply to the lateral and vertical components with:

$\eta_{o,y}$ = 0.0265

$\eta_{o,z}$ = 0.0962

v = steady wind speed (m/s) at elevation (z)

The longitudinal turbulence intensity at any elevation is calculated by:

$$\sigma_x = \left[\int_{n_{min}}^{n_{max}} \phi_x\,dn\right]^{1/2}$$

where the frequency range($n_{min} \le n \le n_{max}$) encompasses the significant response frequencies of the wind turbine system. Similar equations apply to the lateral and vertical turbulence intensities, σ_y and σ_z, respectively.

The upper frequency limit, n_{max}, for a teetering rotor system is set so that high-frequency (small extent) turbulence may be disregarded. The lower frequency limit, n_{min}, for a system having blade pitch control is set so that low frequency (large extent) turbulence may be disregarded.

The upper frequency limit, n_{max}, is that which corresponds to the discrete gust having a circular cross-section which engulfs 50 percent of the disc diameter with 50 percent correlation across the separation distance D/2. All gusts having frequencies less than n_{max} are conservatively assumed to engulf the entire rotor. n_{max} is calculated by solving the following equation for the coherence function, which measures the frequency - dependent correlation of two wind velocities separated by a distance Δ in the disc plane:

$$R = \exp - (Kn\Delta/V_{z_r})$$

The coherence decay rate, K, in the coherence function is given in terms of the steady wind speed. for speeds less than 27 m/s (60 mph) by

$$K = aV_{z_r} - bV_{z_r}{}^2$$

with

$$a = 0.37\ (m/s)^{-1}$$

$$b = 0.005\ (m/s)^{-2}$$

The resulting equation for n_{max} is

$$n_{max} = \frac{-\ln(0.5)V_{z_r}}{D/2(aV_{z_r}-bV_{z_r}{}^2)}$$

The lower frequency limit, n_{min} , is computed from wind turbine dynamic response analysis as being that which corresponds to a discrete gust which produces a variation in rotor torque less than 5 percent from its steady - state value. The discrete gust is assumed to be given by

$$V_{min}(t) = \pm 2\ \sigma_{min}\left[1-\cos\left(\frac{2\pi t}{T_{max}}\right)\right], 0 \le t \le T_{max}$$

where

$$\sigma_{min} = \left[\int_0^{n_{min}} \phi(n)dn\right]^{\frac{1}{2}}$$

$$T_{max} = \frac{1}{2\ n_{min}}$$

If the blade pitch control system is not operating for a particular wind speed, n_{min} is set equal to zero. If n_{min} exceeds n_{max} for a particular wind speed, the effects of wind gusts may be neglected.

The longitudinal gust amplitude has the Gaussian distribution:

$$P(A \ge Ap)_x = \int_{-\infty}^{Ap} \frac{1}{\sqrt{2\pi}\ \sigma} \exp\left[-\tfrac{1}{2}\left(\frac{A}{\sigma}\right)\right] dA \Big|_x$$

where A = amplitude of gust, m/s
Ap = particular value of A

Similar equations apply for the lateral and vertical amplitudes, Ay and Az, respectively.

Discrete gusts are assumed to engulf the entire rotor disc, and to have the shape and period given by

$$V_x(t) = \bar{V}_x + A_x \left[1 - \mathrm{Cos}\left(\frac{2\pi t}{T}\right)\right], \quad 0 \le t \le T$$

where

$$T = \frac{1}{2n}$$

and $\bar{V}_x$ is the mean wind speed.

The turbulence frequency, n, in the above equation is determined assuming a narrow band representation of the turbulence as follows. If there is a system natural frequency in the range $n_{min} \le n \le n_{max}$, n is set equal to that system frequency. If no system natural frequency is present in the frequency range of interest, an equivalent frequency is determined from the following relationship:

$$n = \frac{1}{\sigma}\left[\int_{n_{min}}^{n_{max}} n^2 \, \phi(n) \, dn\right]^{1/2}$$

5.1.6 Material Selection

5.1.6.1 WTS Rotor Blade

A program established to evaluate the various material options for the rotor blade was initiated with a large list of candidate materials as suggested by the Boeing Materials and Processes Staff. The minimum requirements for the material selection were established as shown in Table 5-6.

This list was reduced to six candidate materials as shown in Table 5-7 along with the pertinent technical data. It should be noted that in this field of candidate materials, the use of materials using copper, or with copper added, was considered attractive for weathering corrosion protection. However, during the review of the above materials, it was concluded that copper introduced into the steel may result in large scatter in the Charpy impact results, crevice corrosion, and surface pitting. It was also determined that A-588 steel is worse from a pitting corrosion standpoint than A-572 or A-633 steel without copper. In addition, A-588 would be subject to crevice corrosion painted or unpainted, while A-572 and A -633 would be less susceptible under the same conditions. Hence, none of the materials selected shall be a copper bearing steel.

As a result of this investigation, A-633 Grade A and A-572 Grade 42, Type 1 or Type 2 meet all engineering requirements and can be used interchangeably when manufacturing the rotor. The final selection of a baseline rotor blade material was made on cost and availability.

Table 5-6. Criterion for Materiel Selection

Strength	Yield Strength Ductility Reasonable Charpy @	36,000 PSI Min. 16% Min. Elongation -40° F.
Manufacturing	Good Weldability Min. Bend Dia.	 3t Min.
Quality Assur.	Clean Material [1]	
Materiel	Material for Lowest Life Cycle Cost	

[1] The clean material criterion necessitates the use of desulphurized steel for the reasons noted below:

- Reduces Inclusion Sizes
  - Lowers Risk of Unacceptable Flaw Sizes
- Spherical Shaped Inclusions
  - Allows Acceptance of Larger Inclusions
  - Allows Elimination of Angle Beam Ultrasonic Inspect.
- Cleaner and More Homogeneous Material
  - Reduced Welding Repairs
  - Better Formability
- Improved Mechanical Properties
  - Improved Charpy Values
  - Improved Ductility
  - Improved Short Transverse Properties

ASTM A572 GR 42 Type 1 or Type 2 and ASTM A-633 Grade A steels were the subject of queries to steel mills for prices, deliveries and comments on the relative availabilities through local steel service centers (Distributors).

As the mills will cost various requirements considering variables such as widths and lengths, they were requested to use 1/2 inch x 120 x 420 with 50,000 pounds quantity as a base. Although these prices may not reflect exact MOD-2 procurement costs, they are satisfactory for the selection of materials on the same basis. The prices and extras are as follows:

	ASTM A-572	ASTM A-633
Base Cost	$ .2110/Lb.	$ .2445/Lb.
Normalizing	.0445/Lb.	Included Above
Low Sulphur	.029/Lb.	.029/Lb.
Charpy	.0125/Lb.	.0125/Lb.
Ultrasonic	.035/Lb.	.035/Lb.
Bend Test	No Charge	No Charge
TOTAL	$ .332/Lb	$ .321/Lb.

Table 5-7. Significant Characteristics - Final Material Selection Choices

ASTM No.	Grade	Type	Cond.	F_{tu} min KSi	F_{ty} min KSi	% Elong. in 2"	Bend test ratio pin dia. thick	Charpy test capabil-ity	Weld-ability factor	Corr. resist. factor
A-537	–	CL1	N	70	50	22	1.5	-76°F 20 ft-lb	.60	2
A-572	GR 42	–	A/R	60	42	24	1.0	-40°F 20 ft-lb	.54	1
A-572	GR 42 +Cu	–	A/R	60	42	24	1.0	-40°F 20 ft-lb	.55	2
A-588		ALL	A/R	70	50	21	1.0	-40°F 20 ft-lb	.56	4
A-633	GR A or B	–	N	63	42	23	2.0	-40°F 25 ft-lb	.47	1
A-633	GR A or B +Cu	–	N	63	42	23	2.0	-40°F 25 ft-lb	.46	2

The current mill lead time of each material is approximately seven (7) weeks after contract award to the shipping date from the mills. (One mill, Armco, quoted 10-12 weeks.) Neither material is available in local distributor stocks due to the low sulphur requirements but, if sulphur were not considered, the A-572 would be somewhat more available.

The above costs do not include cost o transportation from the East Coast mill to Seattle. These costs would be \$.0575/Lb. for weights from 40,000 to 90,000 pounds per shipment, \$.0378 per pound for 90,00 to 120,000 pounds and \$.0362 per pound for over 120,000 pounds.

As a result of this study, ASTM A-633 Grade A steel has been selected for MOD-2 rotor blades. This selection (over A-572) is made on the basis of cost, since availability and material characteristics are essentially equal. However, A-572 Gr 42 is considered to be a suitable optional material.

5.1.6.2 WTS Tower

The material selection data for the WTS rotor was also reviewed for applicability to the tower. The material selection criteria for the tower are the same as for the rotor blade with the exception of the Tensile Yield requirements. Since a substantial portion of the tower is designed by maximum tension stress, it is necessary to have a minimum yield strength of 50,000 psi (allowable working stress = 0.6 x Fty). Since ASTM A-572 Gr 50 meets the Tensile Yield Strength of 50,000 psi and the other engineering requirements, it was chosen for the tower material.

5.1.6.3 WTS Nacelle Primary Structure

The material selection data for the WTS rotor blade was reviewed for applicability to the Nacelle. The structure is not subjected to a severe fatigue environment and most members are designed to an allowable working stress similar to the tower (in accordance with the AISC Specification).

Since the design uses structural steel members in an assortment of rolled steel shapes, it is desirable to use an economical and readily available material in existing available shapes. For this reason ASTM A-36 steel has been chosen for the Nacelle Primary Structure.

5.2 CONTROL ANALYSIS

The analysis of the blade pitch control system was accomplished by utilizing a digital simulation of the WTS longitudinal torsional dynamics and control algorithms.

5.2.1 Simulation Description

The simulation was built using the "Easy 5" program, a versatile user oriented simulation program capable of highly detailed dynamic elements and automated analysis techniques.

"Easy 5" consists of two sub-programs. The model sub-program allows the user to assemble a model of the dynamic system to any degree desirable. Many elements of the dynamic system were programmed using "Easy 5 standard components". These "standard components" are generalized subroutines that model various dynamic elements by allowing user input of spring constants, damper constants, inertias and specific characteristics of elements such as generators and regulators. If a standard component did not exist, the element was modeled with fortran statements and equations. The analysis subprogram of "Easy 5" allowed calculation and presentation of some of the following analytical methods; stability matrix, root locus, frequency response, stability margins, state vector for steady operations, and eigenvalue sensitivity.

WTS start-up, below rated wind operation, transition between modes, and above rated wind operation were analyzed while developing the control algorithms required to minimize system disturbance torques and maximize power production.

5.2.2 Pitch Control System Design Goals

This system is designed to meet the following requirement of the WTS System Specification:

a. Below rated wind speeds the control subsystem shall maintain discrete blade tip pitch angles for selected wind speed bands for optimum efficiency.

b. At and above rated wind speeds, the control subsystem shall regulate blade tip pitch angle to maintain rated power output in the presence of steady state and gust wind conditions.

The gust condition used is a MOD-2 WTS "Design Gust" (Vw ± 28% and <13.8 sec duration) which was developed from the wind gust criteria specification in section 5.1.5. It is determined by statistical operations after truncation of the power spectral density curve of expected gusts across the 300 foot diameter rotor to include only gusts below 20 seconds. Truncation of gusts longer than 20 second in duration was based upon the assumption that the blade Pitch Control System would hold gust torque increases at the generator to 5% of the torque (Tr) at rated power production. A "Load Gust" of Vw ± 45% and 20 second duration is the truncation point for which the 1.05 Tr assumption was made.

The drive train and generator characteristics are such that the absolute maximum torque acceptable is 1.5 Tr. Therefore, the "Design Gust" (Vw ± 28% @ 13.8 sec) must produce < 1.5 Tr at the generator. However, the gust profile specified has (1-cos) characteristics and the gust onset for the "Load Gust" and the "Design Gust" are nearly indistinguishable for the first six seconds.

5.2.3 WTS Model Description

A functional block diagram of the MOD-2 WTS torsional dynamics as simulated in "EASY 51" is shown in Figure 5-25. Each block is summarized in the following paragraphs.

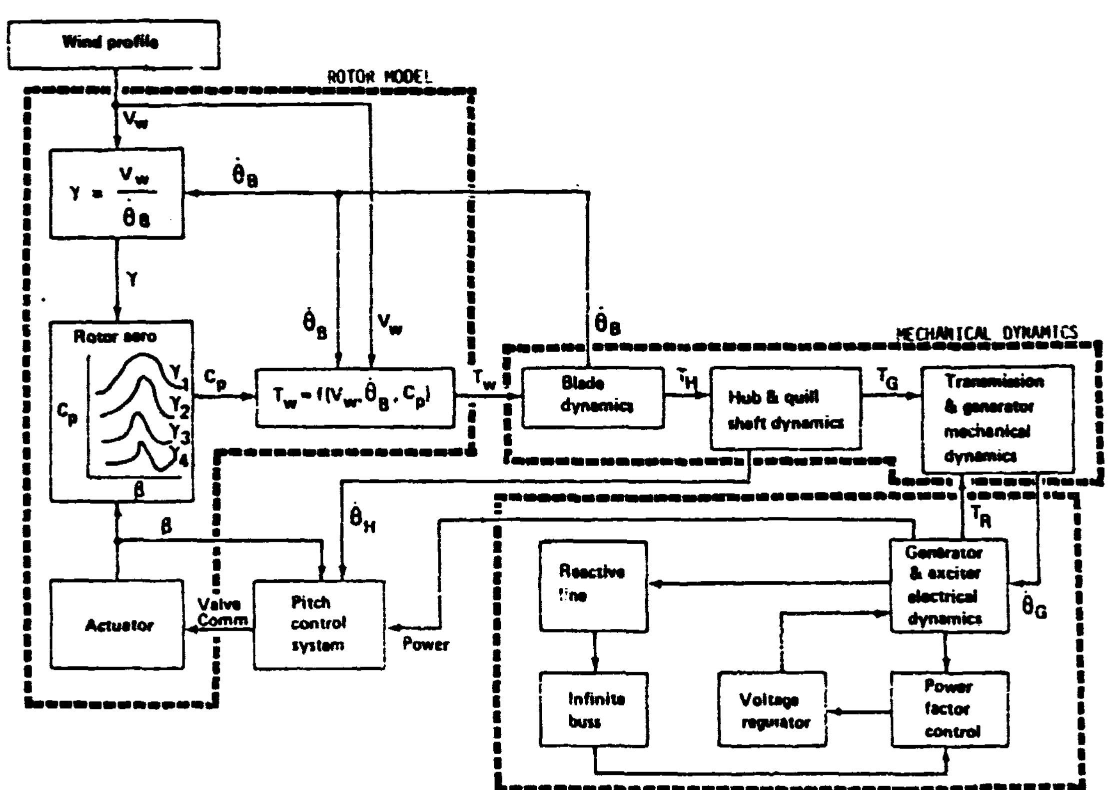


Figure 5-25. MOD-2-107 "Easy" Simulation Block Diagram

5.2.3.1 Rotor Model

Torque on the rotor due to wind inputs was calculated as a function of wind speed, rotor rate and Cp. The equation and elements are:

$$T_w = Cp\, \pi \frac{\rho}{2} \quad V_w^3 \quad R^2 / \dot{\theta}_b$$

where: T_w = Torque extracted from wind (ft-lb)

C_p = Coefficient of power (variable)

ρ = Air density (slug/ft^3) (Constant)

V_w = Wind speed (ft/sec) (variable)

R = Rotor radius (ft) (constant)

$\dot{\theta}_b$ = Rotor rate (RAD/sec) (variable)

5.2.3.2 Mechanical Dynamics

The WTS mechanical dynamics were modeled using two different rotating rates; 17.55 RPM (1.8379 rad/sec) for the blade, hub, quill shaft and transmission front end and 1800 RPM (188.4956 rad/sec) for the transmission backend, high speed shaft and generator. The quill shaft inertia was small so it was distributed between the hub and transmission inertias. Also, the high speed shaft complaince was ignored and its inertia was lumped with the generator and transmission inertias.

The fixed blade first mode natural frequency of the rotor/quill shaft is the dominant mode of the drive train. The configuration of the MOD-2-107 placed the first mode at about .9 RAD/Sec. A blade pitch actuator loop band width of 6.28 RAD/Sec (1Hz) was chosen to provide an adequate response margin for that dominant frequency. A 1Hz actuator loop response is also consistent with hardware available and is well within capabilities achievable if faster response is necessary.

The resulting equations of motion used in the simulation for the Mechanical Torsional Dynamics are:

$$I_B\ddot{\theta}_B = K_{BH}\,(\theta_H - \theta_B) + D_{BH}\,(\dot{\theta}_H - \dot{\theta}_B) + \text{Twind} - D_B\,\dot{\theta}_B = T_B + \text{Twind} - D_B\,\dot{\theta}_B$$

$$I_H\ddot{\theta}_H = K_{HG}\,(\theta_G/\,102.56 - \theta_H) + D_{HG}\,(\dot{\theta}_G/\,102.56 - \dot{\theta}_H) - D_H\dot{\theta}_H - T_B = T_H - T_B - D_H\,\dot{\theta}_H$$

$$I_G\ddot{\theta}_G = -\text{TREACT} - D_G\dot{\theta}_G - T_H$$

5.2.3.3 Electrical Elements

Models consisting of detailed Parks* equation generator and brushless exciter, power factor controller, voltage regulator, reactive transmission line and an infinite bus were included in the simulation to completely describe the electrical power system as it affected the torsional dynamics and therefore the blade pitch control system.

* "Two Reaction Theory of Synchronous Machines", R. H. Park, AIEE Transactions, July 1929, June 1933

5.2.3.4 Blade Pitch Control System

Representation of the blade pitch control system in the simulation is segmented as follows.

5.2.3.4.1 Rated Wind

To minimize reliance on wind sensor output for mode switching decisions, the MOD-2-107 Blade Pitch Control System is switched on blade angle and measured power. For the below rated wind (Vr) operating mode, the blade pitch command is switched to either 3^o or -1^o depending on the measured power. For power conditions producing less than .9 MW output, the fixed blade command will be 3^o. As the power increases in increasing winds, the fixed blade command will change to -1^o when the power exceeds 1.1 MW. From that condition in decreasing winds, the fixed blade command will change back to 3^o when the power drops below .9MW. This overlap (hysteresis) was built in to preclude blade limit cycle for power conditions around the switch point.

5.2.3.4.2 Above Rated Wind (Vr)

Above Vr, blade pitch control utilizes the difference (error) in measured power from the desired (rated) power and the error in measured hub rotation rate and the desired rate.

The blade angle command is produced as the sum of the gain adjusted components of (a) integral of power error (b) proportional power error and (c) hub rotational rate error. The difference between the commanded blade angle and the measured blade angle produces the blade pitch rate command.

To preclude 2P duty cycle of the actuator a narrow notch filter at 2P is placed in the command path to the actuator servo valve. The notch filter in that location assures that blade pitch motion at 2P induced by environmental conditions will be washed out of the signal to the servo valve as well as those 2P signals arising from the blade pitch command equations.

The blade pitch angle will decrease as the wind speed decreases until -1^o is reached around Vr. At that point the mode is set to fixed blade pitch operation and the blade angle command is held at -1^o. At increasing high wind speeds the blade pitch angle will increase until an upper blade pitch angle limit is reached and the system will go into a controlled shutdown.

5.2.3.4.3 Start Up

Because of the aerodynamic nature of the blades at low rotor speeds, the start-up sequence was designed with four separate algorithms that are switched as a function of rotor rate. For blade rotor rates up to five rpm the blade pitch is controlled as a function of wind speed and rotor rate to provide maximum rotor acceleration from static breakaway. From five to ten rpm the blade pitch is controlled for maximum rotor acceleration as a function of rotor rate and wind speed. The third algorithm from 10 rpm to 17 rpm controls the blade pitch to be at an angle to produce zero power at 17 rpm. From

17 rpm to synchronous rate a closed loop integral controller of rotor rate will control the pitch until the electrical synchronizer provides the signal to go on line and produce power. Since these algorithms are designed to accelerate the rotor as rapidly as possible, loss of energy during start-up was minimized. The rapid acceleration also minimizes the excitation of tower modes as the rotor passes through resonant frequencies.

5.2.4 Simulation Results

The investigations listed below were conducted on the model described in section 5.2.3. The results of these investigations are summarized in 5.2.4.1 through 5.2.4.5.

1. Above rated wind gust response 45 mph
2. Above rated wind gust response 28 mph
3. Below rated wind gust response
4. Evaluation of Start Up Algorithms
5. Gain and Phase Margin Evaluation
6. Root Locus Evaluation
7. Digitizing Error Evaluation

5.2.4.1 Gust Response

The system time response to a 1.45 Vw gust of 20 second duration at 45 mph shows the power and reaction torque excursions due to the gust to be about ± 6%. The total blade pitch motion from nominal was an additional 7.5 degrees. Blade pitch rate commands were within ± 1.26 deg/sec.

The response to a 20 second duration 1.45 Vw gust at 28 mph shows the power and reaction torque excursions to be about ± 8%. A 10.5 degree blade angle change is required to unload the excess wind from the rotor. Less than ± 2.8 deg/sec of blade pitch angle rate command were needed under these wind conditions.

The design gust applied at 14 mph shows relatively benign reaction for this fixed blade condition. A torque increase of about 330% at the rotor produces an increase of less than 40% of rated torque at the generator.

5.2.4.2 Start Up and Synchronization

A series of runs were conducted to investigate the mechanical dynamic characteristics during the start up and synchronization process for low wind (14 mph) and high wind (45 mph) conditions. The initial portion of each start up (from zero rpm to 8 rpm) was calculated using the rotor performance program GEM-1. The remainder of the start up process including the synchronization were investigated using the "EASY 5" simulation.

In the 14 mph case, approximately 10 minutes were required to accelerate the rotor from zero rpm to 8 rpm. Synchronous speed (17.5 rpm) was first reached after 13 minutes. The rotor rpm overshoot was less than 2% with synchronization being capable at anytime after that.

In the 45 mph case, approximately 1 minute was required to accelerate the rotor from zero rpm to 8 rpm. An additional minute was then required to reach 17.5 rpm. Again the rpm overshoot was less than 2%.

5.2.4.3 Frequency Response

Gain and Phase relationships as a function of frequency were investigated for 14 mph, 28 mph, and 45 mph.

For fixed blade operation at 14 mph the gain margin is 50db and the phase margin is 10 degrees. For active blade operation the gain margin is nearly 50db and the phase margin is about 25 degrees for 28 mph. The gain margin is about 12db and the system exhibits gain stability for all frequencies for active blade operation at 45 mph.

5.2.4.4 Root Locus

A system root locus plot for variations in Power Error Gain (Powgn) and Rate Error Gain (RTGN) were produced to determine an adequate gain set for the control loop.

The plot showed that decreasing the rate gain provides better damping of the lightly damped pole at 3.4 rad/sec but decreases damping on the first structural mode pole near 2.5 rad/sec. A power gain of .02 was necessary to satisfy the gust response requirements of Section 5.2.1. The chosen combination of gains satisfies both stability criteria and gust response requirements of Section 5.2.1.

Pole sensitivity investigations show that the system remains stable for gain changes of $\pm$ 5%.

5.2.4.5 Sample Data Error Analysis

In the actual control system, conversion errors and digitizing to 12 bits will produce granularity in the converted signals and noise in the most significant bit (MSB) may result in half of full scale perturbations.

A 28 mph simulation with and without the truncation errors associated with digitizing all analog signals in the control system was run. The effects were indistinguishable on power production.

The investigation also showed the magnitude of the disturbance torque at the generator caused by MSB one sample errors is less than 2%.

5.3 SYSTEM PERFORMANCE ANALYSIS

This section explains the approach used in assessing system performance during trade studies, specification and constraints analyses, failure modes and effects analyses, and other analyses performed during the design phase. The cost of electricity (COE), which was a key driver in these analyses, is affected by two major factors -- system cost and the annual energy output of the WTS. System cost assessment is addressed in section 4.4 of this report. The following paragraphs describe the overall performance analysis approach, rotor performance assessment, system efficiency assessment, and energy output analysis.

5.3.1 Performance Analysis Approach

Figure 5-26 illustrates the overall approach used in the performance analysis. The basic input to the analysis includes wind spectra, site conditions, and system efficiency data (including rotor performance). Every effort was made to include test data and inputs from NASA when appropriate. Specific performance data, WTS design characteristics, and annual energy output were determined with the aid of the energy output computer program (EOCP) which is an extension of the Boeing WTS Energy Program (MEO-1) The annual energy output was interfaced with cost data to determine COE (section 5.5)

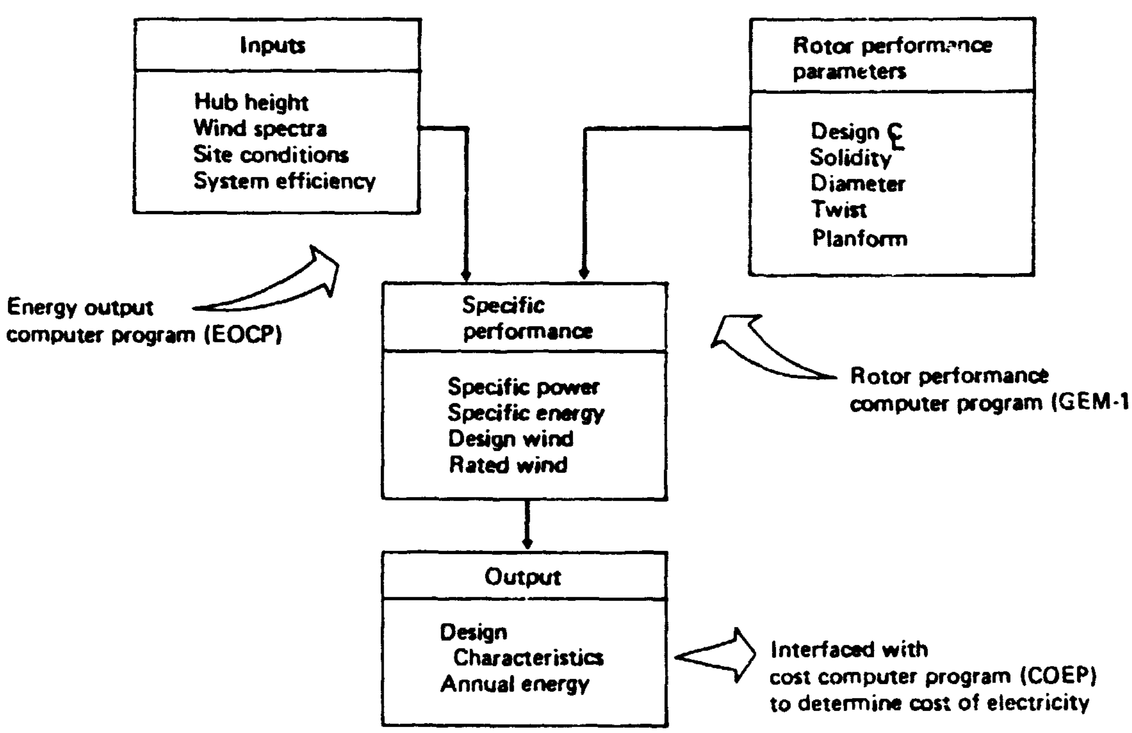


Figure 5-26. Performance Analysis Approach

5.3.2 System Efficiency

Section 3.3 discusses total system efficiency, or power coefficient, which is a combination of the efficiencies associated with the performance - affecting components. The multiplicative nature of these efficiencies is indicated in Table 5-8.

Table 5-8. Maximum System Efficiency (CPM)

Element	Current estimate
Maximum rotor power coefficient (at design wind speed)	.426
Drive efficiency	.974
Generator & electrical efficiency	.950
Heading control efficiency	.993
Blade profile drag loss effected by dirt	.980
Accessory loss	.996
System Cp_m	.382

As can be seen, the components which contribute heavily to the system efficiency are the rotor, drive train, and electrical subsystems. The rotor performance is presented in section 5.3.3, while the remaining components are discussed in section 5.3.4.

5.3.3 Generalized Rotor Performance

For a constant RPM wind turbine system, the rotor has a variable efficiency component. The rotor is most efficient at its design wind speed and is less efficient at its other wind speeds, as a result of many aerodymanic effects. The design wind speed was chosen to maximize annual energy output as discussed in section 4.4.1.4. This variation in rotor efficiency is included in annual energy calculations.

The generalized rotor performance curve is a convenient method for incorporating the variable rotor efficiency into system trade studies. An example of one such curve is shown in Figure 5-27. This particular curve was used to analyze the MOD-2-107 WTS for the PDR presentation. Note that the non-dimensional format of these curves provides flexibility for system trade studies where rotor performance variation is required.

The computer program GEM-1 is used to generate a generalized rotor performance curve. A particular rotor design is inserted into this program, and the rotor Cp is calculated for various combinations of wind speed and collective pitch setting. The calculations include the influence of the vertical wind gradient and aerodynamic losses at the blade tip and pitch interface. The envelope curve formed by the maximum rotor Cp at the various wind speeds is then non-dimensionalized to form the generalized rotor performance curve for that particular rotor design.

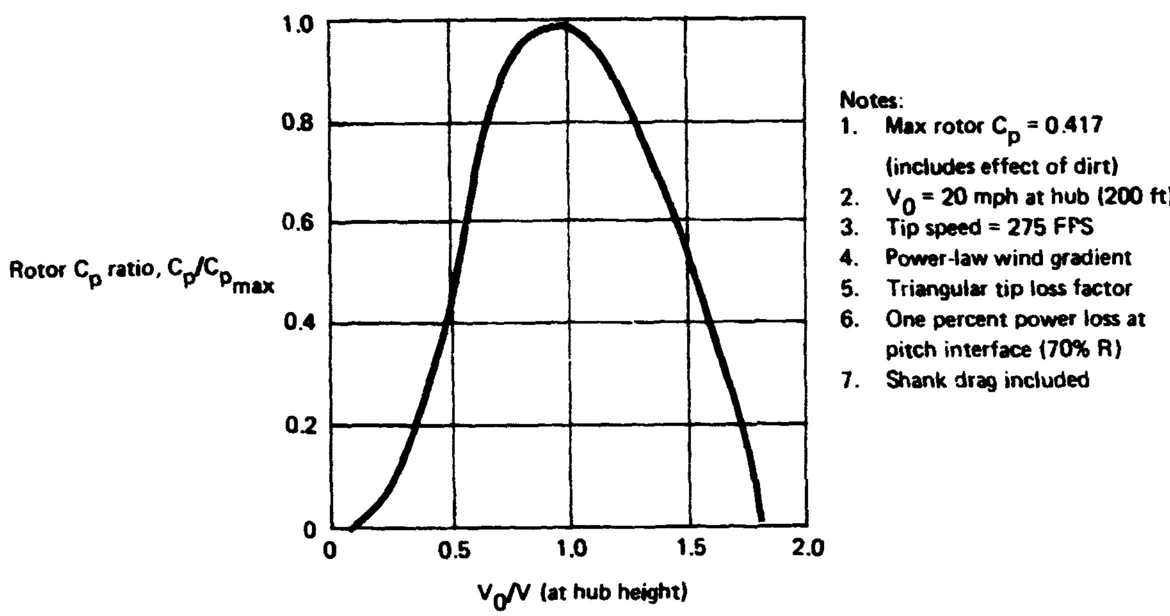


Figure 5-27. Generalized Rotor Performance for the MOD-2-107 Rotor

5.3.4 Remaining Component Performance

The hub and drive losses shown in Table 5-8 are due to friction in the teeter bearings, rotor shaft bearings and the high speed flexible coupling. At part power operations, the loss remains constant. The gearbox power loss is due mainly to gear mesh losses and to windage/churning (air/oil resistance) within the gearbox. As shown in Figure 5-28, the efficiency remains nearly constant down to about half of rated power. The relatively high efficiency is inherent to the epicyclic gearbox design; its gears are compact and the gear tooth contact velocities are low, resulting in low mesh losses.

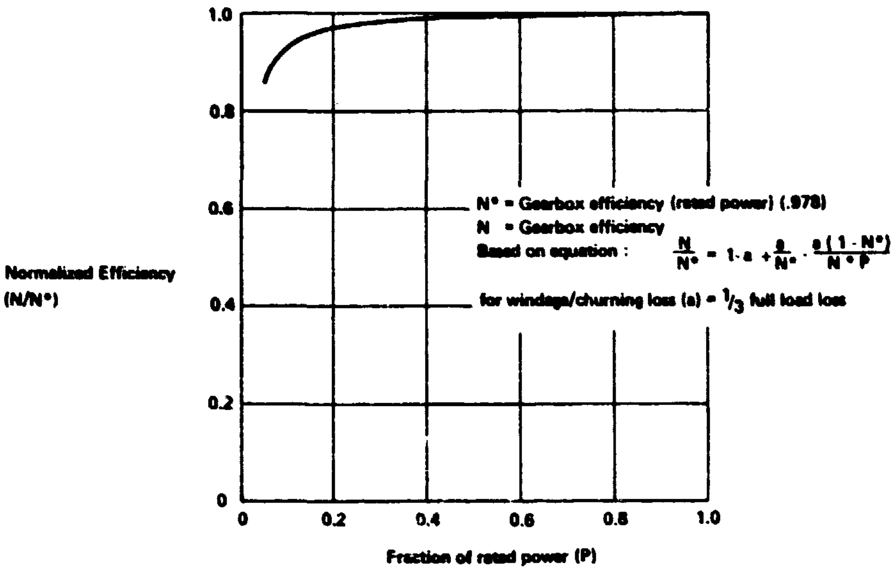


Figure 5-28. Gearbox Efficiency Variation

The generator absorbs power due to windage, field excitation, copper resistance and internal friction losses. For the generator used, the efficiency remains nearly constant as output power is reduced to about half power. The partial power efficiency of the generator is shown in Figure 5-29.

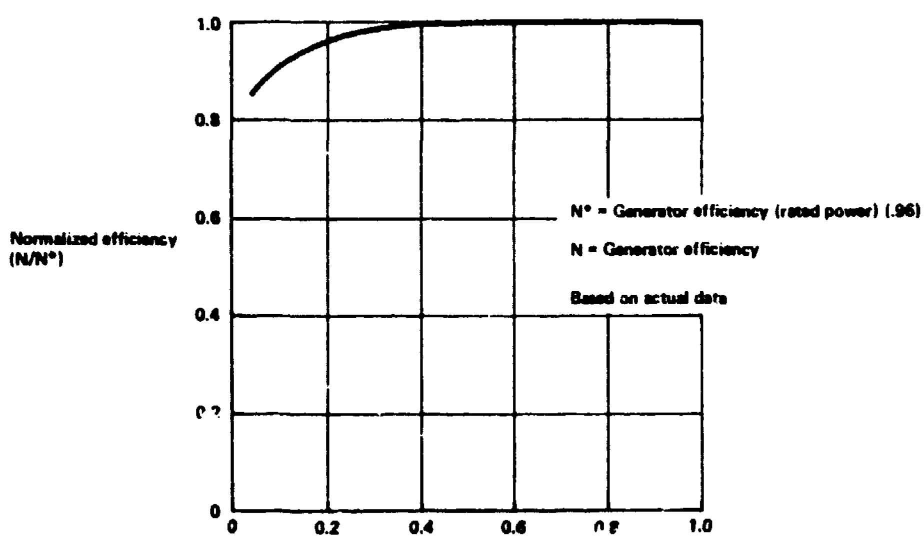


Figure 5-29. Gene. .or Efficiency Variation

The accessory loss listed in Table 5-8 is an average of electric power consumed, which is supplied by either the operating WTS or by the utility grid. It includes power used by electric motors that drive the hydraulic pumps, as well as by instrumentation, control, lubrication pumps, cooling fans, exterior and interior lighting, and by maintenance functions. The value is computed for the duty cycle of the WTS on a site with 14 mph mean speed. The listed line resistance and transformer losses are constant with part power operation.

5.4 SAFETY, RELIABILITY, MAINTAINABILITY AND LOGISTICS

Recognizing the importance of developing a safe and reliable system, considerable effort was expended to ensure that the MOD-2 design was inherently reliable and free of safety hazards to the public and maintenance personnel. Early in the preliminary design phase, a thorough failure mode and effect analysis, (FMEA) covering more than 750 potential failure modes was completed, resulting in numerous design changes to eliminate or reduce the consequences of hardware failures (section 4.5). Detailed reliability failure rate estimates were prepared down to the Tier V level. Each component was assigned a failure rate based on past experience with similar components in the electrical utility and commercial aircraft industries. Trade studies were conducted on individual major components to arrive at a balance between reliability and cost with the objective of achieving the lowest cost of electricity. Selective redundancy was applied where the need was indicated by the FMEA or initial reliability estimates.

In order to achieve a high in-commission rate (availability), it is also necessary to ensure that equipment failures are easily repaired and the system restored to operational status. Each component was examined from the standpoint of accessibility, the need for preventative maintenance, spares support and materials handling. Detailed estimates of repair times and maintenance manhour requirements were prepared and a maintenance concept developed.

Along with the FMEA, a safety gross hazards analysis was performed to ensure that OSHA requirements were met or exceeded and to incorporate safeguards to the public such as aircraft warning lights, an ice detection system and a rotor crack detection system.

The results of these analyses are presented below.

5.4.1 Safety

A MOD-2 safety analysis was conducted to identify personnel hazards associated with the operation and maintenance of the MOD-2 WTS and to ensure that such hazards are eliminated or reduced to an acceptable level. Potential safety problems were identified by reviewing the MOD-2 drawings and specifications, failure mode and effects analysis and preliminary maintenance scenarios. All potential hazards were noted and corrective action implemented where necessary. Since the MOD-2 is an unmanned system, hazards can only occur: (1) from aircraft impacting the tower, nacelle, or rotor, (2) to the public as a result of rotor structural failures, ice being thrown off the rotor, unauthorized entry into an operating system, or (3) during maintenance.

Table 5-9 contains a summary of the potential safety hazards and the applicable corrective actions.

5.4.2 Reliability

The MOD-2 WTS availability goal is .96 with a minimum requirement of .90. Achievement of a .96 availability is realistic when compared to conventional power plants. The primary cause of low availabilities of conventional power plants are the outages due to the fuel fired steam generator which does not have a counterpart on WTS.

Table 5-9. MOD-2 Safety Precautions

ITEM	CORRECTIVE ACTION
Obstruction to Aircraft	• Compliance with FAA Advisory Circular 70/7460-IE, dtd. 11/1/76, "Obstruction Marking and Lighting"
Hazards to Public	
• Rotor Failure	• Safe Life Design • Structural Tests • Crack detection System-Shuts Down WTS
• Flying Ice	• Ice Detection System-Shuts Down WTS
• Unauthorized Entry	• Steel Door, Locked, and Auto Shut Down in Case of Unauthorized Entry
Hazards During Maintenance	• General Safety Design Features • Occupational Safety and Health Act of 1970 (Public Law 91-596) and applicable State Safety Regulations • MIL-STD-1472, Human Engineering Design Criteria for Military Systems, Equipment and Facilities • IEEE Standard 142-1972, IEEE Recommended Practice for Grounding of Industrial and Commercial Power Systems • ANSI C2 American National Standard, National Electrical Safety Code, 1977 Edition • Maintenance Personnel Safety Features • Capability to remove person on stretcher • Fire detection and extinguishing system • Emergency exit doors and "Rescumatic" device to allow egress from either end of nacelle • Ability to lock rotor in horizontal and vertical positions (lock on low speed shaft) • Maintenance scenario and estimated O + M cost assume "buddy" system • Operations and maintenance manuals will contain safety cautions

Cost vs. availability/reliability trade studies were conducted to arrive at design solutions that yielded the lowest cost of electricity. These analyses show that the .96 availability is a cost effective goal. Availability estimates were compiled for all major MOD-2 WTS components down to the piece part level using failure rates and repair times being experienced on similar components in commercial applications.

With the exception of single thread structural items such as the rotor, the MOD-2 has been designed such that at least two simultaneous, unrelated failures must occur before the system sustains severe damage. Wherever economically feasible, fail safe concepts have been incorporated into the design (see 4.5, FMEA). Table 5-10 contains a summary of the availability analysis.

5.4.3 Maintainability and Maintenance Concepts

The design goal of .96 availability can only be achieved if the system is highly maintainable. All MOD-2 drawings have been reviewed by a maintainability specialist to ensure ease of access and to enable each component to be removed and replaced or repaired in place. The support concepts for a 25 unit farm have been developed and are summarized here:

- 2-Shift Coverage-2-man crews 6 days per week, on call Sundays
- Use of outside services for shop repairs, special tasks and heavy equipment rental
- 100% Spares availability
  - Electronics and small items in panel truck
  - Major items stored at utility substation
- Maintenance equipment-portable tools and fixtures, for materials handling $250,000 per farm

Preliminary estimates of the required spares, test equipment and other logistics considerations have been accomplished and are discussed in section 5.4.4.

Detailed maintainability demand frequency and repair time estimates have been prepared. Table 5-11 contains a summary of this data. The major MOD-2 maintenance features are shown in Figure 5-30.

5.4.4 Logistics

The MOD-2 logistics considerations were analyzed to: (1) determine the least cost support concepts, (2) identify maintenance equipment requirements and their impact on the WTS design, (3) determine long lead spares requirements and (4) compute preliminary logistics costs in order to provide a realistic estimate of the cost of electricity. The following items were analyzed:

- Tools and Handling Equipment (including test equipment and installed fixtures)
- Spares Requirements (for initial assembly and checkout, a single unit site and for a 25 unit farm)
- Preventative Maintenance Requirements

Table 5-10. MOD 2 Availability Analysis

Item (Control number)	Number of system failures per year [1]	Mean time to repair (hours) [2]	Average annual outage hours: Hrs./yr.	Average annual outage hours: % of total	Availability
Rotor (4.0)					
Blades and hub (4.1, 4.2)	.82	60	55.8	22.3	.9937
Pitch control mechanism (4.3)	2.11	11	23.2	9.3	.9974
Drive train (5.0)					
Low speed shaft,					
Bearings and					
electrical distrib. (5.1)	.83	60	57.3	23.0	.9935
Quill shaft 7 couplings (5.2)	.15	48	7.2	2.9	.9992
Gearbox & gearbox					
sensors (5.3)	.48	30	14.5	5.8	.9983
High speed shaft, couplings,					
rotor brake (5.4, 5.5)	.23	11	2.5	1.0	.9997
Lubrication system (5.6)	— [3]	—	—	—	—
Generator (5.7)	.09	48	4.4	1.8	.9995
Nacelle (6.0)					
Structure & wind indicators (6.1)	.60	8	4.8	1.9	.9995
Yaw drive system (6.3)	1.40	28	39.8	16.0	.9955
Electrical cables and slip rings (6.6)	.36	9	3.2	1.3	.9996
Generator accessory unit (6.8)	.77	7	5.2	2.1	.9994
Tower (7.0)					
Tower assembly (7.1)	.10	8	.8	0.3	.9999
Electrical cables and					
equipment (7.2, 7.6)	1.06	7	7.4	3.0	.9992
Control subsystem (7.7)	3.0	8	24.0	9.6	.9973
Totals	11.90	20.9	248.3	100	.972
80% of scheduled maintenance (from Maintenance Analysis)			35 Hours		.967

[1] Includes application of 75% duty cycle where appropriate

[2] Includes all causes of downtime (Administrative delays plus hands-on time).

[3] Redundant system for gearbox. Sump system for low speed shaft and generator included in those subsystems

Table 5-11. MOD-2 Maintenance Analysis

Item	Number of unscheduled maintenance actions per year [1]	Repair/inspect time (manhours per year)		Total annual maintenance manhours	
		Scheduled	Unscheduled	Manhours	% of total
Rotor (4.0)					
Blades and hub (4.1,4.2)	.82	28.8	42.0	70.8	20.4
Pitch control mechanism (4.3)	5.22	4.2	75.4	79.6	23.3
Drive train (5.0)					
Low speed shaft, bearings and Electrical distribution (5.1)	1.16	7.0	36.4	43.4	12.7
Quill shaft & couplings (5.2)	.16	–	4.8	4.8	1.4
Gearbox & gearbox sensors (5.3)	.82	13.2	18.7	31.9	9.4
High speed shaft, couplings, rotor brake (5.4,5.5)	.31	2.4	3.0	5.4	1.6
Lubrication system (5.6,	.40	2.4	3.4	5.8	1.7
Generator (5.7)	.08	–	2.8	2.8	.8
Nacelle (6.0)					
Structure & wind indicators (6.1)	.70	4.2 *	9.8	14.0	4.1
Yaw drive system (6.3)	2.88	6.2	26.3	32.5	9.5
Environmental control system (6.5)	.01	.3	.1	.4	.1
Electrical cables and slip rings (6.6)	.36	1.0	2.4	3.4	1.0
Generator accessory unit (6.8)	.77	–	4.3	4.3	1.2
Electrical facilities (6.9)	10.50 [2]	–	3.1	3.1	.9
Tower (7.0)					
Tower subassembly (7.1)	.11	2.6 *	.7	3.3	1.0
Electrical cables, lightning protection and equipment (7.2, 7.4, 7.6)	1.46	1.0	17.4	18.4	5.3
Control subsystem (7.7)	3.2	–	19.2	19.2	5.6
Totals	28.67 (18.17 excluding aircraft warning lights)	72.5	269.8	342.3	100

[1] Includes application of 75% duty cycle where appropriate

[2] Primarily aircraft warning lights

* Excludes painting time which is accomplished at the same time that the rotor is painted.

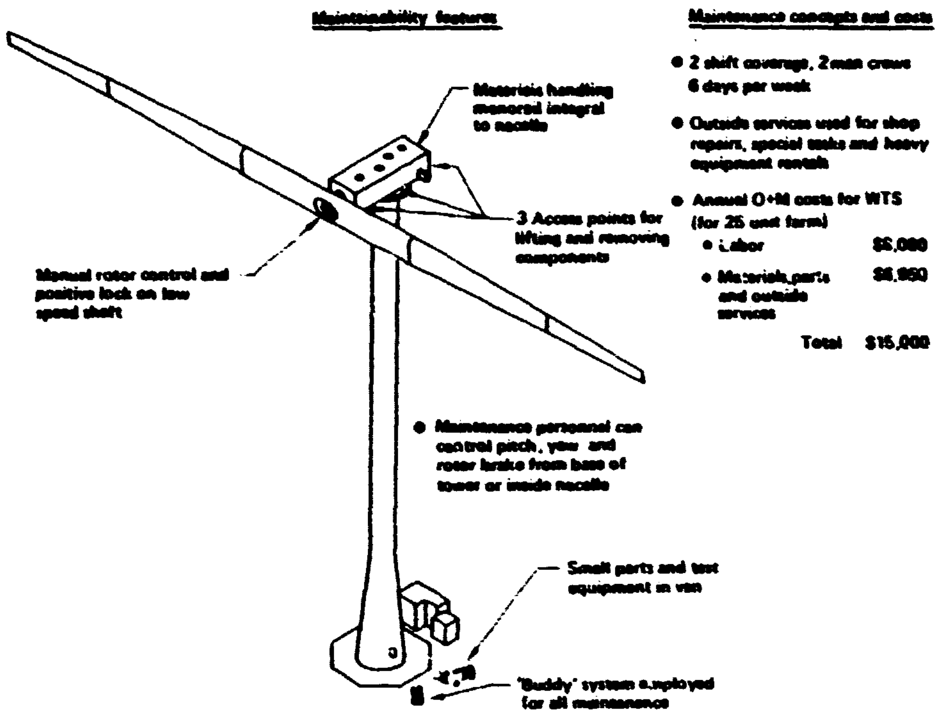


Figure 5-30. Operations and Maintenance Concept

5.4.4.1 Tools and Handling Equipment

Preliminary lists of tools, handling equipment, test equipment and installed fixtures were compiled based on expected maintenance frequencies, the WTS geometry and nacelle floor plan. Tables 5-12, 5-13 and 5-14 contain the preliminary lists of necessary support equipment.

5.4.4.2 Preventative Maintenance

Both system availability and annual O & M costs are partially dependent upon the time required for preventative maintenance. A review of all MOD-2 components at the Tier V level was conducted to identify the required preventative maintenance; the results are summarized in Table 5-15.

5.4.4.3 Spares Requirements

A preliminary spares analysis was conducted to identify the kind and number of spares required for: (1) the initial assembly and checkout of the first system, (2) operation of a single unit, and (3) operation of a 25 unit farm. This analysis is used to identify long lead items and to derive a realistic estimate of projected spares costs. A computer program was developed that

Table 5-12. Tools and Handling Equipment Preliminary List

Item	Used for	Farm size 1	Farm size 4	Farm size 25	Cost per unit (new)	Annual rental cost [1]
Cherry picker (100 ft)	Access to rotor tip controls	R	B	B	$100,000	$5,550
Winch and cables (90-ton)	To remove rotor	B	B	B	80,000	–
Fork lift (10 ton)	Handling generator	R	R	R	–	120
Maintenance truck (2-ton)	To facilitate handling	B	B	B	20,000	–
Fork lift (2½ ton)	To facilitate handling	B	B	B	18,000	–
Trailer (10 ton)	Handling generator	R	R	R	–	30
• Portable crane (10 ton)	Removing generator	B	B	B	3,000	–
Paint sprayer	Painting blade, tower, nacelle	B	B	B	2,000	–
Hoist-on truck (10 ton)	Hoisting generator, gears	B	B	B	1,500	–
Portable hoist-personnel lift	To inspect and repair rotor	B	B	B	1,500	–
Welding equipment (arc)	To repair rotor & nacelle struct.	B	B	B	1,000	–
Hand tools	To repair rotor & nacelle struct.	B	B	B	500	–
• Saddle to hold tip	To repair rotor	B	B	B	500	–
Come-alongs (1,2,3,4 ton)	To facilitate handling	B	B	B	500	–
Misc. elect tools & fittings	To repair electrical equipment	B	B	B	300	–
Hydraulic fluid purge equip.	To purge hydraulic system	B	B	B	200	–
• Scafolding for hub hookup	To inspect and repair rotor	B	B	B	200	–
Misc. ropes, cables	To facilitate handling	B	B	B	200	–
Flood lights on stand	Night work on pad	B	B	B	150	–
Electric winch (½ ton)	To lift boom and other equipment into nacelle	B	B	B	150	–
Stands-hold gearbox gears and case	Gearbox repairs	B	B	B	150	–
Portable heater	Maintenance in cold weather	B	B	B	150	–
• Portable blocks – lock teeter brake	Lock teeter brake	B	B	B	100	–
Oil injection pump	To check gearbox spray patterns	B	B	B	100	–
• Endplate lifting device	Gearbox repairs	B	B	B	100	–
• Spindle extension tools	Gearbox repairs	B	B	B	100	–
• Trunnion tools (1 set)	Gearbox repairs	B	B	B	50	–
Extension cords	Work on rotor, tower and pad	B	B	B	50	–
Hydraulic tube flaring tool	Install/repair hydraulic tubing	B	B	B	50	–
Hydraulic tube bending tool	Install/repair hydraulic tubing	B	B	B	50	–
Standard dollies (2)	To facilitate handling	B	B	B	50	–
Boroscope	Gearbox inspections	B	B	B	50	–
240 Ton ringer	To remove nacelle	R	R	R	–	300
Spools of wire (various sizes)	To repair electrical equipment	B	B	B	50	–
Bolt extractor	Gearbox repairs	B	B	B	50	–
			Total		$231,000	

B = Buy

R = Rent as needed

[1] = Maintenance demand rate x rental cost

• = Unique to MOD-2

Table 5-13. *Installed Fixtures for Materials Handling*

Item	Use
Rotor	
Two eyebolts, one on each rotor tip, to tie stabilization cables (max load - TBD)	Stabilization of rotor during removal
Fixtures on each side of rotor center section, to tie removal cables (max load - 45 tons each)	Rotor removal
Fixture to lock teeter brake	Locking teeter brake
Nacelle	
Eyebolts above low speed shaft on nacelle ceiling	To assist in lifting motor-pump, accumulators and reservoir
Eyebolts above low speed shaft and hold-downs on nacelle floor	To assist in removal of low speed shaft rear bearing and seals
Eyebolts on nacelle at front and rear access doors to allow attachment of snatch blocks for raising and lowering equipment (max. load - 4 tons)	Numerous repairs
Mounting hardware to hold fixtures to accomodate removal of the rotor. Fixtures must hold 90 tons	Rotor removal, rotor inspection and repair, control tip removal
Dual monorail rigging and eyebolts in nacelle	To handle gearbox parts and shaftings/couplings
Nacelle fixtures to accommodate portable crane	Generator removal
Reinforced eyebolts on nacelle for removing generator	Generator removal
Eyebolts on rear nacelle wall to hook on come-alongs	Generator removal
Nacelle eyebolts above yaw hydraulics and above yaw motor	Motor-pump and yaw motor removal
Tower	
Eyebolts imbedded in foundation to allow holddowns for rotor and control tip removals, and to anchor winches and come-alongs	Rotor removal, and to assist in lifting equipment to nacelle
Two bolts imbedded in foundation on each side of the tower beneath the rotor to allow tiedown of scaffolding cables	Rotor inspections and repairs
Bolts imbedded in foundation, near the tower, to anchor 90 ton winch	Rotor removal
Bolts imbedded in foundation beneath rotor to allow stabilization tie down during rotor removal	Stabilization of rotor during removal

allowed a cost trade to be conducted for each item at the Tier V level. The cost to carry a spare(s) must be less than the penalty of lost electricity when the spare is not available.

i.e. cost to carry spare < probability of failure x value of lost power

$$.18\ (\text{cost of spare}) < \left[1 - \left((e^{-\lambda t})(1 + \frac{\lambda t}{1!} + \frac{(\lambda t)^2}{2!} \cdots + \frac{(\lambda t)^n}{n!})\right)\right] \times \text{MDT} (\$35/\text{hr})$$

where n = number of spares, t = reorder time,
λ = failure rate, MDT = downtime

Table 5-16 contains a summary of the preliminary spares list.

Table 5-14. Test Equipment, Preliminary List

Item	Used for
• Ultra sonic tester	Rotor inspections, nacelle critical welds
• Dye penetrant	Rotor inspections
• Multimeter	Electrical trouble shooting
• Hi-pot tester	Electrical trouble shooting
• Viscosity tester	Gearbox oil testing
• Demulsibility tester	Gearbox oil testing
• Contamination analysis equip.	Gearbox oil testing
• Oil changing equipment	Gearbox and yaw control gearbox
• Portable wind speed indicator	To check nacelle wind
• Hydraulic oil changing equipment	Speed indicators
• Instrument to check yaw brake pressure	Yaw brake checks
• Ammeter	General trouble shooting
• Instrument to measure hyd. oil temperature switch operation *	Pitch & Yaw switches
• Heat source (small blower)	To check air temperature sensors
• Timer	To check critical time constants
• Electrical-relay testers	To check generator protection relays
• Synchronizer tester *	For trouble shooting
• Dual trace scope, logic analyzer *	For trouble shooting

* Unique to 25 unit farm (outside services used for single unit)

Table 5-15. MOD-2 Preventative Maintenance, Preliminary List

Item	Maintenance Action	Interval (months)	Time required (hours)	Annual time required (hours)
Rotor (hub & tips)	Inspect for cracks	12	8.0	8.0
Rotor, tower, nacelle*	Paint	108, 52	168.0, 8.0	20.0
Pitch hydraulics	Visual for leaks & fluid level	2	.3	1.8
Pitch filters	Check for contamination	2	.3	1.8
Rotor pitch change mechanisms and pitch position indicator	Check for leakage and proper relief valve operation; calibrate pos. ind. (done in conjunction with rotor crack inspection)	12	1.0	1.0
Low speed shaft bearings & seals	Check condition & lubrication	2	.5	3.0
Teeter stop & caliper	Check condition	6	1.0	2.0
Slip rings	Check condition	12	2.0	2.0
Gearbox	Check oil (test) and all sensors:	6	2.0	4.0
	Particle detector	2	.3	1.8
	Oil pressure	2	.1	.6
	Oil temp switch	6	.1	.2
Gearbox (cont.)	Oil Filter filter	2	.1	.6
	Gearbox oil flow switch	2	.1	.6
	Check gears	12	2.0	2.0
Gearbox oil filter	Change	2	.5	3.0
Rotor brake disc	Check condition	2	.1	.6
Rotor brake unlock switch	Check condition	2	.1	.6
Rotor brake overtemp switch	Check condition	2	.1	.6
Gearbox lubrication system	Check condition	2	.2	1.2
Bearing lubrication system	Check condition	2	.2	1.2
Yaw bearing and mounting	Visual for deformation	12	1.0	1.0
Nacelle control unit power supplies and reference voltages	Calibrate	6	.5	1.0
Nacelle overheat detector	Check operation	12	.3	.3
Wind speed indicators	Calibrate	6	.5	1.0
Yaw disc brake pressure	Check	2	.5	3.0
Yaw hydraulics	Check for leaks	2	.1	.6
Yaw brake discs	Check for wear	6	.3	.6
Yaw hydraulic oil level	Check	2	.1	.6
Yaw hydraulic oil filters (2)	Change	6	.2	.4
Yaw & pitch over-temp switches (2)		60	1.0	.2
Air-conditioner thermo-stat	Check	12	.3	.3
Battery	Check	2	.1	.6
Battery	Replace	60	2.0	.4
Tower	Visual inspection	12	1.0	1.0
Ground intrusion system	Test	2	.1	.6
Bonding straps - check for corrosion & connection integ.	Visual inspection	12	2.0	2.0
Fire detection and extinguishing system	Check and service	6	.5	1.0
Synchronizer	Check condition	2	< .1	Negligible
Ground current relay	Check condition	6	.2	.4
Differential protection relay	Check condition	6	.2	.4
Fail safe system	Test **	2	.1	.6
			Total	72.6

* Rotor, tower and nacelle painted at same time. Rotor control tips painted at 4 1/2 year intervals.

** Also test fail safe system after recharging accumulators

Table 5-16. Spares List - Preliminary

Item	Number of spares		
	Single unit		25 Unit farm
	A & CO *	Operations	
Rotor			
Blade	–	1	1 (One side)
Hub	–	1	1
Tip Pivot Spindle			
Bearings	1	2	3
Seals	1	2	3
Pitch change mechanism			
Actuators	1	2	3
Reservoir	1	2	4
Accumulator	1	1	2
Start-stop valve	1	2	3
Relief valve	1	2	3
Teeter brake & release valve	1	2	3
Servo valve	1	2	3
Shuttle valve	1	1	2
Check valve	1	2	3
Filter	2	2	50
Motor	–	1	2
Pump	1	2	3
Position sensors	1	2	3
Pressure switch	1	2	3
Level switch	1	2	3
Air pressure switch	1	2	3
Temperature switch	1	1	3
Pitch lock sensor	1	2	3
Feather lock mechanism	1	1	2
Fasteners, rod ends	1	2	3
Drive train subassembly			
Low speed shaft and bearings			
Radial bearing	–	1	1
Teeter bearing	–	1	2
Rad./thrust bearing	–	1	1
Teeter stop (caliper, elastomer)	–	1	3
Slip rings	Repair kit	Repair Kit	Repair Kit
RPM indicator	1	1	3
Shaft position indicator	1	1	3
Quill shaft couplings	–	0	1
Gearbox	–	0	Selected bearings and high speed gears
Particle det. switch	1	1	2
Oil pressure switch	1	2	3
Oil temp. switch	1	1	2
Filter pressure switch	1	2	3
Gearbox oil flow switch	1	2	3
High speed shaft and couplings			
High speed shaft	–	1	1
High speed shaft couplings	–	1	1
Rotor brake			
Caliper	1	0	1
Disc	1	1	10
Unlock switch	1	1	2
Overtemp. switch	1	1	2
Lubrication system			
Gearbox lubrication M-pump	1	1	3
L.S. shaft lub. M-pump	1	1	3
Gearbox filter	2	2	10
Generator	–	0	1 Set of bearings
Nacelle			
Nacelle instrumentation			
Nacelle intrusion device	–	1	2
Overheat detector	1	1	2
Position indicator	1	1	2
Wind speed indicator	1	1	3
Wind direction indicator	1	1	3

* Assembly and checkout

Table 5-16. Spares List - Preliminary (Cont)

Item	Number of spares		
	Single unit		25 Unit farm
	A & CO	Operations	
Yaw drive system			
Brake disc	2	2	10
Pinion	1	1	2
Gearbox	–	1	2
Hydraulic motor	1	1	2
Hydraulic pump & motor	1	1	2
Reservoir	–	1	2
Heat exchanger	–	1	1
Accumulator*	1	1	1
Relief valves	1	2	5
Solenoid valves	1	1	2
Control valve	1	1	2
Yaw # 1 oil filter	2	2	50
Yaw # 2 oil filter	2	2	50
Check valves	1	2	3
Needle valves	1	2	3
Brake calipers	1	1	2
Yaw accumulator pressure sensor	1	2	3
Yaw oil level sensor	1	2	3
Yaw pressure sensor	1	2	3
Yaw oil overtemp.	1	1	2
Rotor brake acc. pres.	1	2	3
ECS fan	–	1	1
Generator accessory unit			
Gen. circuit breaker	Repair kit	Repair kit	Repair kit
Voltage regulator	1	1	2
Pwr. factor controller	Repair kit	1	3
Field current relay	1	1	2
Gen. protection relay	1	1	3
Potential transformer	1	1	2
Current transformer	1	1	2
Voltage transducer	1	1	2
Current transducer	1	1	2
Real pwr. transducer	1	1	2
Reactive pwr. transducer	1	1	2
Accessory pwr. transformer	1	1	2
Lamps, maintenance	2	3	12
Aircraft strobe lamps	5	10	50
Tower subassembly			
Nacelle acess device	Repair kit	Repair kit	Repair kit
Ground intrusion sensor	–	1	3
Electrical equipment			
Bus tie breaker	Repair kit	1	2
Synchronizer	1	2	4
Differential protecting relay	1	1	3
Meters (8)	–	1 Each	2 Each
Potential transformer	1	1	2
Current transformer	1	1	2
Generator tot. run hrs.	–	1	2
BTB operating cycles	–	1	2
16KV to 480V transformer	1	1	2
480v to 115/280 transformer	1	1	2
Battery	–	1	3
Battery charger	–	1	2
Battery heater	–	1	2
Power output transformer	–	1	1
Manual fuse disconnect switch	–	1	2
Microprocessor	2 sets of cards	2 Sets of cards	3 Sets of cards, 1 chasis
Local CRT/Keyboard	1	1	2
Manual control	–	1	2
Utility substation			
Communication processor	–*	1	2
Printer – keyboard	–**	1	2
Display panel	–*	Switch kit	Switch kit

* Not used for single unit

**Identical to local CRT/keyboard

5.5 COST ASSESSMENT AND COST OF ELECTRICITY

This section presents the methods used to arrive at the turnkey costs, the major data used in deriving these costs, the development of O & M costs and the resulting cost of electricity. Figure 5-31 illustrates the approach used to prepare the 100th unit cost estimates. This approach was used throughout Task II and is applicable to all configurations studied. The hardware layouts and drawing parts lists, and vendor inputs were used as the basis for developing the manufacturing process work sheets, tooling concepts and production plans. Material estimates were initiated based on preliminary make/buy decisions.

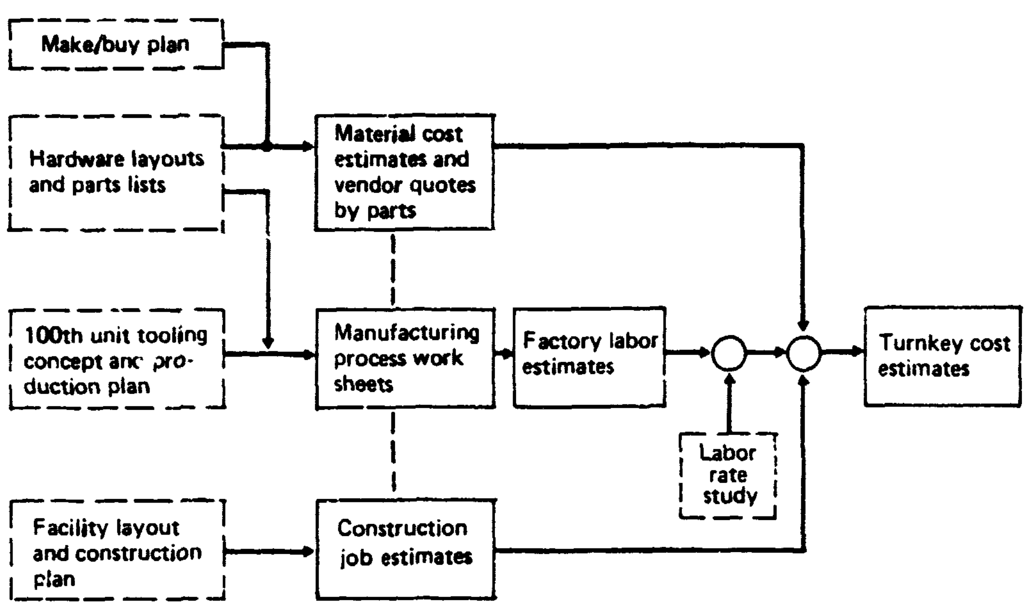


Figure 5-31. 100th Unit Cost Approach

The facility layout and construction plans (section 6.1.2 and 3.2.4) were developed in parallel with the manufacturing and procurement efforts and formed the basis of the construction job estimates. Factory man hour labor estimates for in-house manufactured items were prepared from the manufacturing process work sheets and these were converted to dollars based on the labor rate described in the next paragraph. Finally, all three costing activities were brought together to form the basis of the turnkey costs summarized in Table 5-17 and discussed in section 5.5.1.

The 100th unit cost estimates shown in Table 5-17 are based on the following groundrules:

(1) All costs are in 1977 dollars (no escalation or interest during construction)

(2) Costs include full burden.

(3) The manufacturing costs are based on a 20 unit per month production rate, a 30 year facility life and a 20 year life for tooling and equipment. The plant is dedicated to MOD-2 production and is located in the Southeast.

(4) Transportation costs are based on conventional rail and truck transport over 1000 miles distance.

(5) Construction costs are based on a 25 unit farm.

Table 5-17. MOD-2 Turnkey Cost Summary (100th Unit)

DASH NO.	QUANTITY	DESCRIPTION	100 Unit Cost - $ MATERIAL	MANUFACTURING
1.0	1	Site Preparation	110,428	51,100
2.0	1	Transportation	25,320	4,000
3.0	1	Erection & Checkout	58,900	78,000
4.0	1	Rotor Assembly	150,430	178,463
5.0	1	Drive Train	378,527	--
6.0	1	Nacelle Subassembly	153,086	31,000
7.0	1	Tower Subassembly	270,800	--
8.0	1	Spares & Maintenance Equipment	35,000	--
8a.0	1	Non-recurring	35,000	--
9.0	1	Total Initial Cost	1,217,491	342,563
Fee	1	10%	121,749	34,256
		TOTAL	1,339,240	376,819

5.5.1 Turnkey Accounts

Exhibit A of the contract specifies that the cost data shall be based on the following elements or "turnkey" items of the wind turbine system:

(1) Site preparation (incl. tower foundation)
(2) Transportation
(3) Erection (incl. functional test and checkout
(4) Rotor subassembly (incl. pitch controls)
(5) Drive train subassembly
(6) Nacelle subassembly (inc. yaw system & electronic controls)
(7) Tower subassembly (incl. electrical power system)
(8) Initial spares & maintenance equipment

Cost estimates were based on design layouts and collected for each item on the level of the current parts lists as shown in Table 5-18 for the pitch control system. The total cost of the pitch control system is one of the cost items in Table 5-19 which shows the cost summary for the entire rotor subsystem. Each cost element of the "turnkey" cost account (Table 5-17) was treated in the same way as this example.

Table 5-18. Blade Pitch Control Detailed Material Cost Report (Tier V)

Turnkey Account: 4.3.3 Drawing: K6267-SMD-129

Find Number	Quantity	Make/ Buy	Item description	100th Unit Cost
2	1	Buy	Motor - Electric	295
3	1	Buy	Flange adapter	(incl)
4	1	Buy	Pump - Hydraulic	2,378
6	3	Buy	Accumulator	701
8	1	Buy	Filter assembly	280
9	1	Buy	Heat exchanter	234
10	2	Buy	Filter assembly	114
11	1	Make	Manifold	9
12	1	Buy	Solenoid valve	45
13	2	Buy	Solenoid valve	115
14	1	Buy	Relief valve	20
15	3	Buy	Check valve	85
16	1	Buy	Check valve	44
17	4	Buy	Pressure switch	260
18	1	Buy	Temperature switch	20
19	1	Buy	Temperature switch	20
20	2	Buy	Differential pressure swit	423
22	1	Make	System fill fitting	40
23	1	Make	Pilot operated check valve	40
24	1	Buy	Accumulator	158
25	1	Buy	Cap-fill fitting	(incl)
26		Buy	Hydraulic tubing	116
27	1	Buy	Hydraulic fluid	133
1	1	Make	Hydraulic inst-low speed shaft	5,510

Table 5-19. Rotor Subsystem Cost Report (Tier III and IV)

Dash No.	Quantity	Account Number	Description	100th Unit Cost Material	100th Unit Cost Manufacturing
-164	2	4.1.1	Mid-section assembly	40,462	53,000
-166	2	4.1.2	Tip assembly	24,761	33,300
	2	4.1	Blade	65,223	86,300
-163	1	4.2	Center section	71,520	86,000
-172	2	4.3.1	Blade tip actuator	1,805	400
-168	2	4.3.2	Blade hydraulics	4,463	500
-129	1	4.3.3	Pitch control - low speed shaft	5,510	1,000
-45	1	4.3.4	Hydraulic reservoir	709	4,063
K-6266-JH -329	2	4.3.5	Electric cables	1,200	200
	1	4.3	Pitch control	13,687	6,163
	1	4.0	Rotor	150,430	178,463

5.5.2 Labor Rate Definition

All of the manufacturing labor estimates were prepared in terms of basic factory labor (BFL) hours. BFL hours are the direct hands-on manufacturing hours and must be factored to account for support functions, rework effort, etc. Boeing has conducted extensive studies of both labor rates and support function costs at Boeing and in the medium steel fabrication industry. The results are shown in Table 5-20. The computed wrap-around rate of $25.00 per hour, exclusive of plant, equipment, tooling and fee, is a conservative rate based on comparisons with on-going medium steel fabricators in the Southeast.

Table 5-20. Labor Rate Computation

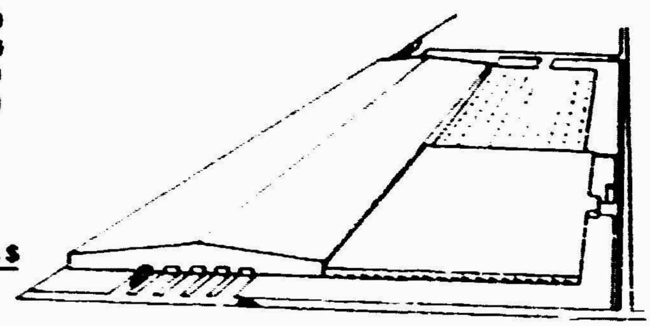
ITEM	COST/HR.
Basic Factory Labor	$ 7.50
Miscellaneous and Pickup	.375
Rework	.375
Manufacturing Development	.04
Manufacturing Engineering	.375
Distributed Direct	1.12
Quality Control	.60
Finance	.225
Overhead	2.75 *
Fringe Benefits	5.35
General and Administration	1.37
Taxes	.54
Subtotal	20.62
Contingency	4.38
TOTAL	$25.00

* Less plant and equipment

5.5.3 Manufacturing Costs

The manufacturing costs of the 100th production unit are based on an optimized production facility producing 20 wind turbines per month. The optimized production facility has a single, automated production line running three shifts in the fabrication area and two shifts in the major assembly positions with a total employment of approximately 1300 people. The space requirements and cost (in 1977 dollars) of this plant are shown in Figure 5-32. This plant will manufacture, assemble and checkout the complete rotor, drive train, and nacelle. The tower and other components are manufactured by specialized vendors. The plant will be located on a site with railroad and road accessibility in an area with adequate availability of skilled labor. The general location of the site is expected to be near the region of WTS deployment.

• Space requirements	Square feet
• Manufacturing & assembly floor space	546,000
• Stores, receiving, shipping	206,000
• Quality assurance labs	20,000
• Office space, support	63,000
• Parking	720,000



• Costs	Millions $
Tooling	28.2
Equipment (Machinery, etc.)	21.6
Factory	43.1
Total	92.9

Figure 5-32. Factory with Single Product on Line with a Rate of 20 WTS per Month

For economics analyses, the plant is depreciated over 30 years and the equipment and tooling depreciated over 20 years. The corresponding first year depreciation charge is $8.5 million (using the sum of the year's digits method of depreciation). At the 20 WTS per month production rate, this results in a $35,000 non-recurring charge per unit to be included in the WTS price.

The costs shown in Table 5-17 reflect a mature, automated production line in a dedicated plant, with initial problems resolved and learning improvements realized. Construction and transportation costs are based on job estimates. Hardware costs are based upon the material and labor requirements of each sub-assembly, and include purchased parts and subassemblies.

5.5.4 Operations & Maintenance Costs

The MOD-2 operations and maintenance concepts are based on remote, unattended operation of a 25 unit farm. They rely upon thorough reliability, maintainability and safety analyses conducted during the conceptual and preliminary design phases of the program as described in section 5.4. Studies were conducted that traded initial costs vs. annual maintenance costs; other logistics elements such as preventative maintenance and spares levels were optimized to achieve the lowest cost of electricity. The MOD-2 operations and maintenance costs are shown in Figure 5-33 for a single WTS. These costs, as well as the maintenance equipment and initial spares costs which appear in the turnkey cost summary (Table 5-17) are based upon the analyses presented in section 5.4.

ITEM		ANNUAL HOURS PER 25 UNIT FARM	
• Unscheduled Maintenance		6,800	(from R & M analysis)
• Scheduled Maintenance		1,800	(72 hours per WTS per yr.)
• Administrative Tasks		960	(2 hours per unscheduled event)
	TOTAL	9,560	hours

9,560 hours x $20 per hour ÷ 25 WTS = $7,650 per WTS per year

	O & M Cost per WTS	
• Labor	$8,000	(2-2 Person shifts/day 6 days/week plus contingency for Sundays ÷ 25 WTS)
• Parts & Outside services	6,950	(Averages $340 per unscheduled event and $450 annual equipment rental charges)
TOTAL	$14,950 ~ $15,000	

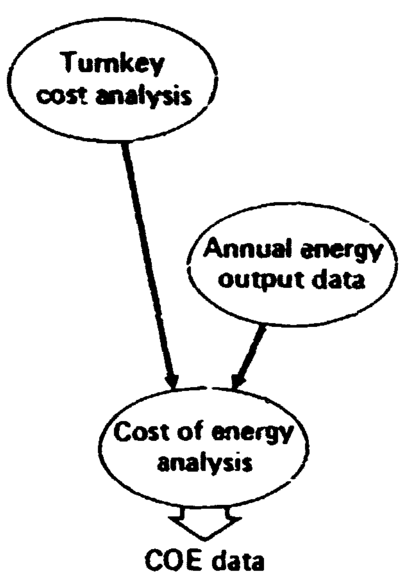
Figure 5-33. O & M Cost Estimates

5.5.5 Cost of Electricity Formulation

The energy pricing method used to project the MOD-2 WTS energy costs is the levelized fixed charge rate approach. It derives a levelized energy price necessary to recover the costs to a utility for purchasing, installing, owning, operating, and maintaining a MOD-2 WTS.

The cost of electriciy (COE) is derived from the turnkey and O & M costs, energy output and availability. The specific formulation used for the cost of electricity calculation is shown in Figure 5-34. A summary description of each term in the equation follows:

- FCR = levelized fixed charge rate which includes return on capital, income tax, property, tax, and insurance. FCR is sensitive to cost of capital, capitalization method, income tax rate, and system lifetime.
- IC = initial turnkey cost of the energy system which includes complete cost exposure to the utility for purchasing, installing and setting up logistics for the energy production system.
- AOM = annual operation and maintenance (O & M) cost which includes operating budgets and maintenance budgets.
- AEP = anticipated annual energy production of the energy system. AEP takes into account energy production losses attributed to the unavailability of the energy system equipment and the unavailability of the energy source (i.e. wind).



- Cost of electricity

$$COE = \frac{IC \times FCR + AOM}{AEP}$$

- FCR = annualized fixed charge rate = 18% per year
- IC = total WTS cost=$1,720,000
- AOM = annual operations & maintenance cost = $15,000
- AEP = annual energy production = 9.75×10^6 kWh
- COE = 3.3¢/kWh

Figure 5-34. COE Determination

5.6 WEIGHTS ANALYSIS

The requirements for accurate and comprehensive weight and mass properties of the WTS was decreed by the need:

1) to define material types and gross quantities so that material costs could be arrived at and controlled early in the design cycle of the WTS.

2) for mass properties to be available for loads and structural dynamics analysis.

3) to exercise weight/cost control by means of weight targets being allocated to individual parts of the WTS.

To achieve this end, an existing aerospace mass properties computer program was used. The inputs required for the program were obtained by an in-depth analysis of all the individual design lay-out drawings as they became available. The weights of manufactured detail parts were calculated, and the weights of purchased items were obtained either from vendors or by using known weights of existing parts that were similar to the specially designed items required for the WTS. Contingencies of up to 20% were included in critical development item hardware to cover the possibility of unknown design requirements arising. In addition, it was recognized that material tolerances alone could have a significant impact on the weight of the WTS and an allowance was added to all purchased materials to account for tolerance overages. A further allowance was included to cover miscellaneous small items that were not detailed on the design lay-out drawings. As the lay-outs became more explicit,this allowance was reduced. The weights so obtained, each with its material coding letters, were then input to the program.

The inputs to the program included the center of gravity of each part about the WTS 'x', 'y', and 'z' reference axes, the calculated radii of gyration about the parts axes, it's fore and aft dimensional limits, it's designated group and section codes (for example see Figure 5-35), it's descriptive title, the material it is to be made from, and the fabrication method. A total of eighteen different inputs were made for each of the more than 900 parts at the current level of analysis. These inputs were continually revised throughout the conceptual and preliminary design stages to incorporate design improvements and updated structural analysis.

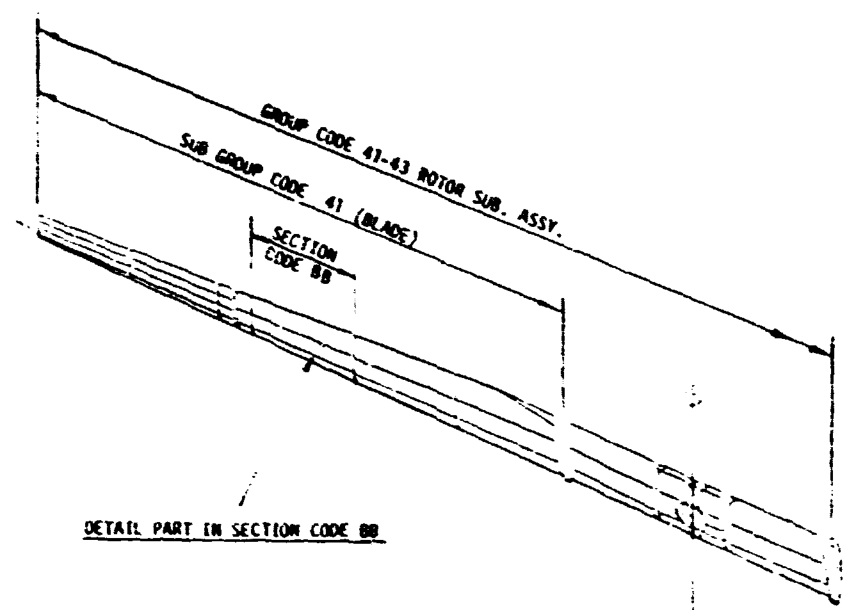


Figure 5.35. Mass Properties Program Group Coding

The computer program output provided reports selected from a listing of thirty-seven that are available. The reports used most frequently for MOD-2 were as follows:

1) Mass Distribution

   The program collects all parts with the same sub group code and analyzes each part in turn. The c.g. of the part is positioned between the forward and aft limits and the weight divided equally either side of the c.g. This weight is then uniformly distributed between the limit and the c.g. in one inch increments. This process is repeated for each detail part and the results for all parts are totaled, to give a complete mass distribution in one inch increments. Each increment is printed out with its weight, and cumulative shear and moment.

2) Section Summary.

Collects weights and c.g. of all parts in a section and summarizes.

3) Group/Section Summary.

Collects weights of all sections within a sub group and totals weight.

4) Inertia Summary.

Collects weights and c.g's of all parts in a section and calculates a composite c.g., inertia, and product of inertia. Summarizes results by sub group.

5) Material Summary.

Sorts parts by type of material and fabrication method and summarizes, to give total steel plate, forgings, paint (etc.) used.

6) Tier V Listing.

Lists each individual part in a sub group with its weight, c.g's, and roll, pitch, and yaw moments of inertia, and summarizes each sub group.

7) Tier IV Listing.

Lists total weight and c.g for all items in a sub group and summarizes by group.

8) Tier III Listing.

Lists total weight and c.g., target weight, and over/under target weight for all groups, and summarizes total MOD-2 values.

The estimated weight of the MOD-2 WTS is shown in Table 5-21. The weight statement is derived from a build-up of components, design details and sub-assemblies. It includes all WT hardware, electrical cabling and controls, instrumentation, fluids, heating/cooling requirements, maintenance and safety provisions, etc., from the base of the tower through the rotor. More than nine hundred unique items make up the detailed Tier V weight statements for the subassembly group codes in Table 5-21.

Weight control has been exercised throughout the conceptual and preliminary design of the WT. Minimum weight, although desirable, in many cases has not been the deciding factor where a decrease in weight would result in an increase in cost. As an aid in monitoring and control, initial target weights were established at the sub assembly level. The Tier III weight statement (Table 5-21) shows the relationship to target goals. The current weight is approximately 45,000 lbs. lower than the target weight allocations. The unit weight of 232 lbs/kW reflects a system designed for minimum cost of energy.

The weight and mass properties task has therefore been a significant input to, and a monitor of, the cost optimization process.

Table 5-21. MOD-2-107 Wind Turbine System Mass Properties Status Summary

GROUP	ELEMENT	CURRENT WT.	OVER/UNDER. WEIGHT	TARGET WEIGHT
41	BLADE	100336.00	10536.00	89800.00
42	HUB	67722.00	12222.00	55500.00
43	PITCH CONTROL	1509.00	-191.00	1700.00
41-43	ROTOR SUBASSEMBLY	169567.00	22567.00	147000.00
51	LO SPEED SHAFT+BRNGS	22865.00	-4335.00	27200.00
52	QUILL SHAFT+COUPLING	9483.00	-2217.00	11700.00
53	GEARBOX	39000.00	-66000.00	105000.00
54	HISPEED SHAFT+COUPLG	600.00	-600.00	1200.00
55	ROTOR BRAKE SYSTEM	280.00	-320.00	600.00
56	LUBRICATION SYSTEM	6664.00	4664.00	2000.00
57	GENERATOR	17000.00	2000.00	15000.00
58	D-TRAIN GROWTH ALLOW	8000.00	8000.00	0.00
51-58	DRIVE TRAIN	103892.00	-58808.00	162700.00
61	NACELLE STRUCTURE	33680.00	3180.00	30500.00
63	YAW DRIVE	17742.00	542.00	17200.00
64	ROTOR SUPPORT STRUCT	7152.00	1452.00	5700.00
65	ENVIRON.CONTRL+FIRE	870.00	-230.00	1100.00
66	CABLING+ELEC.FACIL.	645.00	245.00	400.00
67	INSTRUM.+CONTROLS	690.00	-10.00	700.00
68	GEN. ACCY.UNIT	2500.00	-1500.00	4000.00
61-68	NACELLE	63279.00	3679.00	59600.00
71	TOWER	246536.00	-6464.00	253000.00
72	CABLE INSTALLATION	4130.00	3230.00	900.00
73	CABLE TRANSITION	500.00	-1000.00	1500.00
74	LIGHTNING PROTECTION	300.00	0.0	300.00
71-74	TOWER SUBASSEMBLY	251466.00	-4234.00	255700.00
41-74	TOTAL INCL. GROWTH	588204.00	-36796.00	625000.00
75	D-TRAIN GROWTH ALLOW	-8000.00	-8000.00	0.00
41-75	TOTAL EXCL.GROWTH	580204.00	-44796.00	625000.00

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5.7 MANUFACTURING DEVELOPMENT

The Manufacturing Development activities were oriented toward supporting the Engineering preliminary design effort while the manufacturing shops acquired some familiarity with the unique fabrication processes of the WTS. The development activities were primarily associated with fabrication of the rotor blade and were divided between welding and forming processes.

5.7.1 Weld Development

Engineering selected three materials as possible candidates for the rotor blade. Manufacturing prepared welding specimens from each of these materials, A-588, A-572 GR 42 and A-633 Gr A. Various thicknesses and joint designs were welded and evaluated. From a manufacturing standpoint, the three materials were equally weldable and sound weldments were produced.

Twelve weld joint designs were offered by Engineering as possible configurations for the rotor design. Each of the joint designs was welded using the three candidate materials with various weld wires, back-up materials, edge preparations and welding positions. The degree of ease or difficulty in consistently producing and inspecting each of the joint designs was passed on to Engineering. The more difficult designs were eliminated or revised for the rotor design. Development work is continuing as Engineering designs for material gages are finalized and pending the final decision on the need for stress relieving the weldments.

5.7.2 Forming Development

The Forming Development Program was initiated to determine whether the rotor blade skin panels could be formed to the contour tolerances required by the design. Other objectives were to determine the maximum length of panels that could be formed and identify tooling requirements for an acceptable product.

Chip forming, by a progressive series of light hits on a press brake, appeared to be the most economical method of forming limited quantities of skin panels. Consequently, a survey was made of fabricators having press brakes up to sixty feet in length in a tandem position.

Because of the blade tilt limits of the presses it was determined that a skin panel having the varying contour of the rotor blade design would have to be limited to about twenty five feet in length. Three vendors were selected to form the development program skin panels as discussed in section 5.7.3

5.7.3 Fabrication of Buckling Test Specimens

The final phase of the manufacturing development program was the fabrication of a full scale 35 foot long section of the blade. The section of the blade was selected from station 360 to station 780. Station 360 represented the field joint and this section of the blade presented some of the most difficult skin panel forming problems. The skins were 3/8 and 1/2 inch gauges of A-588 material. The trailing edge panel (22 feet in length) transitioned from a constant radius at station 360 to a sharp trailing edge of the airfoil at the outboard end. This represented the most severe forming condition of the entire rotor.

Flat pattern layouts and templates made to the inside mold line of the parts were furnished to the vendors along with the material. The vendors were able to layout the material to net length, cut the weld edge preparation on the ends and form the contour to within 1/16 inch of the contour templates. After the excess width required for forming was cut off, the panels were ready for the weld operation.

Subsequent weld fabrication of the section was accomplished with a minimum of difficulty using soft tooling techniques. The fabrication demonstrated the manufacturing capability for fitting up the formed components and producing a satisfactory weldment. The experience also provided valuable information for designing weld fixtures for the rotor. The completed 35 foot section was then successfully subjected to the buckling test as discussed in section 6.3.1. The section has subsequently been stress relieved. Inspection for contour changes has found no discernable distortion as a result of the heat treatment.

The hub section trailing edge panel is shown as received from the vendor in Figure 5-36.

5.8 PRODUCIBILITY STUDIES

An integral part of the MOD-2 development has been a joint Material/Manufacturing/Quality Control/Engineering effort to reduce the cost of each system element to a minimum. Essentially, every component has been subjected to layout review, trade studies, and special study meetings where appropriate, to achieve cost reduction consistent with the desired performance and life goals. Producibility improvements have been incorporated in each major sub system area of the MOD-2 WTS, in the manufacturing plan for large scale production, and in the erection plan.

Some specific examples of successful producibility study application are:

(1) A substantial reduction in the use of expensive forgings in the rotor and low speed shaft.
(2) Selection of low cost, low risk materials for the rotor that are readily available and that weld particularly well.
(3) Selection of an all welded steel trailing edge that not only reduced cost and weight but that also eliminated the risk of service bond delaminations.
(4) Choice of the very economical corrugated steel as the nacelle external cover.
(5) Change to a field welded tower joint configuration, replacing the more expensive bolted joints previously proposed.
(6) Reconfiguration of the foundation to permit replacing a substantial part of the required concrete with much more economical earth fill.
(7) Selection of an epicyclic gear box design resulting in substantial cost and weight reductions.

A rather special example of the producibility effort was demonstrated during a month long cost reduction study activity, resulting in a cost of electricity reduction of approximately 17%.

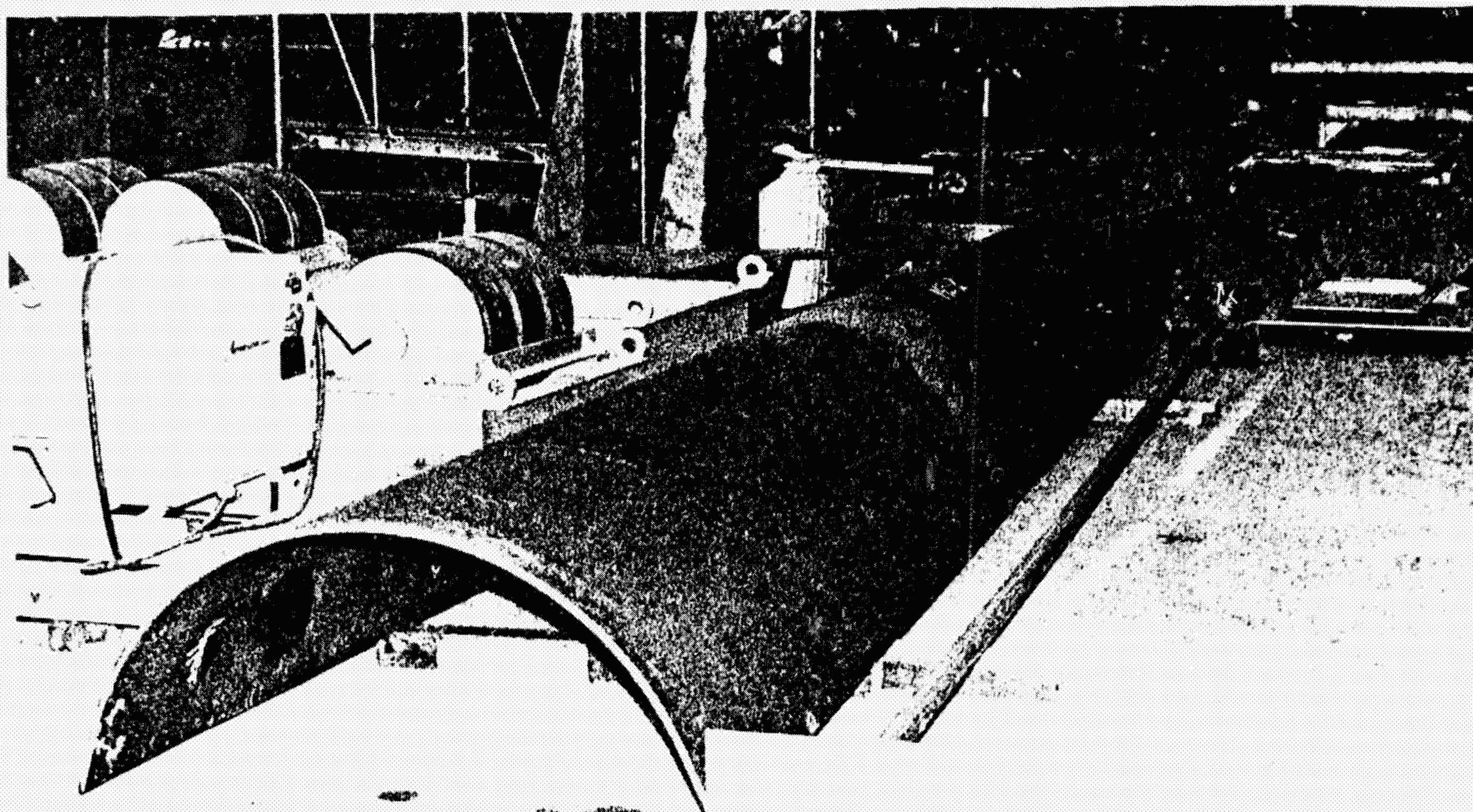

Figure 5-36. Hub Section Trailing Edge Panel

6.0 MANUFACTURING, ASSEMBLY, AND TEST

As the conceptual and preliminary design phases progressed, a number of activities oriented towards updating the initial concepts of manufacturing, assembly, inspection, test, transportation and erection of the MOD-2 WTS were pursued. Inputs were provided to the designers with respect to the manufacture and test of the various concepts under consideration. Refined cost and manpower estimates were developed for cost of electricity studies. Test programs were conducted to assist in confirming design decisions or providing design data.

When work progressed into the preliminary design phase, plans were developed as preliminary documents for the Manufacture, Test, and Quality Assurance activities to be implemented in the post-design phases of the program. Estimates were provided to update the cost of electricity of the production units. A development test program was implemented at the start of the preliminary design phase and included both test and manufacturing development activities. One result of these activities was to provide additional confidence in the design approach and analysis used in arriving at the preliminary design configuration. A second important result was the development of the procedures and processes to be used in fabricating and assembling the MOD-2 blade and the increase in confidence in our ability to produce the blade as designed.

This section includes a description of the manufacturing and construction procedures, and a summary of test activities.

6.1 MANUFACTURING AND QUALITY ASSURANCE

Manufacturing includes the fabrication, assembly and preparation for shipment of selected items of the MOD-2 WTS. This includes providing the manpower, tooling, equipment and other resources to accomplish this manufacturing task.

Quality assurance ensures the manufacturing activities result in a quality product. This includes surveillance as well as physical inspection of the hardware during manufacturing and test operations.

This section summarizes the manufacturing and quality assurance procedures for the MOD-2 WTS. The quality assurance procedures presented herein are derived from the BEC developed, NASA approved, Product Assurance Plan. Manufacturing procedures are based on preliminary plans prepared for the prototype unit fabrication. Similar plans and associated manufacturing scenarios which have been developed for moderate-to-high production rates are used in the development of cost of electricity estimates as described in section 5.5.

The moderate-to-high production rate scenarios are envisioned to be consistant with proceding from the first prototype units produced with modest tooling to full scale production of thousands of units produced with highly automated hard tooling. Manufacturing methods and sequences, and quality assurance procedures are independent of the production rate. Tooling and productivity, or manufacturing manhours, are strongly affected by the production rate. In general, facility requirements increase and the tooling becomes increasingly automated and complex as the production rate is increased, although the basic functions performed by the tools are the same as those described here for the prototype unit production. Manufacturing manhours, on the other hand, decrease dramatically with increasing production rate, offsetting the increasing tool cost.

6.1.1 Materials and Components

The manufactured components of the MOD-2 WTS are primarily made from structural quality steel as described in section 5.1.6.

Components for the system will be purchased to engineering specifications or vendor part numbers. Some components will be shipped directly to the installation site. Those components requiring installation will be processed through Boeing receiving inspection functions prior to being stored.

Components will be integrated into systems and installed in assembled components. When required, functional tests will be performed on subsystems prior to the integration in the system. Those components which require disassembly after testing to facilitate shipping, will be match-marked prior to disassembly to assure proper assembly at the installation site.

6.1.2 Manufacturing Methods

All manufacturing methods employed are well within the current state of the art and are standard industry practices. The rotor skins are formed using the "chip" technique in a press brake. The thickness and large dimensions of the rotor skins require facilities larger than those available at the Boeing Company. The skins will be subcontracted to vendors having larger forming equipment. The machining of ribs, fittings, and bulkheads is accomplished on large conventional mills or numerical controlled equipment which is available at Boeing.

The principle method of joining components of the primary structure is by welding. Tungsten inert gas or metal inert gas welding techniques will be used. Welders will be qualified for each joint configuration and material thickness range. In some cases, stick electrodes will be used to fabricate components from structural beams, mechanical tubing and plate.

Tooling will be used to locate, position, or restrain components for welding and interface drilling. Welding will be sequenced to minimize distortion and make-up joints will be used to compensate for weld shrinkage.

The tooling policy for the prototype units of the MOD-2 WTS is to keep tooling to a minimum. Adequate tooling will be used for critical processes and interfaces and when the cost of the tooling can be justified by reduced assembly process costs or is essential to the program schedule.

The major items of tooling are expected to be the weld fixtures for the rotor components. Typical tooling concept for the weld fixture for the rotor tip is shown in Figure 6-1. A series of floor mounted tooling headers will be erected to control the lower surface contour and retain the axial twist of the assembly. Removable upper members of the tool will allow sequential loading and welding of skin, spars and bulkheads.

Similar concepts are envisioned for the rotor mid-blade section and for the rotor center section. Other major tooling items will consist of specially designed slings for handling and installing heavy items of equipment. Lesser items of tooling that will be required (i.e., contour templates, drill templates, etc.) are non-designed tools and are fabricated to accepted standard tooling practices.

6.1.3 Manufacturing Sequence

The following paragraphs and associated illustrations describe the sequence of manufacturing operations for fabrication and assembly of the MOD-2 WTS.

6.1.3.1 Rotor Fabrication

The rotor has five major assemblies. The assemblies, consisting of two tips, two mid-section blades and a hub center section are fabricated concurrently. After the individual assemblies are completed, they are joined to make a complete rotor for installation on the low speed shaft previously installed in the nacelle.

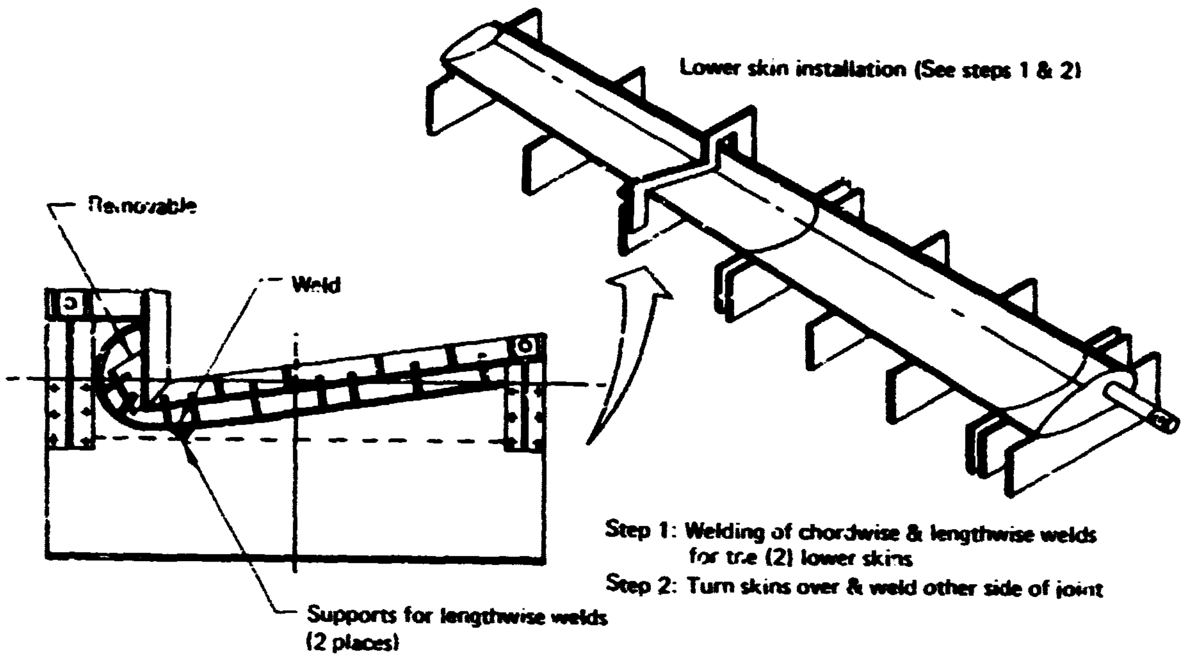

Figure 6-1. Outboard Section (Sta 1260 to Sta 1800)

The rotor tip is fabricated in two steps. First, the spindle box containing the pitch spindle and structural members between stations 1260 and 1320 is welded into a unit. This assembly is a convenient size for final machining operations and provides a dimensionally stable assembly that ensures alignment of the shaft with the remaining tip structure. The second step integrates the spindle box into the total tip weld assembly accomplished in the tip welding fixture.

After completion of all weld operations, the ballast weights are installed, the assembly is cleaned and painted,and the spindle sleeve and associated bearings seals and retainers are installed. The assembly is weighed and balanced by a final adjustment of ballast weights and then is joined with the blade and hub center section to make a complete rotor.

The manufacturing sequence for the mid blade is similar to the rotor tip. A subassembly of the spindle box is welded and machined before integration into a complete blade. Drilling of the Sta 360 field splice joint holes is done after all weld operations are completed.

The hub center section is fabricated in several stages. The bearing support assembly and the bearing and teeter brake support assembly are welded, surfaces are machined,and mounting hole locations drilled. The "D" shaped sections in the center of the hub are welded as sub-assemblies. The "D" sections are mated with the bearing support assemblies and match drilled. These holes are the index points for location in the center section weld fixture. The outboard portions are then built up to mate with the center portion. After weld completion, the blade attach hole pattern is drilled, the assembly is cleaned and painted and is joined with the blades and tips to complete the rotor assembly.

6.1.3.2 Final Assembly

The MOD-2 WTS will be assembled as shown schematically in Figure 6-2 preparatory to in-plant integration testing. The following text describes, in general terms, the manufacturing activities associated with this task. Although listed sequentially, some of the operations may be performed concurrently. Methods studies of man loading and accessibility will determine the optimum sequencing that is compatible with cost and schedule targets.

The nacelle is procured by BEC from a subcontractor. Electrical conduit and wiring for the nacelle interior lighting, convenience outlets for accessory power and the external lighting will be installed as soon as the nacelle is received and while accessibility is unrestricted.

Cable runs from the yaw slip ring location to equipment locations will be routed. Circuit breaker panels will be installed. Hookup of the cabling to the equipment must necessarily be deferred until the equipment is installed.

The nacelle will be temporarily shored up at each end for installation of the yaw drive system. The yaw bearing, manufactured by a subcontractor, is jacked up into position under the nacelle and the attach bolts installed. The nacelle will then be mounted on the simulated tower section in the final assembly and checkout area. The yaw brake disc will have been mounted on the tower. The brake calipers will be installed. The drive motor will be mounted on its support.

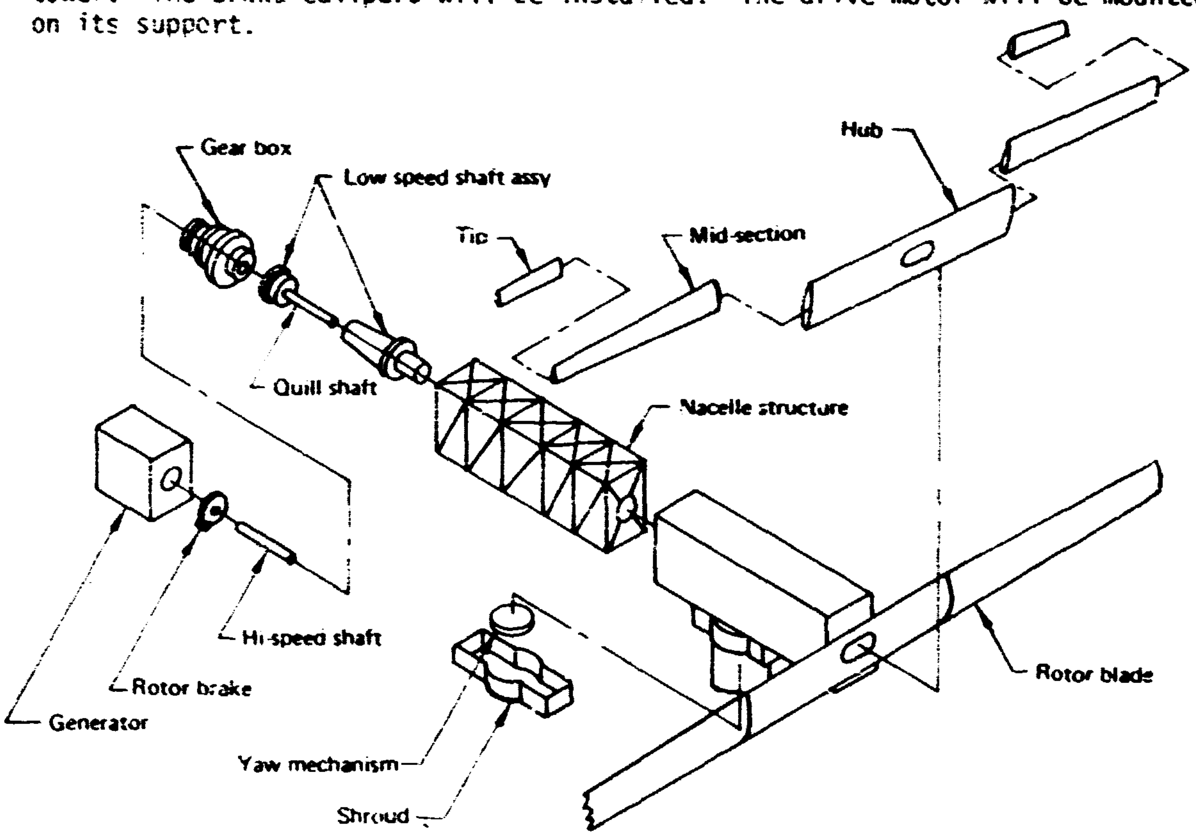

Figure 6-2. MOD-2 Assy Sequence Overlay

The hydraulic module will be installed in the nacelle and plumbing work completed. The yaw system shroud will be bolted in place. A checkout of the yaw drive system may be made at this time.

The low speed shaft, as supplied by the subcontractor, will be mounted in a work stand for the installation of hydraulic equipment and plumbing associated with the pitch control system. The drive train slip ring will be installed and electrical wiring to the rotor will be routed. The shaft bearings will be installed and the quill shaft inserted in the low speed shaft. The assembled low speed shaft will then be mounted on the bearing supports of the nacelle and aligned. The gear box will be mounted and shimmed to align with the quill shaft. Finally, the generator and high speed shaft will be installed and adjusted for alignment. Associated control equipment and cooling air ducting will be installed and all electrical and hydraulic hookups will be completed.

The rotor center section, blades and tips have previously completed the structural assembly operations and have been painted and individually weighed and balanced. The teeter bearings will be installed in the center section. The center section will be installed on a supporting fixture. Hydraulic lines for the pitch control system will be installed and routed to the outboard extremities of the center section. Electrical cabling and wiring for the instrumentation will be installed at the same time.

The mid blades will be joined to the hub section by the installation of bolts at the field splice joint. Plumbing and electrical wiring in the hub section will be hooked up to the previously installed lines in the blades.

The tips will be joined to the mid blades by insertion of the spindle sleeve in the blade socket and the fasteners installed. The pitch actuators will be hooked up and final electrical and hydraulic connections will be made.

Electrical continuity checks will be made. The hydraulic lines will be flushed and filled. After checks are made for leaks and final inspection has been completed, the functional operation of the pitch control system will be verified. Upon completion of the testing, the completely assembled rotor will be transferred from the test fixture to the low speed shaft installed in the nacelle. A final hookup of lines from the low speed shaft to the rotor completes the manufacturing activities, except for the disassembly and preparation for shipment, when the inplant tests are finished.

6.1.4 Quality Assurance

This section describes objectives, philosophy and plans concerning quality assurance inspection and testing techniques and criteria for the MOD-2 WTS.

A Product Assurance Program Plan, D276 10500-1, was developed in accordance with the NASA-LRC Statement of Work and Exhibit "C" of contract DEN 3-2. This customer approved plan defines the system to be used to ensure compliance with the safety, reliability and quality assurance requirements of the contract.

6.1.4.1 Inspection of Fabrication, Assembly, and Installation Operations

All manufacturing and inspection operations for the MOD-2 WTS project are being preplanned. Evidence of work completion will be traceable to the individual performing the operation through the use of personal stamps issued to the individual employee performing the operation.

The operation will be a "controlled" manufacturing process wherein requirements (drawings, specification, documents), processes, methods, tools and equipments are specifically defined in a sequential manner to facilitate an orderly flow. Mandatory inprocess inspections and non-destructive examinations will be inserted at appropriate points to ensure an adequate quality level.

Specific inspection controls to be used on the MOD-2 WTS project include the following:

(1) Use of raw materials and standards previously accepted during receiving inspection operations.

(2) Controlled tooling (templates, drill jigs, etc.) inspected and certified for use.

(3) Inprocess surveillance of manufacturing operations.

(4) Dimensional verification using calibrated devices such as micrometers, scales, calipers, and gauges.

(5) Nondestructive testing of welding using ultrasonics, radiography, dye penetrants, etc.

All planning for the MOD-2 WTS project is reviewed and approved prior to release. Inspection operations are added during the review cycle. Critical hardware characteristics are noted on the planning and will be 100% inspected.

Quality assurance ensures that the MOD-2 WTS "as designed" configuration is accurately portrayed by the planning. Production planning is released only after the "as planned" configuration equals the "as designed" configuration. After manufacturing completion of the product, the "as built" configuration is compared to the "as designed" configuration. Differences between the "as built" and the "as designed" configurations are immediately resolved.

Alignment requirements of the MOD-2 WTS structure buildup will be controlled by tooling. All inspections will be to engineering tolerances and will be performed using state-of-the-art techniques. All required inspection equipment is currently in the Boeing inventory or in the case of purchased hardware, is a standard common to industry.

6.1.4.2 Special Product Analysis

Material for the rotor will be purchased to requirements that preclude the presence of detrimental flaws. Acceptance will be predicated on results of chemical analysis, tensile properties tests, and charpy v-notch impact tests. No ultrasonic inspection testing is planned. The orientation of defects in rolled plate are laminar. Laminar defects are not detrimental unless they are at an edge which would affect the integrity of the weld. Therefore, no shear wave ultrasonic inspection is planned. Penetrant, rather than longitudinal UT, will be utilized to inspect all weld prep edges before welding is started.

Welding of the rotor will be per AWS structural welding Code D1.1-75, including the use of qualified weld procedure specifications. Qualified and certified welders and inspectors will be utilized during fabrication of the rotor.

Proposed weld joints have already been reviewed for inspectability. Non-destructive test methods and acceptance criteria have been evaluated which indicate that state-of-the-art NDT are available and usable to detect allowable flaw sizes.

6.2 TRANSPORTATION AND INSTALLATION

With the exception of the tower and certain electrical equipment, the MOD-2 WTS will be pre-assembled at the Boeing Company in Seattle. The rotor and drive train will be assembled with the nacelle and alignment checks completed prior to partial disassembly and shipment. The final rotor drive train alignment will take place at the WTS site. All components have been sized and weight limited so that they can be transported via conventional rail and/or highway modes.

Site preparation activities will be initiated while the WTS is being manufactured. The yard work, WTS foundation, and utility interface equipment installation will all be completed by the time the WTS arrives at the site. The tower assembly erection, and the nacelle and rotor assembly and mounting will be accomplished by using a 240 ft gin-pole. After drive train alignment, functional check-out, and utility personnel training, the WTS will be turned over to the user utility.

Preliminary transportation and site activity plans have been prepared to support the development of detailed schedules and to provide a basis for cost estimating. Prototype system schedules and costs are based on the installation of a single WTS unit at a site approximately 1000 miles from Seattle. Production planning has been based on the installation of a 25 WTS unit farm at a similar site. The results of these planning activities are presented in the following paragraphs. System checkout and acceptance test plans are presented in section 6.3.

These plans and the cost estimates developed from them are sensitive to the characteristics of the site selected for installation of the MOD-2 WTS. All transportation and site activity plans and cost estimates will be modified accordingly, following site selection. The specific site characteristics adopted as groundrules for the preliminary planning and cost estimating are as follows:

(1) The site is fully accessible by conventional over-the-road transportation.
(2) The site is located on generally flat terrain with a substrate presenting no unusual or adverse fractures.
(3) The site is assumed to be 1000 miles from the manufacturing plants.
(4) All construction work is assumed to be done in ideal weather conditions.
(5) All permits and licenses are furnished by the user utility.
(6) All utilities, water, power and access roads are provided within 400 feet of the tower by the user utility.
(7) Craft labor required to support the construction phase is available within commuting distance.
(8) Concrete for the foundation was priced at $40 per cubic yard.

6.2.1 Transportation

Until specific WTS sites are selected, transportation related activities are focused on the following elements: (1) preparation of design constraints dictated by transportation size and weight limits, (2) analyses of the type and number of specialized containers and handling fixtures needed to move components from the factory to the WTS site, and (3) preparation of the transportation cost estimates for inclusion in the MOD-2 turnkey costs. A formal documented transportation plan will be prepared and submitted at the detailed design review.

The design constraints were developed based upon railroad and trucking information with the objective of being able to ship WTS components to at least 90% of U.S. destinations. These constraints were incorporated into the MOD-2 design requirements as follows:

Transportability - All system elements with exception of the tower foundation shall be designed, fabricated and assembled so as to allow their transport by commercial carriers, utilizing standard commercial shipping and handling practices.

Weight and Dimensional Constraints - The WTS components shall be designed to meet the following weight and dimensional constraints if possible. Exceptions will require evaluation to determine if accomodations can be arranged.

TRANSPORTATION	MAX. WEIGHT	MAX. LENGTH	MAX. WIDTH	MAX. HEIGHT
Rail	300,000 lbs.	144'	10'8" *	12'
Truck	100,000 lbs.	150'	14'	10' 12' with lowboy

* Can be increased to 12 feet for cylindrical items that are 55 feet or less in length.

Qualitative Handling and Transportability Requirements - Handling, tiedown and slinging points shall be incorporated in the design of equipment to facilitate transportability, warehousing, and handling requirements. Full design consideration also shall be given to the safety aspects of equipment, its sectionalization and disassembly capability for transport purposes, and on-site reassembly.

A preliminary analyses of specialized packaging and handling requirements has been performed in order to compute realistic cost estimates and to identify any long lead items. No long lead items were identified. Shipping fixtures for the production program will be reused. Specialized fixtures will be required for the blades, gearbox, tower, low speed shaft, quill shaft and couplings, yaw bearing, generator and nacelle. The rotor will be shipped in five pieces (Figure 6-3). The preassembled nacelle structure with its mechanical/electrical installations will be transported to the site by truck. Due to weight limitations, the gear box and generator will be shipped separately. The tower will be shipped in sections and the electrical switchgear and transformers will be shipped individually. Once the items are delivered to the site, covered storage for the electronics and critical machined surfaces will be provided by tarpaulins and by temporary storage within the tower.

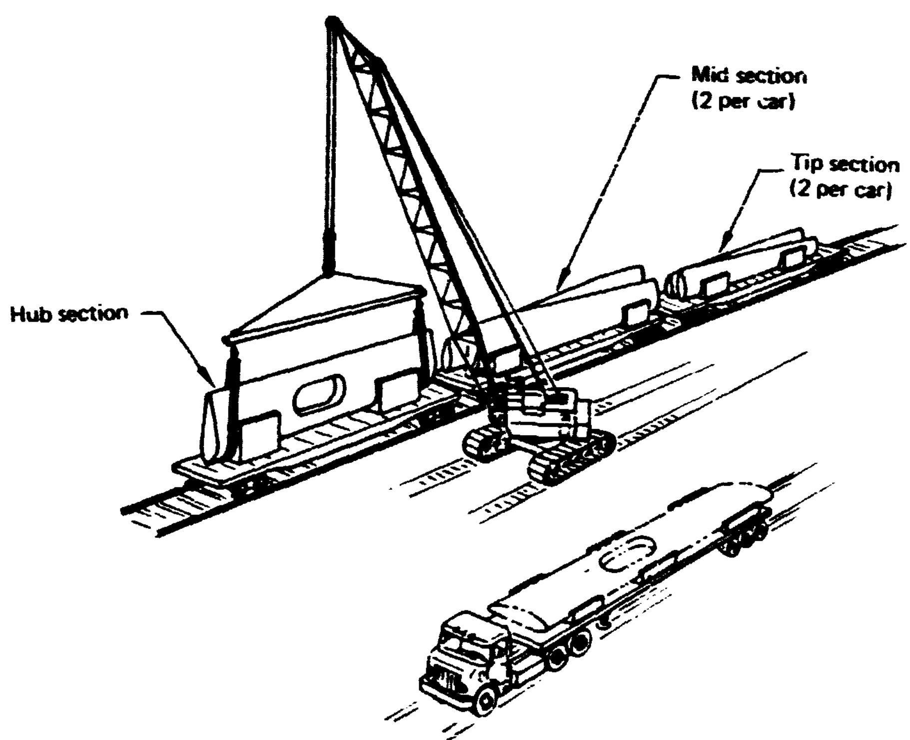

Figure 6-3. Standard Truck and Railroad Cars

6.2.2 Site Preparation and Installation

The detailed design of the WTS foundation and utility interfaces will be initiated immediately after site selection. Such factors as soil characteristics and seismic zone have a significant impact on the foundation design. Boeing will work directly with the user utility to coordinate the WTS-to-utility interface. This activity will occur in parallel with a detailed site survey where such things as topography, soils investigation samples, roads, transportation routes, requirements for temporary facilities, types of equipment required for site preparation, and available on-site utilities are defined in sufficient detail to support the site design. The government and/or user utility will furnish electrical distribution lines, and communications lines and access roads to the site.

The activities associated with site preparation include:

1) clearing and grading of site
2) set up of temporary structures
3) installation of temporary utilities
4) preparation of a storage yard
5) excavation for tower and miscellaneous other foundations
6) fabricate and set concrete forms
7) install reinforcing steel, and/or bolts, tie down steel and electrical conduits
8) pour concrete
9) remove forms and backfill
10) install underground conduits and copper ground grid
11) compaction and surfacing of assembly lay-down areas

Following concrete cure and completion of site preparation, the erection equipment and the tower components will arrive at the site. The WTS erection procedure utilizes a 100 ton capacity, 240 feet tall, gin-pole with appropriate spreader bars, slings, tag lines and accessories. The gin-pole is mounted on its individual foundation adjacent to the tower foundation. The tower components (approximately 16) are field welded into three sections. The tower sections are installed on the foundations and circumferential welding of the three sections is completed. With the tower erected, the man-lift and the electrical equipment within the tower is installed. Likewise, the equipment (main transformer, bus tie contactor and field current limiting resistor) located on the electrical equipment foundation and the buried electrical cable from the tower to the yard electrical equipment is installed.

At this time, the nacelle has arrived at the site and is being assembled and prepared for erection on the tower. The on-site nacelle assembly consists of re-installing the yaw bearing, gear box, generator, gear box lubrication system, oil coolers, wind sensors, etc., and completing electrical and hydraulic connections to this equipment. After functional checkout, the nacelle will be erected on top of the tower and bolted to the support flange. Electrical connections at the yaw slip ring will complete nacelle installation.

Assembly of the five rotor sections is accomplished on a lay-down area adjacent to the tower. After rotor field assembly and functional test, the rotor will be hoisted and bolted to the low speed shaft. Electrical connection at the rotor/nacelle slip ring and hydraulic connections will complete WTS erection. The next phase consists of system checkout and acceptance testing as discussed in Section 6.3.

The foregoing discussion described the site preparation and installation of a single prototype unit. The site preparation and installation of a multi-unit farm will be similar to a single unit except that economics of scale will be achieved by the use of reusable concrete forms, specialized test and alignment tools and fixtures, and modified heavy lift equipment. Economies may be achieved by use of a single area for receipt, storage, and assembly of the major WTS assemblies with subsequent transport within the farm area.

6.3 TEST

The test activities carried out during the conceptual and preliminary design phase were predominantly oriented towards supporting the on-going design effort. These test activities were undertaken to confirm proposed design approaches, or to provide data from which a design approach could be selected; in either case, they are categorized as development tests and are summarized in paragraph 6.3.1.

A second category of test activity was pursued during preliminary design. This activity consisted entirely of planning associated with the checkout and acceptance tests of the MOD-2 WTS as defined in preliminary design. This activity is summarized in paragraph 6.3.2.

6.3.1 Development Tests

6.3.1.1 Wind Tunnel Test Program

Objectives:

The primary objective of the wind tunnel test was to obtain data for verifying MOSTAB computer codes used for coupled dynamic analysis of a soft tower wind turbine system.

Several secondary objectives developed as the test program definition was refined and the conceptual and preliminary design progressed. Included in this set of objectives were the comparison of a fixed and a teetered hub rotor, and assessment of a rotor utilizing a controllable tip (as opposed to full span control) for adjusting rotor pitch angle.

Approach:

A one-twentieth scale model of the MOD-2 WTS was designed and fabricated by the Boeing Vertol wind tunnel organization to requirements provided by the MOD-2 project. The original model incorporated full-span pitch control of the blades and a braced tower. Machine scaling of geometry, mass, stiffness and frequency was carried out in the model design. When it became impossible to hold nacelle mass to the proper scaling (due to the weight required for the model torque absorber system), compensating mass was added to the tower to obtain the proper frequency relationship of the tower. When the evolution of the design progressed to the selection of a system of partial span (tip) control of collective pitch, the model was modified to this configuration for a second test.

Program:

Model design was started in December of 1977. Design was essentially completed and a review conducted on March 1, 1978. Fabrication of the model was completed by the end of April, and the test was conducted during the period from May 4 through May 16, 1978. Fabrication of the partial span control blades and other parts was completed by the first week of August; the second series of tests were conducted during the period of August 15 to August 19.

Significant Results:

The following is a summary of the more significant results of the wind tunnel tests (See section 5.1.1 for detailed results and correlation with the MOSTAS computer program):

(1) MOSTAB was shown to provide a good prediction of power and of steady components of flapwise and chordwise bending moments. MOSTAB was found to underpredict the magnitude of the alternating flapwise moment; on the basis of wind tunnel results a factor of 1.65 must be applied to these predictions. A modification to the program was made to incorporate this factor.

(2) The viability of the soft tower concept was proven over the operating wind speed range.

(3) The teetering rotor was shown to be superior to the fixed rotor by lowering flapwise alternating moments by approximately 50 percent.

(4) MOSTAB predictions for performance data is in good agreement with test results for the tip control configuration.

(5) Performance of the tip control model compared well with the full-span model.

6.3.1.2 Hydraulic Reservoir Test

Objectives:

The objective of the hydraulic reservoir test was to evaluate the rotating reservoir which supplies the blade pitch control hydraulics system. For the MOD-2 design this reservoir is mounted on the low speed shaft and therefore rotates at 17.5 rpm. The evaluation was to include an assessment of the degradation of the hydraulic fluid, the possibility of air entering the pump inlet, the operation of a liquid level indicator, and the possibility of oil flow through the air vent.

Approach:

A clear plastic full scale reservoir which permitted visual observation of the interior was fabricated and mounted on a trunnion that simulated the proposed installation on the low speed shaft. The reservoir was connected to a hydraulic bench through swivel connections, and a means of rotating the trunnion at variable speeds was incorporated.

Program:

Test requirements were released on September 15 and test design and necessary procurements started immediately. The test setup was completed on October 25 and testing was conducted during the period October 26 through November 7.

Tests were conducted at rotational speeds of 17.5 and 20 rpm using various combinations of fluid flow and reservoir levels. Static tests were also run with the inlet at various azimuthal locations.

Significant Results:

Initial runs showed that severe fluid aeration occurred when the return line inlet was above the fluid level. It was also noted that some oil fell into the vent pipe with each revolution. The reservoir was modified by adding a diffuser to the fluid inlet pipe and a shield to the vent pipe inlet.

The final test runs indicated that oil flow patterns were acceptable and fluid bulk modulus remained unchanged. There was no oil flow from the air vent and the liquid level plunger force was adequate to operate the low level switch.

6.3.1.3 Blade Buckling Test

The combination of curved and flat sheets 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 used in blade des: . The objective of the blade buckling test was to verify the analysis methods used for buckling under axial compression and bending compressive loading.

Approach:

A full scale section representing the MOD-2 blade preliminary design was prepared using the forming and welding procedures developed for the prototype blade fabrication. This section, 35 feet in length, represented the geometry from the station 360 field splice flanged joint to station 780. In the interest of economy, plates used were standard thickness and therefore deviated slightly from preliminary design sizes. This had no effect on the objective of the buckling program. Load transfer structure was built into the specimen during fabrication.

Test fixtures were designed and fabricated which would permit interfacing the specimen with the structural test strongback.

Program:

Final test requirements were released on September 15. Test fixture and test set-up design was accomplished by October 13. Fabrication of the test fixtures was accomplished in the period between October 16 and November 2, 1978.

The test specimen was fabricated by the MOD-2 manufacturing shop and was available for the test on November 3. Instrumentation of the specimen, installation in the strongback, and completing the test set-up was accomplished by November 13. The test was conducted on November 14 and 15, 1978. Subsequent to the test the specimen was subjected to a stress relief cycle in the Plant II furnace. Measurements were taken before and after the cycle to determine the amount of distortion which could be expected from stress relieving. The test speciment is shown in Figure 6-4.

The test consisted of applying an axial compressive load in the form of a couple and a bending compressive load at the outboard end of the specimen. These loads were applied simultaneously and in increasing percentages of the test load. The specimen was subjected to 148% of the predicted ultimate strength without evidence of buckling.

Significant Results:

The primary objective of 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. See section 5.1.3.2 for a discussion of test results.

A second important result was derived in the form of a significant development of the manufacturing blade assembly procedures, including the welding, forming, and handling. Heat treatment caused no shape distortion.

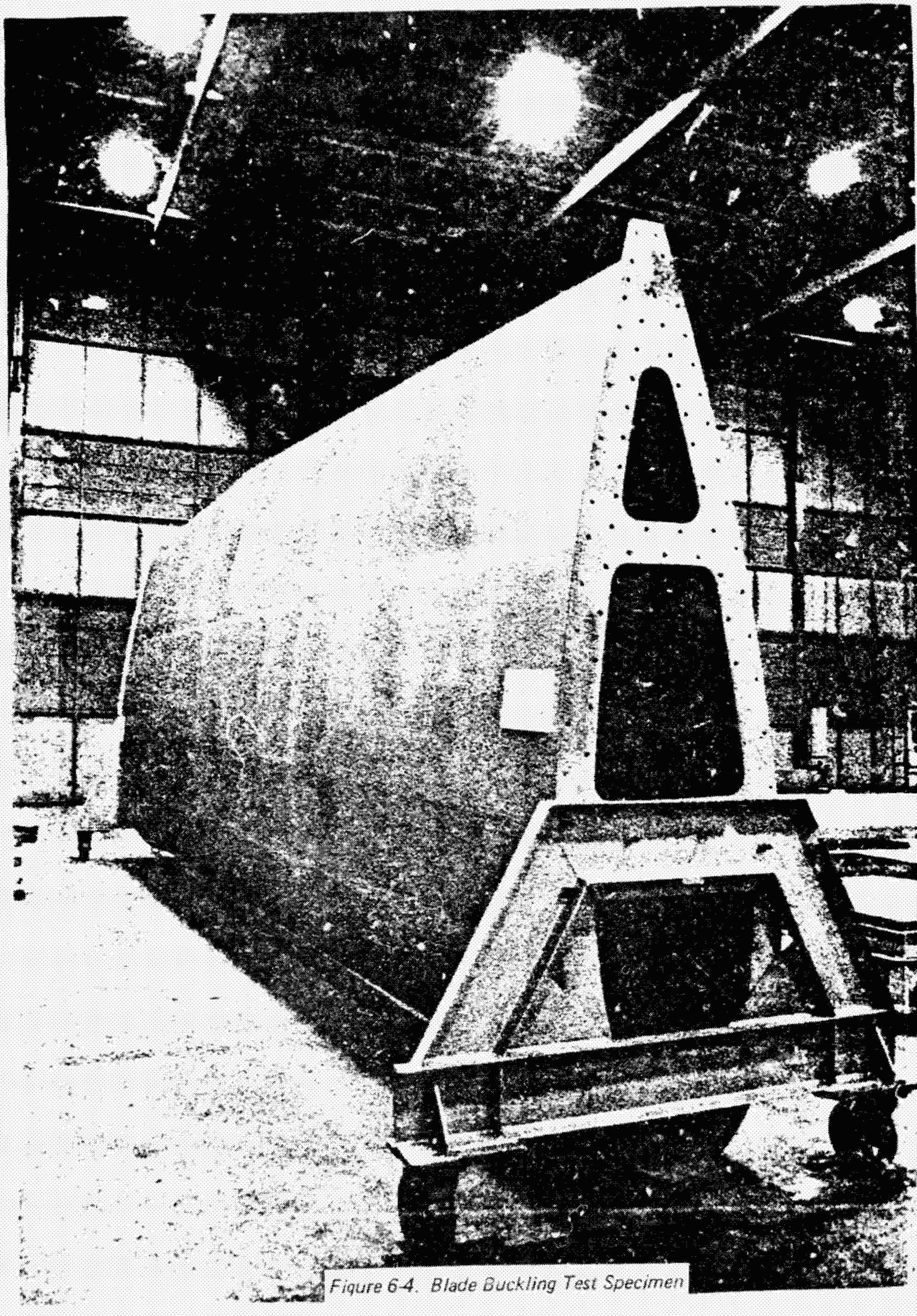

Figure 6-4. Blade Buckling Test Specimen

6.3.1.4 Fatigue Allowables Test Program

Objectives:

The design life of the MOD-2 WTS is 30 years. The rotor blade will therefore experience 2.1 X 10^8 revolutions during this period (4.2 X 10^8 load cycles at 2/rev) and fatigue life becomes the controlling factor in its design. Present data on fatigue life generally extends only to 1 X 10^7 cycles; therefore, the initial objective of the test program was to extend fatigue data to cover the MOD-2 experience and derive fatigue allowables to use for rotor blade design. As the program progressed, the following secondary objectives developed: the verification of the allowables for different spectrum loading conditions, evaluation of three candidate steels, evaluation of the influence of localized residual stress on fatigue life, and determination of the allowables to be used in the blade field splice.

Approach:

A series of tests were conducted on uniaxially loaded surface flawed specimens. The specimens were subjected to spectrum loading representative of expected service loading on the tension side of the blade. Initially, the spectrum was representative of the load profiles expected for the fixed hub rotor for both the upwind and downwind rotor configurations. Specimens were made from A533 steel which was initially expected to be a candidate for the MOD-2. From the results of these tests, a fatigue design allowable model was developed.

When the partial span pitch control concept and its corresponding load spectrum was adopted in the conceptual design phase, and blade material candidates were reduced to three, a second series of tests were conducted to confirm the applicability of the model developed as a result of the first series of tests.

Program:

The first series of tests was started in November, 1977. Twenty-two specimens were prepared and eighteen of these were tested. Each specimen contained a pre-cracked surface flaw .06 inches deep by .25 inches long. Specimens were tested in machines automatically controlled by computer, which permitted programming spectrum load cycles on a 24 hours, 7 day work basis. Five machines were used at the peak of the program. Testing on the first series continued through May 1978. The second series of tests was started in August 1978 and continued through November 1978. During this series the four specimens not used on the first series plus seven additional specimens were tested using spectrum load cycles representative of the tip control configuration. Six machines were used at the peak of this program.

Significant Results:

The primary and secondary objectives were achieved. The results of the test program were used to develop a fatigue allowables model which is applicable to the expected 30 year life of the program. See section 5.1.2 for the description of the fatigue allowable model. The model accurately predicted the expected life of the specimen tested during phase two. Test data showed that the model was applicable to all four of the A grade steels tested.

6.3.2 Acceptance Test

As the conceptual and preliminary design of the MOD-2 WTS progressed, and the major features of the configuration became known, a plan for its testing was initiated. The plan and its later implementation is part of an overall systems engineering approach to the MOD-2 project. The plan identifies the various steps by which incremental assurance in the design and manufacture of the WTS elements will be attained, starting from the tests of components, through assembly and test in the factory, and ending with operational and acceptance tests at the chosen MOD-2 WTS site.

A primary source of requirements in developing the test plan is Exhibit B of the MOD-2 Statement of Work. Other requirements are derived from an analysis of drawings, the structural criteria, and lower level hardware specifications.

The preliminary plan calls for a comprehensive series of tests on the completely assembled nacelle and rotor in the manufacturing assembly area, independent tests of the erected tower at the chosen field site, checkout of the fully assembled WTS at the site to assure that it is operating properly under wind power, and finally, a series of acceptance tests during which data is obtained from which it will be shown that the MOD-2 WTS is operating within the specified design limits and is capable of operating in a utility network. A list of the engineering measurements to be implemented to support the acceptance tests will be included in the release of the system test plan.

APPENDIX A

LIST OF SYMBOLS AND ABBREVIATIONS

SYMBOL	MEANING
a	power scaling exponent
a	windage churning loss
a	length
A	amplitude of gusts
AEP	annual energy production
AH	amphere hours
AISC	American Institute of Steel Construction
Alt	altitude
Ap	particular value of "A"
AOM	annual operations and maintenance costs
b	width
BEC	Boeing Engineering and Construction
BFL	basic factory labor
BTU	British thermal unit (s)
c	cord (airfoil)
c	Weibull constant
c.g.	center of gravity
C_D (C_d)	drag coefficient
C_L (C_l)	lift coefficient
COE	cost of electricity
COEP	cost of electricity program
C_p	power coefficient
C.P.G	compact planetary gear

SYMBOL	MEANING
CPM (CP_m)	maximum system efficiency
CP_{max}	maximum rotor power coefficient
C_r	critical buckling stress
CRT	computer remote terminal
Cu	copper
cu. ft.	cubic feet
C.Y.	cubic yards
D (DIA)	diameter
da/dn	crack growth rate
db	decibal (s)
DC	direct current
DG	diesel driven generator
DOE	Department of Energy
E	modulus of elasticity
EOCP	energy output computer program
E_s	specific energy
f	limit load stress
F_{cr}	critical load
FCR	fixed charge rate (annualized)
FMAT	material factor
FMEA	failure mode and effects analysis
FMFG	manufacturing factor
f_n	frequency requirement
F (P)	load
FPS	feet per second
FR	failures per year

freq.	frequency
ft.	feet
$ft.^2$	square foot
g	acceleration due to gravity
G	modulus of rigidity
GCB	generator circuit breaker
GR	steel grade
hr(s)	hour(s)
Hz (HZ)	Hertz (cycles/sec.)
IC	total WTS cost
in (s)	inch (es)
K	Weibull constant (exponent)
K	one thousand
K_c	buckling constant
K_{max}	maximum stress intensity for each block of cycles
Kol	maximum stress intensity in each spectrum
KPC	pitch control cost factor
KRO	baseline value (cost)
Ksi	thousand pounds per square inch
KV	kilovolt (s)
KVA	Kilovolt amphere (s)
KW	Kilowatt (s)
KWh	kilowatt hour (s)
Kips	thousand pound (s)
lb(s) (#)	pound (s)
L.E.	leading edge
LERC (LRC)	Lewis Research Center

m	meters
M	moment
max	maximum
MDT	mean down time
min	minumum
mph	miles per hour
M.S.	margin of safety
MSB	most significant bit
MTBF	mean time between failure
MW	mega watt (s)
MWh	mega watt hour (s)
n	number of spares
n	circular frequency
N	efficiency
NACA	National Advisory Committee for Aeronautics
NASA	National Aeronautic and Space Administration
O & M	operations and maintenance
O.D.	outside diameter
PF	power factor
P_s	specific power
psi	pounds per square inch
Powgn	power error gain
PT	particle test method
r	radial distance from hub center
R	radius of WTS rotor
R	minimum stress/maximum stress

RAD	radian (s)
rev	revolution
ROT	rotor cost
rpm	revolutions per minute
RT	radio-graphic test method
RTGN	rate error gain
s	stress
s (sec)	second (s)
SCF	structural composites industries
STD	standard temperature day
t	thickness
t	reorder time
T.E.	trailing edge
TFT	transverse filament tape
T_w	torque extracted from wind
UPS	uninterruptable power supply
UT	ultrasonic test method
v	volt (s)
V (V_w)	wind velocity
V_r	rated wind velocity
Vel	velocity
V_{in}	cut-in wind velocity
V_c	design wind velocity
V_o	empirical homogenous wind speed
V_{out}	cut-out wind velocity
V_r	reference wind velocity
VT	visual test method

WT	weight
WT	wind turbine
WTS	wind turbine system
x	reference axis
y	reference axis
yr	year (s)
z	reference axis
z	elevation above ground level
z_o	surface roughness length
z_r	reference height
'	feet
"	inches
¢	cents
$	dollars
%	percent
^oF	temperature, Fahrenheit
^oC	temperature, Centigrade
$F	cost of failure
^{o}	degrees
$<$	less than
$\leq$	less than or equal to
$=$	equal to
$>$	greater than
$\geq$	greater than or equal to

<<<	much less than
>>>	much greater than
λ	failure rate
σ	solidity
ϕ	phase (electrical)
ϕ ($\emptyset$)	wind yaw error
σ	standard deviation
θ_C	collective pitch angle
ϕ_x	longitudinal component of turbulence spectrum
σ_x	longitudinal turbulence intensity
Δ	wind velocity separation distance
ρ	air density
$\dot{\theta}$	rotor rate
Δ	change
ρ	density
μ	Poisan's Ratio
θ	pitch angle
η	efficiency

APPENDIX B

Contract DEN3-2, Exhibit B

Specifications and Constraints as Modified by

Contract Amendment No. 7, dated December 26, 1978

SPECIFICATIONS AND CONSTRAINTS

1.0 User Requirements

1.1 The wind turbine shall generate rated power at all wind speeds between rated and cut-out speeds, and will operate at reduced power output at wind speeds between cut-in and rated.

1.2 The wind turbine shall generate three-phase 60 Hz power in the megawatt range.

1.3 The wind turbine, including all components, shall be designed for a useful utility service life of 30 years.

2.0 System Design Requirements

2.1 General

2.1.1 The wind environment that the wind turbine will be exposed to is that resulting in a yearly mean wind speed of 6.3 m/s (14 mph). The maximum wind loads shall be computed for a maximum design wind of 53.6 m/s at 9.1 m (120 mph at 30 ft) above grade, and a gust factor of 0.

2.1.2 The unit shall be designed for a minimum availability of 90 percent over the service life with special consideration given to servicing and maintenance of critical areas.

2.1.3 Network service protection against wind turbine faults and turbine protection against network faults shall be provided.

2.1.4 The system design shall provide a method for safely transmitting lightning strikes anywhere on the system to the ground.

2.1.5 The color of any coatings and other identification markings used on the wind turbine shall be provided by the NASA Project Manager.

2.2 Rotor

2.2.1 The design shall be of horizontal-axis configuration.

2.2.2 The rotor shall have a minimum diameter of 91.4 m (300 feet).

2.2.3 The wind turbine shall be capable of self-protection in the event of an emergency.

2.3 Control Modes

2.3.1 Manual operation of the wind turbine at the site.

2.3.2 Remote monitoring and control by a dispatcher located a considerable distance from the site. The distance will depend on the selected site.

2.3.3 Unattended, fail-safe automatic wind turbine operation.

2.4 Design Codes and Standards

All applicable design codes and standards shall be used in the design, fabrication, installation and operation of the wind turbine.

2.5 Data System Instrumentation

2.5.1 The engineering instrumentation, which shall be limited to a maximum of 96 channels, shall include the following items:

Component	Measurements
Blades	Strain gages to measure flapwise, chordwise, and torsional strains on each blade.
Hub	Blade pitch angle, strain gages located in areas of stress concentrations.
Shafts	Rotating speeds (high and low), strain gages on low-speed shaft to measure shaft torque, bending moments and thrust loads, blade position in rotation, proximity probes to measure shaft position.
Transmission	Oil temperature
Brake(s)	Temperatures
Generator	Rotating speed, winding temperatures, field current, voltage out (each phase), current out (each phase), KW and KVARS.
Bed Plate	Strain gage
Bearings	Temperatures, vibration
Hydraulic System	Oil pressure, oil temperature, oil level, pump motor temperature
Nacelle	Temperatures at various locations.

Yaw Drive	Motor temperature, angular position of nacelle, strain gages to measure yaw torque.
Environment	Ambient temperature, wind direction, wind speed.

2.5.2 The operation instrumentation shall record parameters such as temperature, pressure, voltage and power which are necessary to meet the operational needs of the participating utility. No rapidly changing signals such as vibration and dynamic strains need be recorded.

3.0 Environmental Constraints

3.1 Wind Environment

3.1.1 Steady Wind Model: The annual distribution of steady wind speeds is given by the following Weibull distribution:

$$p(V \geq V_p)_r = \exp\left[-\left(\frac{V_p}{c}\right)^k\right]_r$$

Where
$p(V \geq V_p)$ = probability that $V \geq V_p$
V = steady wind speed, m/s
V_p = prescribed value of V
c = empirical constant = 7.06 m/s
k = empirical constant = 2.27

The subscript, r, indicates the parameter is evaluated at a reference elevation of 9.1m (30 ft.). The empirical Weibull constants, c and k, specified above define a site with a mean yearly wind speed of 6.3 m/s (14 mph) at 9.1m (30 ft.).

3.1.2 Wind Shear Model: Wind speeds at elevations other than the reference elevation are given by the following equations:

$$V = V_r \left(\frac{h}{h_r}\right)^{\alpha}$$

Where
V = steady wind speed at elevation of interest, m/s
V_r = steady wind speed at reference elevation, m/s
h = elevation of interest, m
h_r = reference elevation = 9.1m (30 ft.)

and

$$\alpha = \alpha_o \left(1 - \frac{\log V_r}{\log V_o}\right)$$

$$\alpha_o = \left(\frac{z_o}{h_r}\right)^{0.20}$$

z_o = surface roughness length = 0.05m (0.16 ft.)
V_o = empirical homogeneous wind speed = 67.1 m/s (150 mph)

3.1.3 Turbulence Model: The wind turbulence associated with a specific steady wind speed at the reference elevation is a Gaussian random process with the following turbulence intensity:

$$\sigma_{r,x} = V_r \left[\frac{1.04}{\ln\left(\frac{h_r}{z_0}+1\right)} \right]$$

$$\sigma_{r,y} = 0.8\ \sigma_{r,x}$$

$$\sigma_{r,z} = 0.5\ \sigma_{r,x}$$

where σ_r = turbulence intensity (i.e., the standard deviation of turbulent fluctuations) associated with V_r, m/s

and the subscripts x, y, z indicate longitudinal, lateral, and vertical directions, respectively. The turbulence spectrum for the longitudinal component of turbulence is

$$\phi_x(n,h,V) = \frac{\sigma_{r,x}^2}{n} \left[\frac{0.164\,(\eta/\eta_{o,x})}{1+0.164(\eta/\eta_{o,x})^{5/3}} \right]$$

where n = circular frequency, Hz
ϕ_x = annual average spectrum for longitudinal component of wind turbulence, m^2/s
η = reduced frequency = nh/V, dimensionless
$\eta_{o,x}$ = dimensionless constant = 0.0144

Similar equations apply to the lateral and vertical components with:

$\eta_{o,y}$ = 0.0265
$\eta_{o,z}$ = 0.0962

The longitudinal turbulence intensity at a given elevation and steady wind speed for a specific wind turbine is given by the following equation:

$$\sigma_x = \left[\int_{n_{min}}^{n_{max}} \phi_x\, dn \right]^{1/2}$$

in which the circular frequency limits, n_{max} and n_{min}, are determined from the response characteristics of the specific wind turbine. Similar equations apply for the lateral and vertical turbulence intensities, σ_y and σ_z, respectively. The corresponding longitudinal gust amplitude is distributed according to the Gaussian random process:

$$p(A_x) = \frac{1}{\sqrt{2\pi}\,\sigma_x} \exp\left[-\frac{1}{2}\left(\frac{A_x}{\sigma_x}\right)^2 \right]$$

where $p(A_x)$ = probability density that a longitudinal gust will occur with amplitude A_x, s/m

A_x = longitudinal gust amplitude, m/s

Similar equations apply for the lateral and vertical gust amplitudes, A_y and A_z, respectively. The turbulence of the winds consists of a set of discrete gusts with Gaussian random amplitudes, as defined above, and with shapes and periods specified by the following equations:

$$V_x(t) = V_x + A_x\left[1-\cos\left(\frac{2\pi t}{T}\right)\right], \quad 0 \le t \le T$$

with similar equations for $V_y(t)$ and $V_z(t)$, and

$$T = \begin{cases} T_o/2, & 2n_{min} \le 1/T_o \le 2n_{max} \\ 1/2n_{min}, & 1/T_o < 2n_{min} \\ 1/2n_{max}, & 1/T_o > 2n_{max} \end{cases}$$

where t = time, s

T = gust period, s

T_o = most probable period of a gust with amplitude A, s

3.1.4 Site Data: Once the site at which the wind turbine is to be installed is identified, any new wind data from this site will be documented and furnished to the contractor.

3.2 Lightning Model

The currents flowing in a lightning flash to ground are separated into three categories:

a. Return stroke surges	Peak current on the order of up to 100,000 A or more. Duration on the order of tens of microseconds.
b. Intermediate currents	Peak current on the order of up to 10,000 A or more. Duration on the order of milliseconds.
c. Continuing currents	Peak current on the order of up to 1000 A. Duration on the order of hundreds of milliseconds.

The current time history for a model lightning flash to ground is shown diagrammatically in figure B-1 with some further details being presented in Table B-1. The flash is a very severe discharge. The model is formulated in terms of certain key points (A through I in figure B-1) at which specific values of current, i, and time, t, are attained. Between successive key points the current is assumed to change in a steady straight-line fashion with time.

The model is essentially developed for applied purposes, and it has consequently been simplified. In physical reality, a severe discharge would have far more strokes and also, more phases of continuing current than indicated on figure B-1; however, the integrated effects of a very severe natural discharge and of the model are similar.

3.3 Seismic Loads

The entire wind turbine assembly shall be capable of withstanding Zone 3 seismic loads without sustaining damage to the assembly or any of its components.

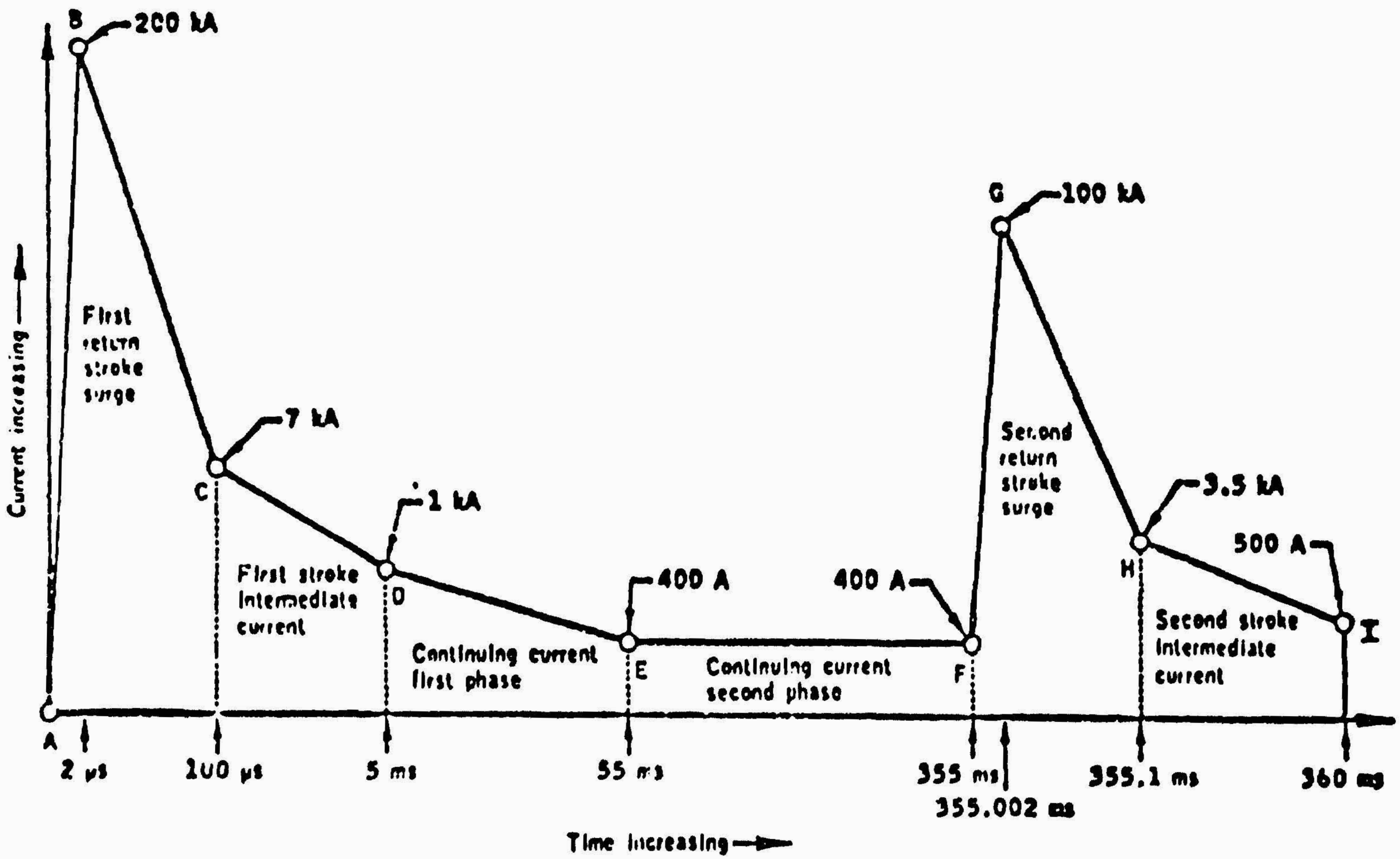

Figure B-1.- Diagrammatic representation of lightning model. (Note that the diagram is not to scale.)

TABLE B-1.- DETAILS OF LIGHTNING MODEL

Stage	Key points		Rate of current change	Charge passing
1. First return stroke surge	t = 0	i = 0		
	t = 2 μs	i = 200 kA	Linear rise - 100 kA/μs	0.2 C
	t = 100 μs	i = 7 kA	Linear fall - 193 kA in 98 μs	~ 10.2 C
2. First stroke intermediate current	t = 100 μs	i = 7 kA		
	t = 5 ms	i = 1 kA	Linear fall - 6 kA in 4.9 ms	19.6 C
3. Continuing current - first phase	t = 5 ms	i = 1 kA		
	t = 55 ms	i = 400 A	Linear fall - 600 A in 50 ms	35.0 C
4. Continuing current - second phase	t = 55 ms	i = 400 A		
	t = 355 ms	i = 400 A	Steady current	120.0 C
5. Second return stroke surge	t = 355 ms	i = 400 A		
	t = 355.002 ms	i = 100 kA	Linear rise ~ 50 kA/μs	~ 0.1 C
	t = 355.1 ms	i = 3.5 kA	Linear fall - 96.5 kA in 98 μs	~ 5.0 C
6. Second stroke intermediate current	t = 355.1 ms	i = 3.5 kA		
	t = 360 ms	i = 500 A	Linear fall - 3 kA in 4.9 ms	9.8 C