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DESIGN STUDY OF WIND TURBINES
50 kW TO 3000 kW FOR ELECTRIC UTILITY APPLICATIONS
VOLUME II - ANALYSIS AND DESIGN

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16. Abstract
This study was one of two parallel efforts conducted to define the wind turbine configuration that would lead to generation of electrical power in a cost effective manner. All possible overall system configurations, operating modes, and subsystem concepts were evaluated for both technical feasibility and compatibility with utility networks, as well as for economic attractiveness. A design optimization computer code was developed to determine the cost sensitivity of the various design features, and thus establish the configuration and design conditions that would minimize the generated energy costs. The preliminary designs of both a 500 kW unit and a 1500 kW unit operating in a 12 mph and 18 mph median wind speed respectively, were developed. This report presents a detailed technical discussion of the various design features and components evaluated, and the rational employed to select the final design configuration. All pertinent technical performance data and component cost data is included. The costs of all major subassemblies are estimated and the resultant energy costs for both the 500 kW and 1500 kW units are calculated.

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SECTION 1. 0

INTRODUCTIÓN

SECTION 1.0

INTRODUCTION

This study was performed by General Electric's Advanced Energy Systems in fulfillment of contract NAS 3-19403, "Program for Conceptual Design, Parametric Analysis and Preliminary Designs for Low Power (50-250KW) and High Power (500-3000 KW) Wind Generator Systems". The program was directed by the NASA Lewis Research Centers' Wind Power Office, Power Systems Division, for the Energy Research and Development Administration and is an integral part of the Federal Wind Energy Program. This effort was one of two parallel studies directed by NASA to evaluate the use of wind turbines to generate electrical power in a cost effective manner.
General Electric was supported in this study by the Hamilton Standard Division of the United Technologies Corporation.

1.1 PROGRAM OBJECTIVES

The overall objective of this study was to determine the incentive for electrical utility companies to incorporate Wind Turbine Generators (WTG's) in an electrical power network. Specifically, the program's activities were directed at:

- Establishing the design and operational requirements for a WTG which is linked to an electrical utility network
- Defining the WTG design and operational concept most suitable for this application
- Determining the most cost effective operating conditions as a function of sites having median winds between 9 and 21 MPH.
- Generating two Preliminary Designs and cost data - one system at a 12 MPH median wind site and one at an 18 MPH median wind site.

Assuming the generation of results favorable to the continued investigation of WTG's for this application, it is intended to use the product of this study (and a parallel effort being funded by NASA/ERDA) as the basis for the final design and construction of actual systems. Therefore, particular emphasis was placed upon a comprehensive documentation of the Preliminary Designs and supporting material.

1.2 PROGRAM REVIEW

In order to achieve the program objectives an 8 month technical effort was planned; the major tasks involved in the study are shown in Figure 1-1. The first step in the program was to characterize the WTG in terms of its configuration, operating mode and subsystem concepts. This task was supported by the Hamilton Standard team in the rotor design area and also by Professor Dr. Ulrich Hutter of the University of Stuttgart (Reference 1-1).

In a parallel effort, the General Electric Utilities System Engineering Division (EUSED) evaluated the applicability of WTG's for use in an electric utility network. This investigation considered both technical and economic requirements of the utility industry. The output of this task was used in the Conceptual Design Task as well as latter tasks.

Results of the Conceptual Design effort served as the basis for developing a design optimization computer code. The purpose of this task was to model the WTG performance for various design conditions. This code was then used in the Parametric Analysis task to determine the sensitivity of WTG cost to the following variables:

- mean wind speed
- power level
- system life
- rotor speed
- rotor diameter
- rated-to-mean wind speed ratio
- rated velocity ratio
- activity factor (a solidity term which relates to the shape of the blade planform.)
- airfoil section
- blade ground clearance
- other secondary variables

The conclusions drawn from the Parametric Analysis were used to determine the design conditions for the Preliminary Designs to be generated in the final task. During this portion of the study a series of trade-offs were made between the rotor subsystem design and the remainder of the WTG System. An outline of the interactions occurring at the subsystem level to arrive at the overall system design are shown in Figure 1-2.

Two Preliminary Designs were generated in this portion of the study: a 500KW unit and a 1500KW unit operating in 12 and 18 MPH median wind sites, respectively. These designs are discussed at length in Section 5.0.

1.3 SUMMARY

This section summarizes the study approach and results from a Wind Turbine Generator (WTG) design viewpoint. The design conclusions presented relate to the overall system design as opposed to individual WTG subsystem design selections. A comprehensive outline of the study conclusions is presented

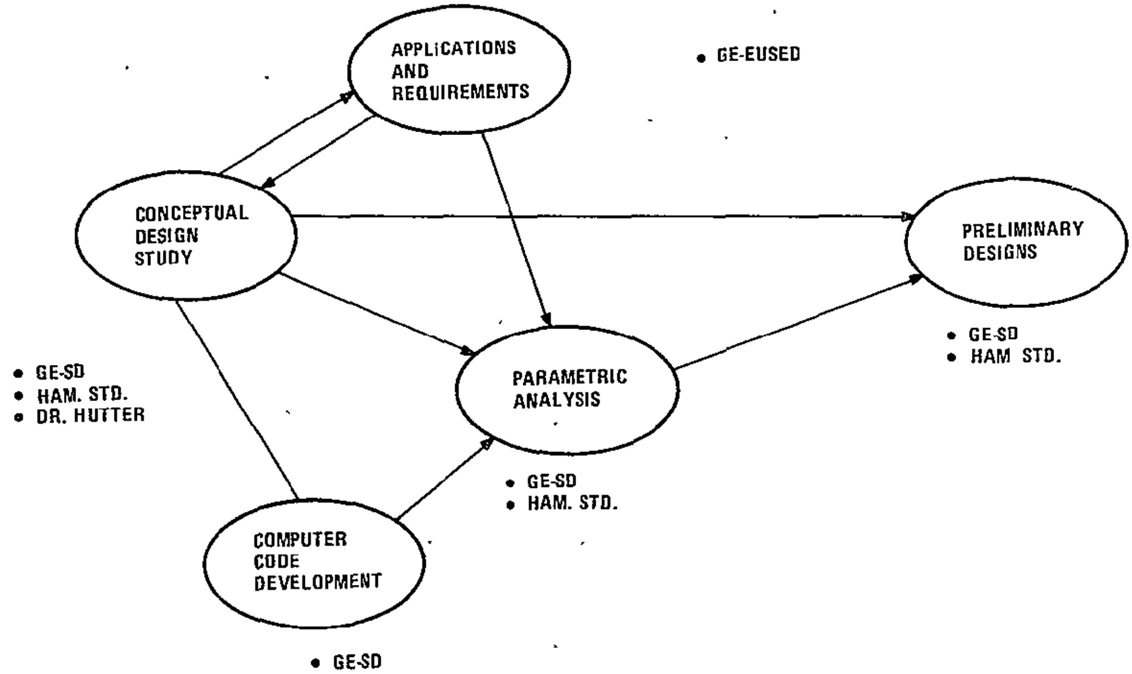

Figure 1-1. Study Approach

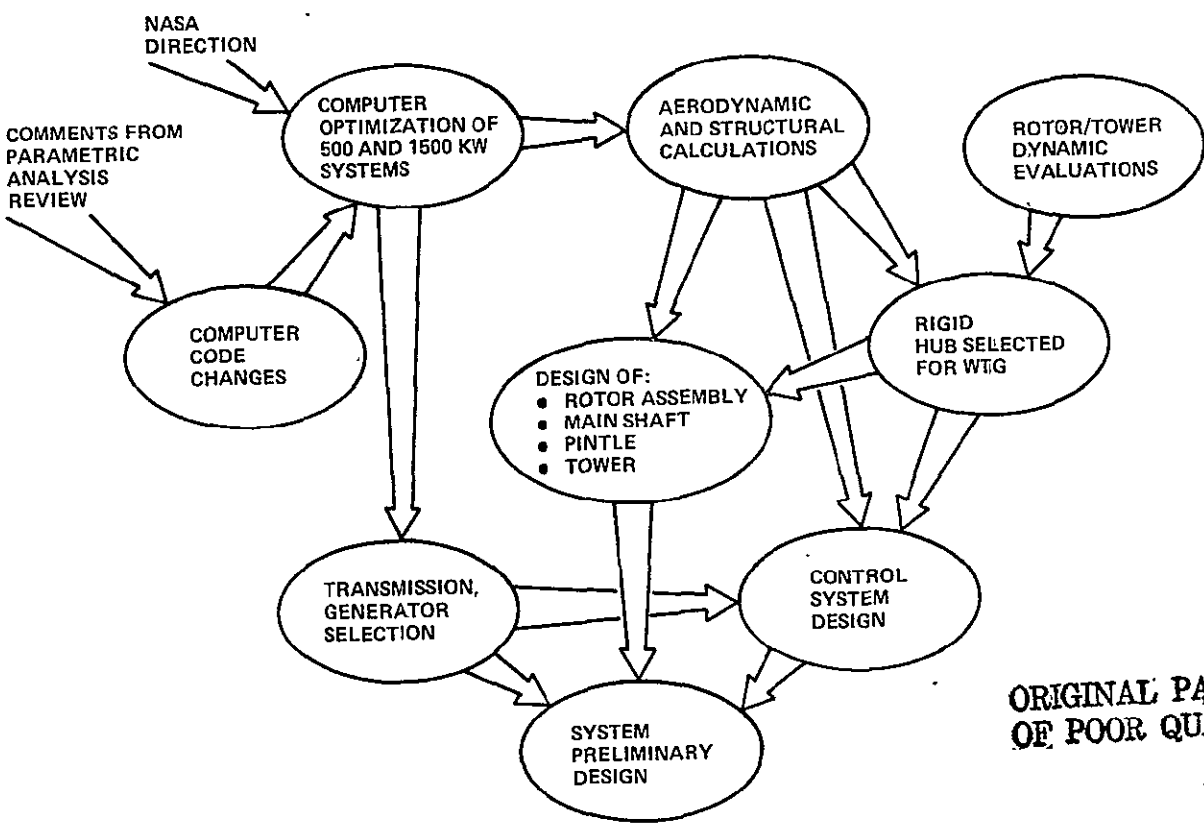

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Figure 1-2. Preliminary Design Work Flow in Section 6.0 of this report.

The theme which dominated the overall conduct of the system design effort was to minimize capital and maintenance costs while maximizing the energy capture. Throughout the study this approach led to the use of commercially available components, designs allowing ease of maintenance and the utilization of a versatile control system.

Two preliminary designs were generated in the study: one for operation at a 12 MPH wind site and one at an 18 MPH wind site. Both of these designs were based upon the same subsystem technologies and overall configuration. An illustration of the general design features is shown in Figure 1-3.

In Figure 1-3 the rotor has been placed downwind from the tower. This location is preferred for a variety of reasons including: the superior stability of this configuration to changes in wind direction and a reduction in the main shaft overhang requirements; further discussion of this point is contained in Section 3.3.2. Attached to the shank portion of the blades are pitch actuators which are instrumental in maintaining a constant RPM under variable wind conditions. Operation at constant RPM, as opposed to constant velocity ratio, was selected since the increase in energy capture which is available with the constant velocity ratio system was not sufficient to offset the cost of either a variable speed transmission or generator.

Forward of the rotor subsystem is the nacelle which houses the mechanical power transmission and electrical generator. During the study, efforts were made to place the gearbox portion of the transmission and the generator on the ground. In the case of a mechanical transmission right angle drives are needed in a size range which are unavailable; since the scope of this effort was limited to state-of-the art hardware this approach was not pursued further. Hydraulic transmissions were also considered, however, they necessitated the use of large, expensive pumps which result in an equally unattractive approach (Section 3.4.2).

The entire rotor and nacelle are mounted on the pintle which transfers the loads imposed on the system into the tower. At the top of the pintle is an azimuth bearing which allows the nacelle to rotate atop the tower. The pintle is connected to the tower by means of a transition structure (not shown in Figure 1-3).

The power levels selected for the two systems are 500KW and 1500 KW for the 12 MPH and 18 MPH wind site units, respectively. These power ratings were determined from the optimization studies which indicated that, for the rotor cost assumptions used (see Section 4.3.1.2), low cost systems could be achieved by going to large blade diameters which provided relatively high power ratings. Figure 1-4 illustrates the parametric results supporting this conclusion. Examination of Figure 1-4 reveals that low cost systems can be obtained by designing for high power levels in good wind sites, however, high power also requires large blade diameters. The fact that lower cost systems could be achieved by going to larger blade diameters was a fundamental conclusion which had widespread implications throughout the study.

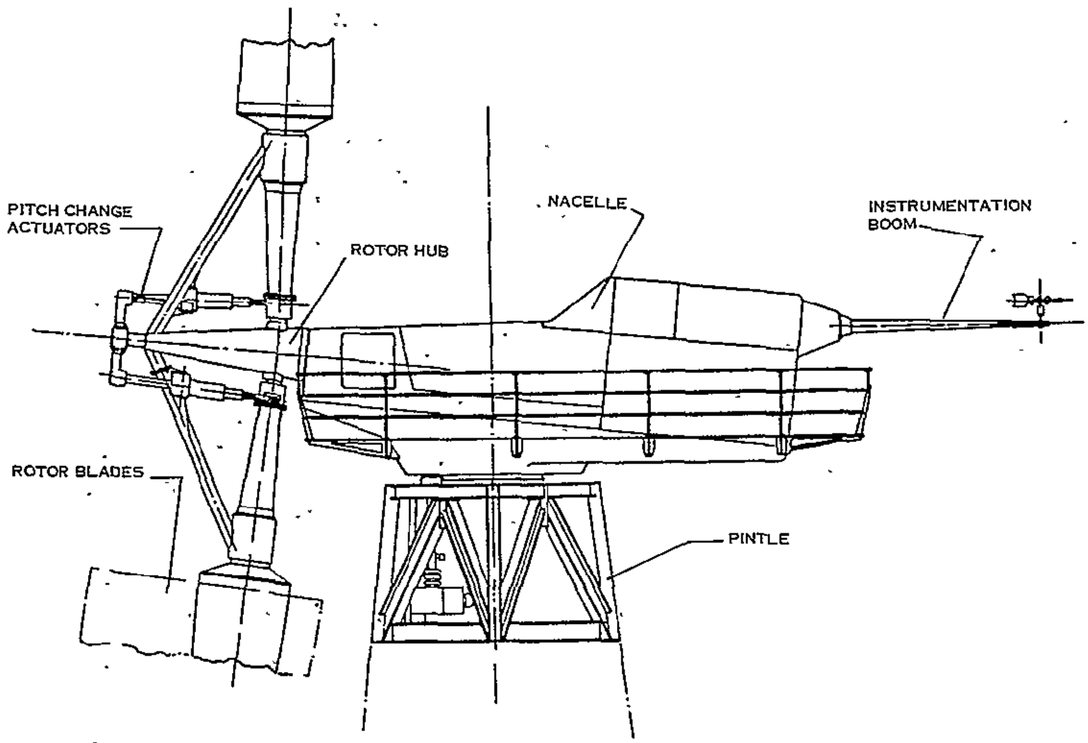

Figure 1-3. Preliminary Design Exterior View

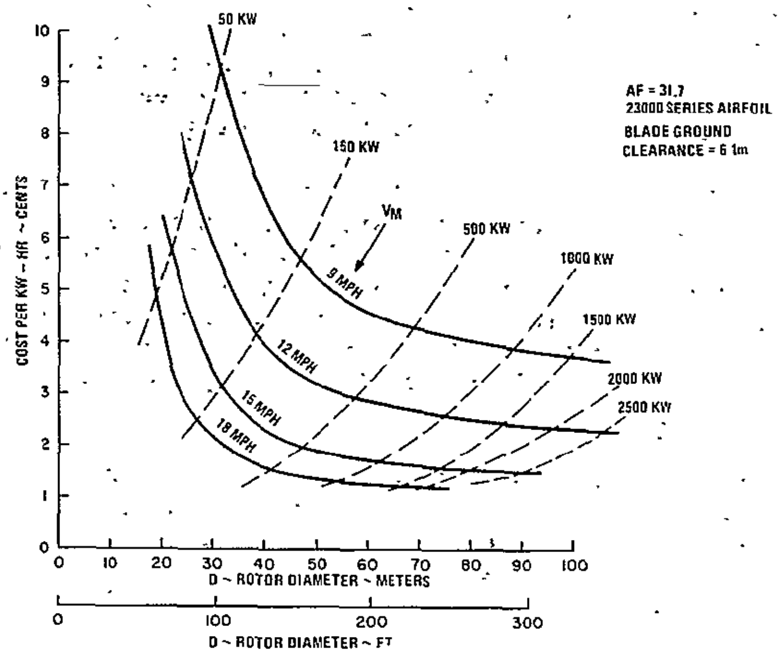

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Figure 1-4. Effect of Blade Diameter Upon Power Generation Cost

The design conditions selected for each of the WTG Systems were selected by a design optimization computer code; these conditions are given in Table 1-1. As shown in the table, the blade diameters for the units are nearly equivalent although one is rated at 3 times the power of the other. This is a result of the wind velocity at which the machines are rated, which is in turn a consequence of the assumed wind site. The predicted annual operating times for the machines are 6257 and 6568 hours for the 500KW and 1500KW units, respectively. These values result in capacity factors* of 42 percent for the 500KW unit and 51 percent for the 1500KW unit. High capacity factors are needed in order to decrease the energy generation costs, mils/KW-HR, which is the figure of merit used in the optimization procedure. Figure 1-5 illustrates the preliminary design layout for the 12 MPH (500 KW) and 18 MPH (1500 KW) units; dimensions are shown for both designs. Both designs utilize a 2 bladed rotor subsystem, however, 2 and 3 bladed systems were considered. The selection of 2 over 3 blades was based upon a trade-off between the lower cost of 2 blades versus the improved performance available with 3 blades (see Figure 3-8). During this exercise it was assumed that the cost of a blade in a 2 bladed rotor equalled the cost of a blade in a 3 bladed rotor.

Other hub designs were also considered including teetered hubs and a single bearing rotor/hub design. A teetered hub offers an advantage in decreasing the dynamic load factors into the tower, however, the increased cost of the teetered hub was judged to be excessive. Again, as discussed in Section 5.2.6, from a fundamental viewpoint, teetering has technical merit and may offer a design advantage under specific design conditions if a low cost design can be formulated. Placing the hub upon a bearing which supports the rotor subsystem was also investigated. This approach resulted in an overly complicated bearing design which was required to take all of the gravity, thrust and bending loads and was not commercially available. Another result of this arrangement is a larger overhanging moment on the pintle structure, thereby, requiring a more expensive pintle. Finally, the maintenance of the hub bearing is complicated since the blades must be removed prior to servicing.

The mechanical power transmission consisting of the main shaft, bearings, couplings and gearbox, and the generator are designed to permit ease of maintenance. The transmission and generator are situated at the forward end of the nacelle to counterbalance the thrust loads and gravity loads of the rotor subsystem. This approach results in decreasing the overturning moment on the blades (due to wind shear) and the bending moment across the azimuth bearing. The transmission is shown in a vertical orientation which permits access to the high speed pinion - a potential wear point.

A rigid bedplate supports all of the nacelle equipment and is also the interface structure with the azimuth bearing. The azimuth bearing is housed within the pintle which transfers the loads from the bedplate to the tower. An azimuth drive mechanism, having a slip clutch, is used to orient the nacelle into the wind. The slip clutch is a safety device which allows the nacelle/rotor to weathervane under specific wind directional changes at a controlled rate.

* Capacity factor is the energy generated within a specific time frame divided by the energy which could be generated if rated power was maintained continuously.

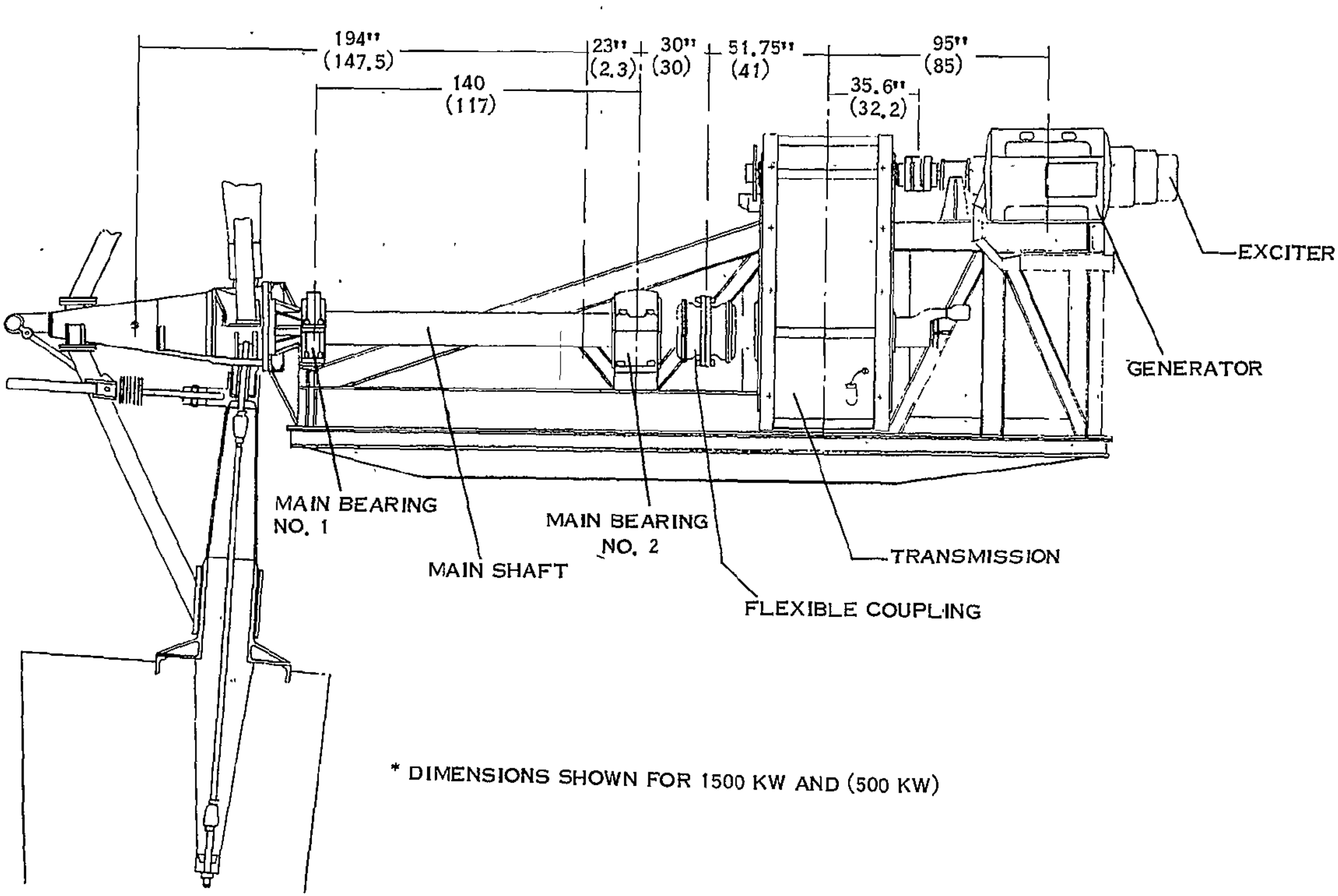


Figure 1-5. Preliminary Design Layout*

TABLE 1-1 PRELIMINARY DESIGN SYSTEM CHARACTERISTICS

POWER RATING, KWE	500	1500
ANNUAL ENERGY, KW-HRS	1.88×10^6	6.62×10^6
CAPACITY FACTOR	0.42	0.51
ROTOR DIAMETER, M(FT)	55.8 (183)	57.9 (190)
RPM	29	40
RATED VELOCITY, M/S (MPH)	7.27 (16.3)	10.1 (22.5)
λ RATED (λ DESIGN = 10)	9.0	9.5
TIP SPEED, M/S (FT/SEC)	84.7 (278)	121 (398)
CUT-IN VELOCITY, M/S (MPH)	3.54 (7.92)	5.11 (11.4)
λ CUT-IN	18.5	18.4
CUT-OUT VELOCITY, M/S (MPH)	17.9 (40)	22.3 (50)
λ CUT-OUT	3.67	4.18
HOURS ABOVE VCI	6257	6568
HOURS AT RATED	2067	2718

Both truss and concrete towers have been designed for the 500 KW and 1500 KW WTG's; these four towers are shown schematically in Figure 5-60. The truss tower is attractive primarily on the basis of low cost and cost predictability in a variety of locations and soil conditions. The concrete tower offers even lower cost for accessible sites having good soil conditions; in addition, this tower concept is generally more aesthetically pleasing to the majority of people. An important aspect of the tower design involves potential dynamic interactions with the rotor system. By avoiding resonant frequencies between the rotor and tower, load magnifications can be reduced, and a less expensive rotor/tower can be safely utilized. This subject is discussed in Section 5.2.6 and in Appendix 8.2. As previously cited, a prime aspect of the design philosophy was to maximize the amount of energy capture in order to minimize power generation costs. The first step in maximizing the energy capture is to utilize the design optimization code to select the optimum RPM and rated velocity ratio for the desired site and power level; this process also involves the selection of the rated wind velocity and blade diameter. The effect of RPM, velocity ratio and power level upon energy capture and energy generation cost is discussed in Section 4.0 and illustrated in Figures 4-3 through 4-6.

Following this procedure, it is necessary to design the control system such that the WTG:

- Synchronizes with the network at the lowest possible wind velocity
- Maintains electrical stability over the entire operating range under steady state and gusting wind conditions
- Allows cut-out at the intended design wind velocity
- Allows resynchronization at intermediate and high steady state wind conditions
- Places the blades in a safe orientation in anticipation of high winds

In addition to maximizing the WTG energy capture, the control system assists in the reduction of subsystem capital costs by operating the WTG in such a manner as to avoid excessive loads on the blades and tower. Under normal operating conditions where wind gusts are experienced, the control system will provide for the proper blade pitch angle to minimize torque variations on the system. During startup the control system will avoid prolonged operation at resonance frequencies. Under storm conditions where the wind velocity exceeds cut-out conditions, the control system will place the blades in a feathered 3-9 o'clock position. In situations where the wind direction is varying at a high rate, the control system will initiate rotation to minimize azimuth overturning moments due to the wind inflow angle. These are some examples of the manner in which the control system can be designed to assist in the reduction of loads on the system.

The costs for the two Preliminary Design WTG's, based on a 100 unit production run are \$935/KW and \$430/KW for the 500 KW/12 MPH site and 1500 KW/18 MPH site designs, respectively.

SECTION 2.0

STUDY GROUND RULES AND REQUIREMENTS

SECTION 2.0

STUDY GROUND RULES AND REQUIREMENTS

This section defines the design guidelines and requirements for the development of the wind driven electric generator system. The section is separated into two requirement areas; System Design and Subsystem Design requirements.

2.1 SYSTEM DESIGN REQUIREMENTS

2.1.1 POWER APPLICATION

The electrical power generated shall be compatible with and regulated to the requirements of existing public utility networks. The high power system is intended for tie-in to such networks. The low power system may be connected to existing networks or have separate loads dictated by its application.

The eventual application of large, cost effective, wind generator systems may be as multi-unit farms located in favorable wind locales supplying power to existing public utility transmission systems. Connections to such systems will require suitable switchgear, transformers, and transmission lines. Design of the high power system shall accommodate the application requirements.

2.1.2 UTILITY REQUIREMENTS

With regard to connecting the WTG to an electric utility network, the following requirements should be met.

1. Voltage - 60 cycle alternating current with 3 to 5% permissible voltage dip
2. Power Factor - power factor for the distribution system is typically 0.98, correction should be made on site
3. Line Reclosing - An automatic reclosing of distribution system circuit breakers will occur for temporary faults
4. Telephone Interference - correct for harmonic voltage to avoid telephone interference
5. Stability - insure stability of operation of the power generating equipment over the full range of environments to avoid voltage dips and outages

2.1.3 GENERATION COSTS

The goal of the designs shall be minimum cost per KWH. These costs shall include capital, cost of capital, amortization of capital over lifetime of equipment, taxes, operation and maintenance costs as a minimum.

2.1.4 OPERATION

The units shall be designed for unattended, failsafe automatic operation including startup, shutdown, and power regulation over the full range of wind operation, as well as manual control.

2.1.5 AVAILABILITY

The units 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.

2.1.6 DESIGN LIFE

All static components, including the tower shall be designed for a service life of 50 years. All dynamic components shall be designed for a service life of 30 years and may include periodic maintenance and replacement if cost effective. The effect of design life on cost shall be determined in the parametric study.

2.1.7 PARTS AND COMPONENTS

The designs shall utilize the latest design, material and fabrication technology insofar as its use minimizes electric power generation costs. When used, the technology shall have a base of proven experience. Unproven cost effective components or approaches shall be recommended to NASA for further investigation.

2.1.8 MAINTENANCE AND SERVICEABILITY

The designs shall provide for safe and easy maintenance wherever possible, including platforms, stairs, removable covers or shrouds, etc. All parts and components shall be designed for easy handling and lifting using available field equipment.

2.1.9 ASSEMBLY

The designs shall provide for a maximum of shop assembly and a minimum of field assembly prior to erection.

2.1.10 TRANSPORTABILITY AND ERECTION

The designs shall give consideration to transportation via existing surface vehicles and ease of field assembly and erection.

2.1.11 APPEARANCE

The designs shall give consideration to architectural aesthetics and public acceptance.

2.1.12 ENVIRONMENTAL

The units shall be designed to withstand the range of atmospheric environments experienced from New England to Alaska or the Caribbean area to hot desert climes.

The unit must therefore be capable of operation in snow, rain, lightning, hail, icing conditions, salt water vapors, wind-blown sand and dust and in temperature extremes of -51°C (-60°F) to 49°C (120°F). If cost effective, designs adaptable to local severe conditions with minimum change will be acceptable.

2.1.13 WIND DATA

To determine the power and/or energy extracted from the wind the velocity duration curves shown in Figure 2-1 should be used. The data shown is characterized by the median velocity wind corresponding to exactly 1/2 of the year (8760 hrs.), and is the velocity for a site as measured at 30 ft. above ground level.

The velocity at height H other than H_o = 30 ft. is given by the following relationship:

$$V = V_o \; (H/H_o)^n$$

where V = velocity at height H

V_o = velocity at height H_o = 30 ft.

n = exponent, assume to be 1/6

2.1.14 ENERGY STORAGE

It is not required that energy storage be included in this study.

2.2 SUBSYSTEM DESIGN REQUIREMENTS

2.2.1 ROTOR

The rotor shall be a propeller type with the axis of rotation being horizontal. The number of blades, blade shape (planform, twist, airfoil shape), rotor location (upwind or downwind of tower), size (diameter), blade coning to reduce stresses, controllable blade pitch and blade life were determined in the study.

The rotor shall be designed to withstand the following wind-loading conditions:

a. Maximum steady wind speed of 120 mph at 30 feet above ground.

b. Gusts shall be per NASA gust model as in Appendix 8.3.

c. Blade unloading caused by tower effect.

d. Blade loading caused by wind shear, wind inflow, and gyroscopic loadings.

The design shall provide for locking of the rotor when the WTG is in the shut-down condition.

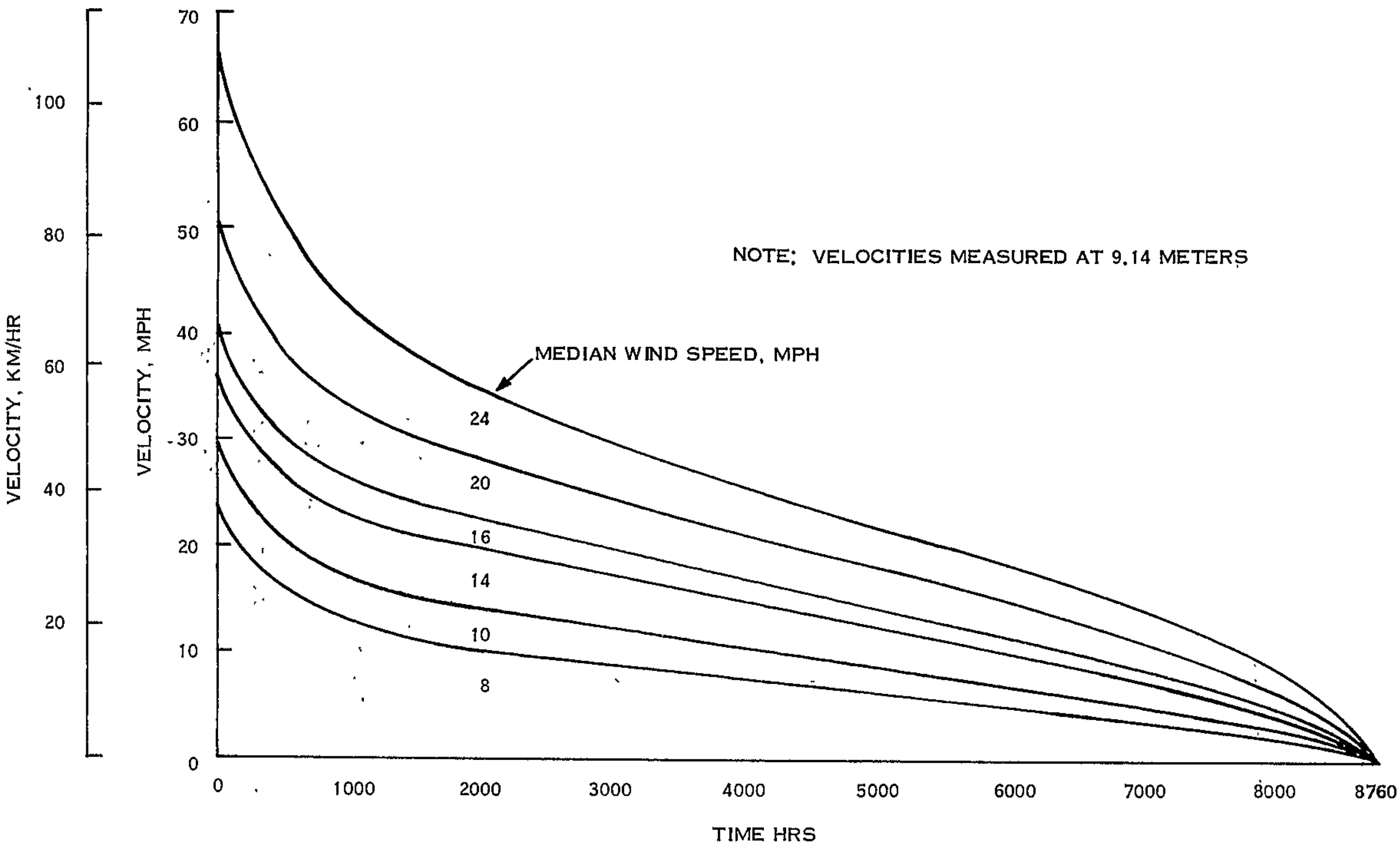


Figure 2-1. Assumed Velocity Duration Curves

2.2.2 TRANSMISSION

The power transmission shall transmit the torque developed by the rotor to the generator at the proper rotational speed. The method of transmitting the rotor torque, the coupling to the electrical generator and locking the rotor is optional.

The contractor shall study various methods and recommend an approach consistent with the general design requirements of the system.

2.2.3 ELECTRIC POWER GENERATING EQUIPMENT

The electrical generating equipment shall produce electric power at a suitable voltage and frequency for tie-in to existing public utility power lines and shall be determined by the contractor. All other aspects of the electric power generating equipment are optional and shall be optimized to meet the goal of minimum cost per kWH.

2.2.4 TOWER

The tower shall be designed for the same wind loading conditions specified for the rotor, the forces imposed by the rotor, the weight of all equipment located atop the tower and all varying loads which may lead to fatigue.

2.2.5 CONTROL SYSTEM

The wind-generator control system must perform three major functions, namely, the orientation of the rotor to face the wind, to control production of electric power over a wide range of wind velocity, including startup and shutdown, and to safeguard the wind generator from damage due to abnormal conditions. The control system shall meet the following requirements in performing these functions.

a. Rotor Orientation - The control system shall point the upwind side of the area swept out by the propeller blades into the wind for all operating and weather conditions. The orientation mechanism shall therefore be capable of 360° pointing. The accuracy of the orientation controls and the rate at which wind directional changes are followed shall be determined in the study. The control system should respond to wind directional changes averaged over a time period of not less than 10 seconds (a longer period may be recommended by the contractor) and be insensitive to fluctuations occurring over a lesser time period. The instrumentation and mechanism to accomplish orientation are optional.

b. Electric Power Control - The control system shall provide for startup, regulation of electric power over a wide range of wind speeds and shutdown of the system, either unattended or manually.

(1) Startup - Unattended startup of the wind generator system shall occur at a cut-in wind speed which is below the rated wind speed for which rated power is achieved. The cut-in wind speed shall be chosen such that useful power will be produced by the wind-generator system and alternating cut-out/cut-in operations will not occur for small variations of wind speed.

(2) Operating Range - The wind generator system shall produce electric power at a suitable voltage and frequency for tie-in to public utility power lines over its entire operating range. The operating range shall be defined by the cut-in wind speed for startup and extend to the cut-out wind speed for shutdown. The cut-out wind speed shall be selected by the contractor and shall be based on cost effectiveness.

The power produced over the operating range shall vary from part load at startup to rated power at rated wind speeds and continue at rated power to cut out wind speed. This requirement shall not apply in the event conceptual design results in a cost effective system capable of safely producing more than rated power above rated wind speed.

The method of accomplishing frequency and power-level control is optional. The contractor may utilize controllable blade pitch, fixed pitch with flaps or a system which converts all available mechanical energy into electrical and then tailors electrical outout to suit the application.

(3) Shutdown - The control system shall shut down the wind generator system when wind speed exceeds cut-out speed or is below cut-in speed. It will probably be necessary to feather or fold the blades on shutdown in order to prevent their damage due to wind speeds up to 120 MPH.

(4) Electrical Load - The control system shall connect the electrical load (tie-in to public utility power lines) whenever the wind-generator system is capable of producing electric power and shall remove the electrical load whenever the wind velocity is below cut-in value, above cut-out value or the demand exceeds the capability to supply.

c. Protective Controls - The control system shall also protect the wind-generator system against damage due to abnormal operating conditions, including excessive wind speeds, overspeeding, overloading, failure of a critical component, etc. Any abnormality which could lead to substantial damage to the wind generator shall be sensed and result in shutdown of the system.

SECTION 3.0

CONCEPTUAL DESIGN

SECTION 3.0

CONCEPTUAL DESIGN

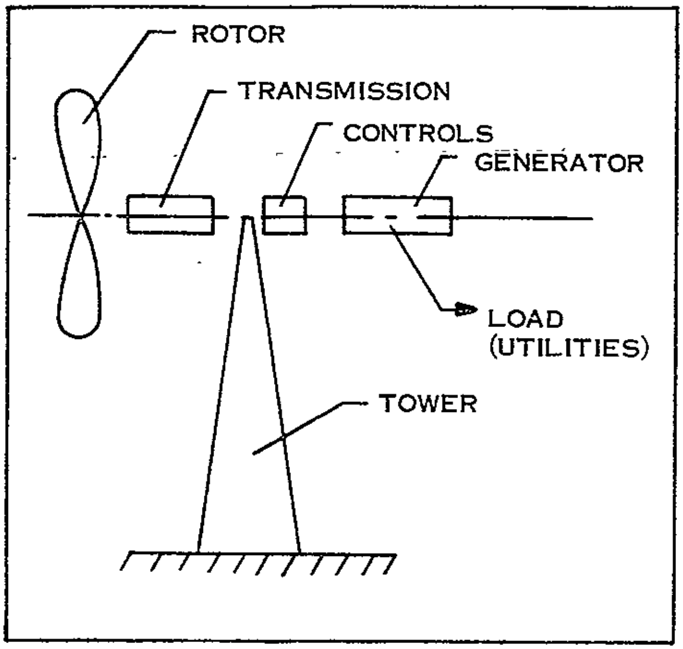
This portion of the study focused upon evaluating the various alternative concepts which could be used in the WTG System. The objective of the task was to select those concepts which offered low power generation costs and high system reliability for eventual use in the Preliminary Designs.

The conversion of wind energy into electrical energy requires the following types of components: a turbine, transmission, generator and tower. The wind turbine rotor converts the kinetic energy in the wind into useful shaft rotational power. The classical relationship between wind velocity, (V), turbine diameter (D) and shaft power (P) is given by $P = KD^2V^3_w$ where the constant (K) depends on the system of units used. Once the wind energy is converted to rotating shaft power, the shaft speed N_r (rpm) is made compatible with the electrical generating equipment via the power transmission. The electrical generator then converts the transmission output shaft power to useful electrical output to, in this case, the electrical utility network. A control system is also required to regulate and protect the system. The general configuration of the components is seen in Figure 3-1 with a tower which provides the necessary support and strength to accommodate the induced loads and also elevate the rotor axis up to heights that provide minimum wind gradients and allow for adequate clearance between the ground and turbine blade tip. The specific details of each of the above components and their interaction and effect on other system components will be explained in subsequent sections.

One of the most important system concepts that was identified and which has the most impact on other components is the system operational mode.

3.1 SYSTEM OPERATIONAL MODE

There are two primary methods of operating a wind turbine rotor; a constant speed (RPM) rotor or a constant velocity ratio (λ) rotor. Understanding the difference between these two modes of turbine rotor operation is necessary to an understanding of the effect these operating modes have on the rest of the wind turbine generator system. The difference between these two modes of operation may be understood by considering how wind energy is converted into shaft torque. The torque force is created in the system from the vector summation of the lift and drag components of the airfoil section used in the turbine rotor blade. The product of the turbine rotor speed (RPM) and the torque force results in shaft power output; the relationship between power output, turbine rotor blade geometry, operating rpm and wind velocity is summarized in Figure 3-2. Recalling the power equation $P = KD^2V^3$ an examination of the "constant" K will show that it is the product of unit conversion constants and component efficiencies given by $K = kC_p$ The term C_p is in reality the turbine blade aerodynamic efficiency; Figure 3-2 shows how this efficiency varies as a function of a dimensionless parameter which is the ratio of turbine rotor blade tip velocity to the wind speed.



MAJOR OPTIONS CONSIDERED INCLUDE

- OPERATING MODE
- ROTOR TYPE
- POWER REGULATION TECHNIQUE
- SYSTEM CONFIGURATION
- COMPONENTS
  - ROTOR
  - TRANSMISSION
  - GENERATOR
  - CONTROLS
  - TOWER

Figure 3-1. System Concept Considerations

Once the rotor geometry (size and airfoil section) is defined and the blade pitch (angle of attack of the airfoil) is established, the rotor may be operated in the two different modes. First consider constant rpm mode. Figure 3-3 shows how λ varies with wind speed. This variation in the velocity ratio results in a continual variation in the power coefficient C_p. Thus, a "constant rpm" operating mode has a varying velocity ratio - λ . This system extracts power with a varying C_p and requires parametric analysis to determine the optimum rpm. The "constant velocity ratio" mode operates with a constant λ usually corresponding to the maximum C_p, and has an rpm variation as shown in Figure 3-4. This system operates at a constant value of C_p and can extract more energy, however, as later sections will show it is not a desirable system.

With either mode of operation it is necessary to control the power into the rotor shaft. Two methods to regulate power into the rotor shaft are available: change the azimuthal orientation of the WTG (YAW) or change the configuration of the wind turbine generator.

In the first approach the relative position of the rotor axis with respect to the wind vector could be changed thereby producing power with only a component of the wind. This change in azimuth for power regulation produces large inflow moments (See Section 5.2.5) and other design problems in terms of turning rates and is not compatible with the design cost goals. Another method of regulation is to cone the turbine blades toward the rotor axis (downwind), see Figure 3-5, which has the effect of reducing the aerodynamic diameter and thereby reducing the power into the shaft. This method requires complex equipment design and in addition, previous designs have experienced dynamic problems.

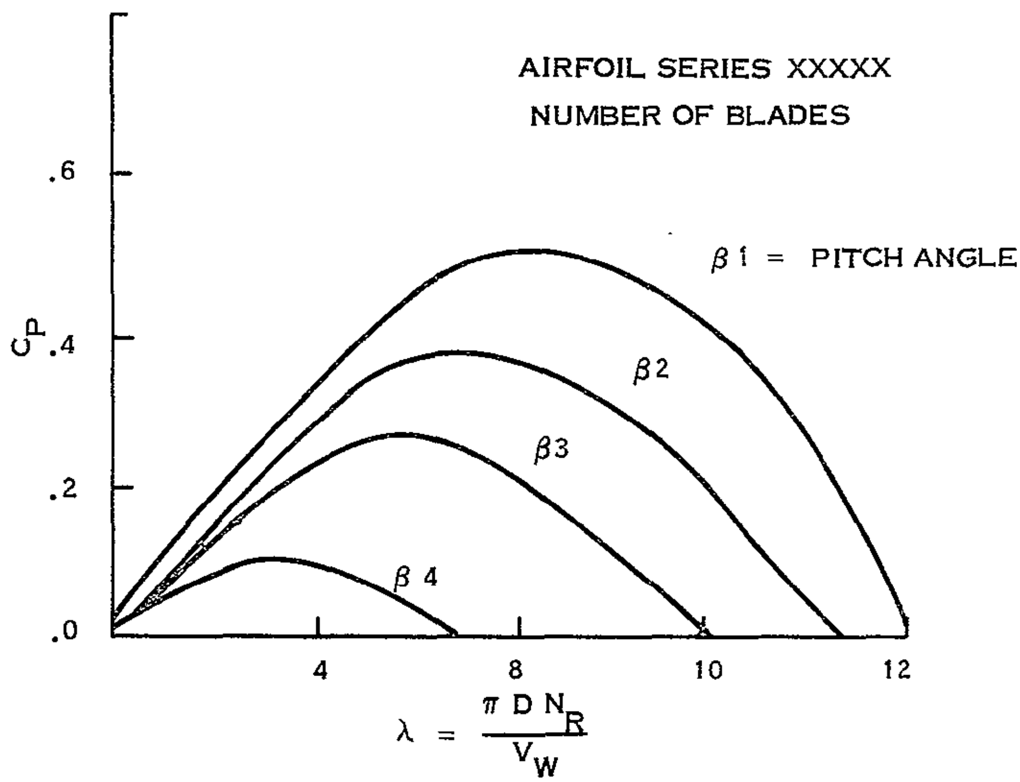


Figure 3-2. Relation Between Power Coefficient and Velocity Ratio

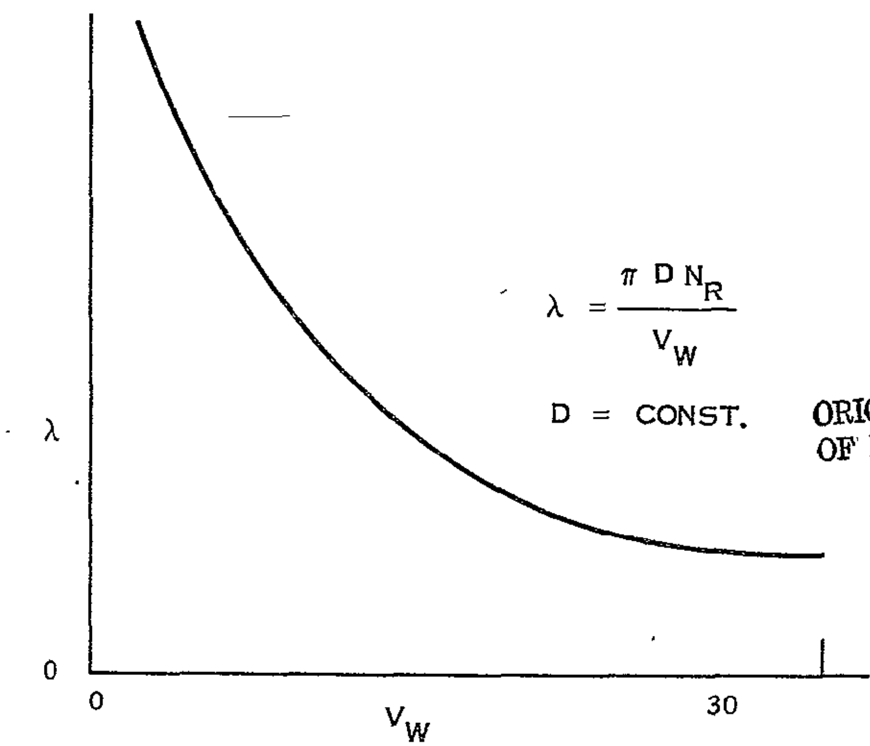


Figure 3-3. Variation of Velocity Ratio with Wind Velocity - Constant RPM System

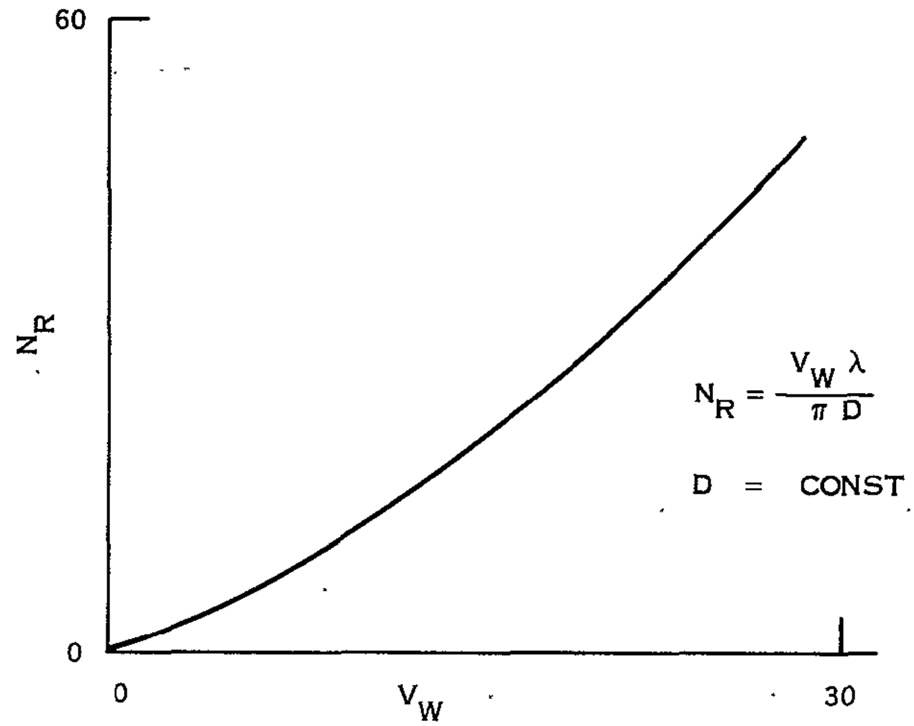


Figure 3-4. Variation of RPM with Wind Velocity

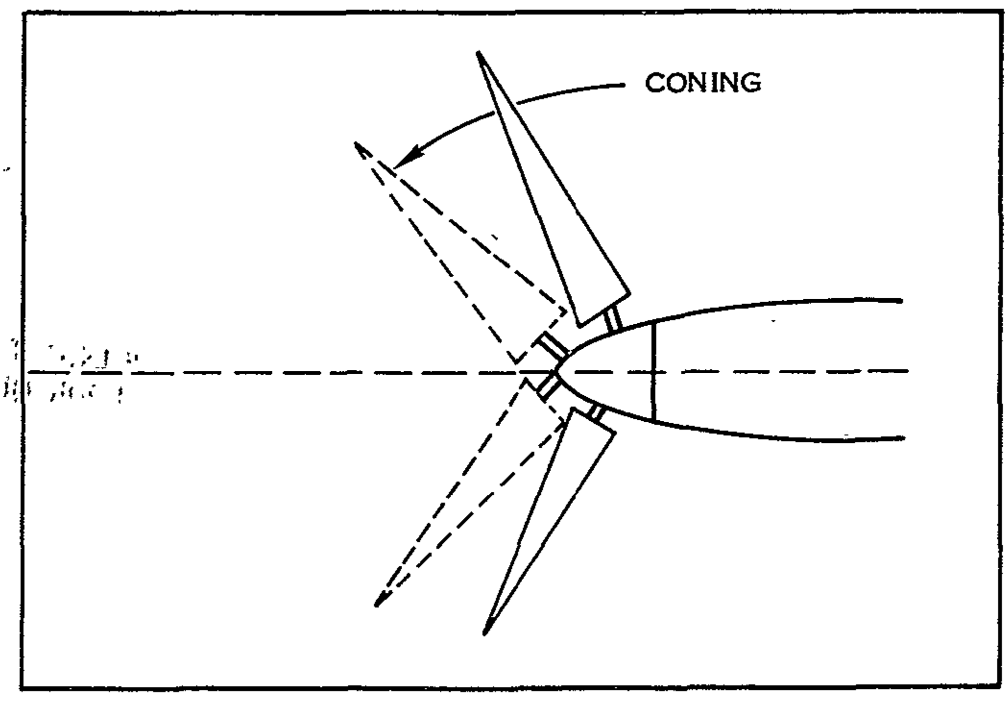


Figure 3-5. Coning for Power Regulation

Other methods of power regulation examined were blade flaps and tip spoilers. However, these configurations and others examined complicate the design of the rotor blade and do not provide an adequate solution to the breakaway problem.

A more direct method is power regulation by changing the pitch of the blades. Figures 3-6 and 3-7 show the results of a performance analysis which considered the effects of operating mode and method of power regulation on blade thrust loads. From these figures it is evident that variable pitch, constant rpm operation is the most desirable method of the approaches considered since the peak blade thrust loads are less.

Table 3-1 shows component combinations necessary to provide a constant voltage and frequency output for a wind turbine generator system. As shown in the table, a constant λ rotor subsystem requires the incorporation of either a variable speed transmission or a variable speed generator. Based on the rotor performance analysis results and the cost and performance results of the system components (transmission and generator), a constant rpm operating mode with a variable blade pitch for power regulation was selected for all subsequent study efforts.

TABLE 3-1. CANDIDATE OPERATING MODES FOR CONSTANT VOLTAGE AND FREQUENCY OUTPUT

OPERATING MODE		POWER REGULATION	TRANSMISSION	GENERATOR	
CONSTANT RPM ROTOR	FIXED PITCH	YAW, CONING	FIXED RATIO	CONSTANT RPM	
	VARIABLE PITCH	BLADE PITCH CHANGE	FIXED RATIO	CONSTANT RPM	RECOMMENDED APPROACH
CONSTANT VELOCITY RATIO ROTOR (RPM α V)	FIXED PITCH	YAW, CONING	VARIABLE SPEED	CONSTANT RPM	
			FIXED RATIO	VARIABLE SPEED	
	VARIABLE PITCH	BLADE PITCH CHANGE	VARIABLE SPEED	CONSTANT RPM	
			FIXED RATIO	VARIABLE SPEED	

- UNACCEPTABLE THRUST LOADS ON ROTOR WITH FIXED PITCH OR CONSTANT VELOCITY RATIO ROTOR
- VARIABLE SPEED TRANSMISSIONS OR GENERATORS ARE NOT COST COMPETITIVE
- SLIGHTLY LESS ENERGY WITH CONSTANT RPM ROTOR (~ 3%)

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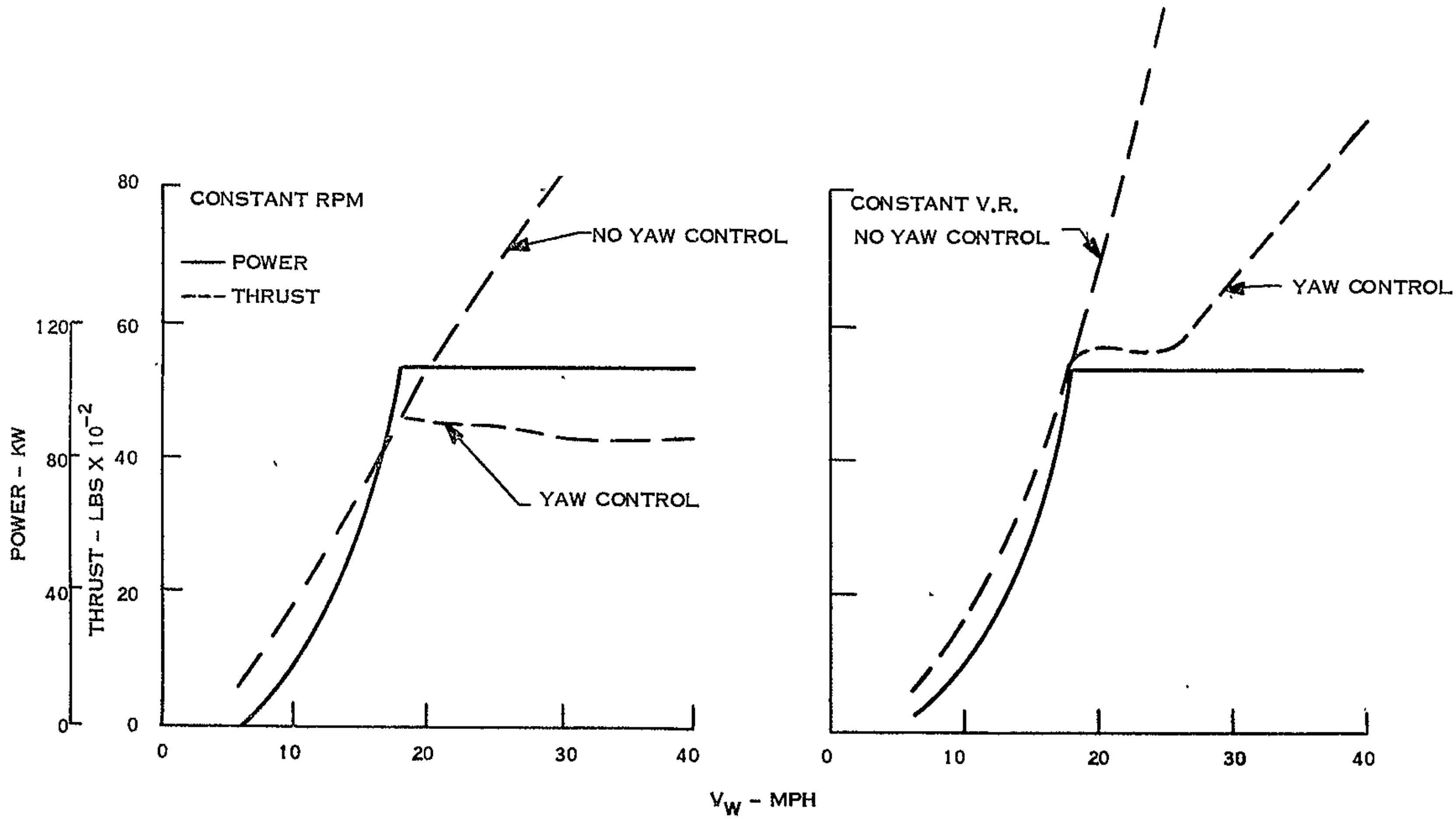
3.2 SYSTEM CONFIGURATIONS

This section describes the results of three trade-offs which were examined concerning the overall WTG configuration. On the basis of the work performed it was decided to use one rotor per tower, place the rotor behind the tower, and to situate the transmission and generator atop the tower.

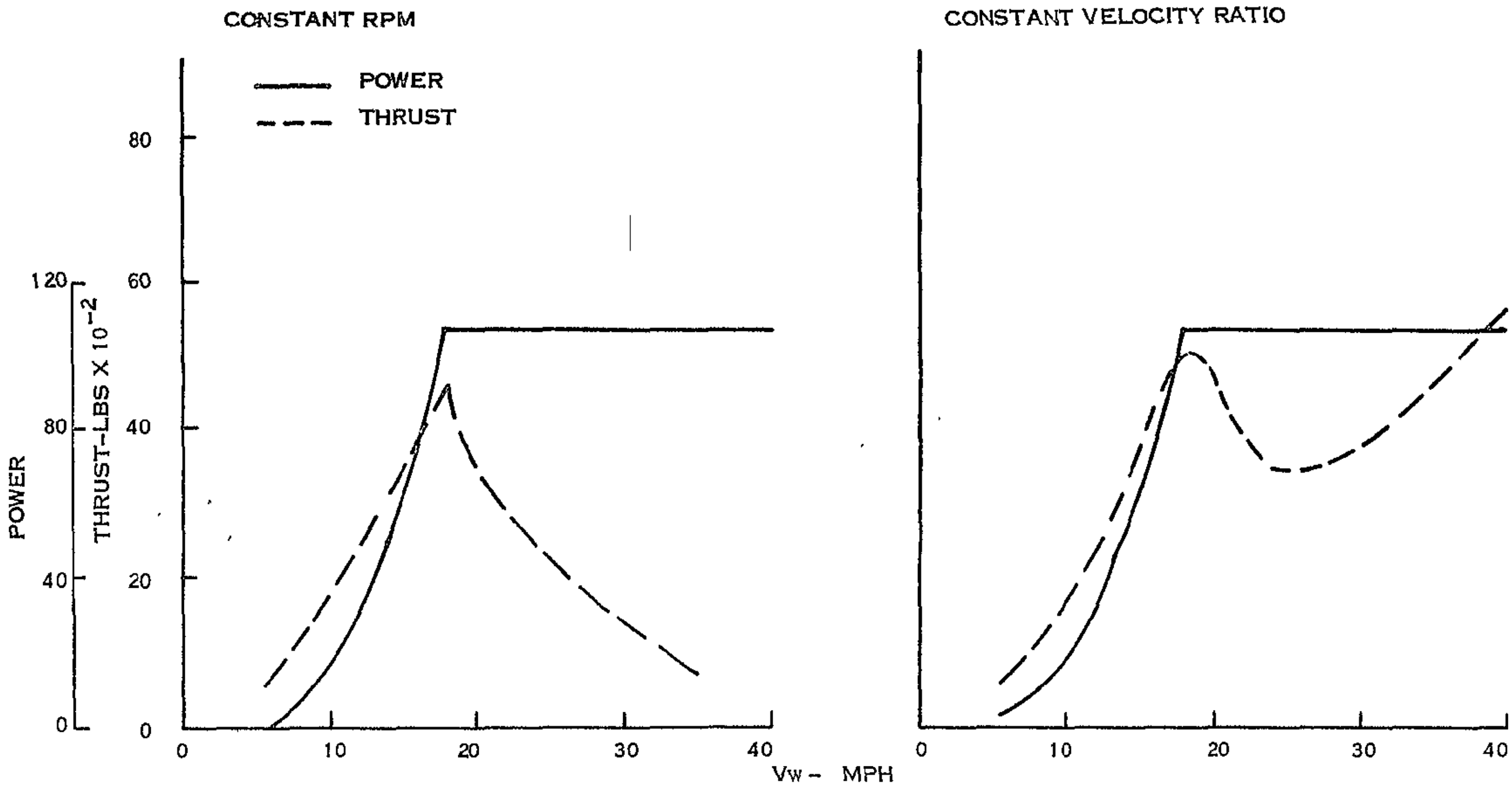
3.2.1 MULTIPLE ROTORS

One of the cost trade-offs that was performed as part of the conceptual design analyses was between multiple and single rotor configurations of equal power



- WITH CONSTANT VELOCITY OPERATION YAW CONTROL RESULTS IN HIGH THRUST LOADS
- WITH CONSTANT RPM YAW CONTROL IS REQUIRED TO MINIMIZE THRUST LOADS

Figure 3-6. Thrust Loads Under Fixed Pitch Operation



VARIABLE PITCH OPERATION RESULTS IN
LOW THRUST LOADS

Figure 3-7. Thrust Loads Under Variable Pitch Operation output. To perform this analysis the following assumptions were applied to each configuration:

- Equal power output
- Identical velocity duration curves
- Equivalent rotor axis height above ground
- Equal quality and material costs for similar components

For equal power generation the diameter of each rotor in a multiple rotor system is given by $D_n = \dfrac{D\ \text{Single Rotor}}{(\text{Number of Rotors})^{0.5}}$

and the thrust by $T_n = Ts/N$. To compare multiple systems with single systems, the cost of the following three configurations were estimated.

1. 3, 33.3 KWe WTG's; a separate tower for each WTG
2. 1, 100 KWe WTG
3. 3, 33.3 KWe rotors on one tower driving one generator through one transmission via 3 mechanical drive inputs.

The cost analysis showed that configuration (1) would cost \$136,700 while configurations (2) and (3) would cost \$89,600 and \$142,100 respectively. The conclusion was that multiple rotor configurations are not as cost effective as are single rotor configurations.

The principal reason for this conclusion is that it is less expensive to build 1 large rotor as opposed to 2 or 3 smaller rotors having the equivalent total power. This fact is supported by the estimated system cost breakdowns shown below in Table 3-2 for the 3 cases considered.

TABLE 3-2. EFFECT OF MULTIPLE ROTOR SYSTEMS ON COST

SYSTEM	ROTOR COST	GENERATOR COST	TRANSMISSION COST	TOWER COST	TOTAL COST, \$
1	102,000	5,400	12,000	17,300	136,700
2	65,000	3,600	11,000	10,000	89,600
3	102,000	3,600	15,000	21,500	142,100

3.2.2 ROTOR LOCATION

Another system configuration trade-off was performed between locating the rotor upwind in front of the tower or downwind behind the tower. The conditions considered in choosing the most cost effective or best engineering approach were:

tower-rotor blade interference, rotor overhang, tower shadowing and the effect of weather vaning.

Tower-rotor blade interference is of great concern since the blades always deflect in the same direction as the wind. Thus, placing the blades upwind of the tower would require larger compensations such as: overhang distances, coning angles and rotor shaft tilt. Each of these compensations cause increases in weight and/or decreases in performance. In addition, the upwind location is inherently unstable, in that changes in wind direction tend to move the rotor out of the wind. On the other hand, a downwind location is inherently stable in that wind direction changes induce a moment which tends to align the rotor with the wind. Further, a downwind location has less rotor overhang, requires less blade coning and less rotor axis tilt to achieve a well balanced economical design configuration.

The only qualitative reason for considering placement of the rotor upwind is to reduce the nacelle and tower flow blockage effects. Flow blockage by the nacelle has been determined to be insignificant for the designs studied. The effect of tower shadowing, however, is more complicated. For truss towers, tower shadowing was found to be an insignificant blade design consideration as compared to the effects of wind shear. For concrete towers, however, tower shadowing may affect the design of the rotor/tower subsystems. Since this situation is not amenable to analytical treatment it could not be pursued within the scope of this study. For a more detailed discussion of tower shadowing, see Section 5.2.5.

Due to the advantages of reducing the rotor overhang requirements and the inherent stability of this configuration the downwind location was selected. The upwind rotor location was not considered further since:

- The tower shadowing effect was judged to be unimportant for the selected truss tower and wind shear conditions.

- Placing the rotor upwind would not eliminate the tower shadowing effect although it may reduce it

Furthermore, if concrete towers were found to present a tower shadowing problem with respect to a downwind rotor location, changes in the rotor/tower design (such as a teetered hub) could provide a solution.

3.2.3 EQUIPMENT LOCATION

The choice of equipment location was examined during the conceptual design phase of the study to determine the cost and system benefits that could result from locating the transmission and generator on the ground. Some of the considerations for locating equipment on the ground are:

1. Higher tower natural frequencies resulting from reduced weight aloft

2. Reduced installation costs

3. Reduced maintenance costs

Basic configurations were developed for locating equipment on the ground for two conditions: transmitting low speed/high torque shaft power to the ground or high speed/low torque shaft power to the ground.

The results of this study are summarized as follows:

Transmitting high speed/low torque shaft power to the ground (transmission on top, generator on ground) produces no cost advantage over transmitting electrical power to the ground. This results from the following facts:

- Since the transmission is over 50% of the nacelle weight it must be placed on the ground to achieve a tower natural frequency advantage.

- Major reductions in installation costs would necessitate placement of both the transmission and generator on the ground because of the high transmission weight.

- The transmission and generator maintenance costs are anticipated to be small (total system maintenance costs are less than 10% of the annual WTG revenue requirement) the transmission is expected to require more frequent inspections and maintenance than the generator (see Section 5.3.2.5).

2. Transmitting low speed/high torque shaft power to the ground (transmission and generator on ground) is very expensive.

   - For a 100 KW system the cost of 1:1 bevel gear set was $25,000; alternatively, a 42:1 worm gear set was costed at $21,000. By comparison a top mounted parallel transmission shaft is about $4,000. At higher power levels the situation is aggravated since lower RPM's (due to blade tip speed considerations) would result in an acceleration of the required torque levels.

   - The tower cost was calculated to be a function primarily of the system power level-less than $5,000 in tower cost could be saved by placing the transmission and generator on the ground.

   - The anticipated savings in installation and maintenance costs was judged to be minor as compared to the cost of the high torque, 100-150 foot long, transmission shaft and bevel gear set.

In addition, universal joints were investigated as a possible method of transmitting low speed mechanical shaft power to the ground after a suggestion from Prof. Ulrich Hutter, University of Stuggart, Germany. A check with manufacturing firms in this country revealed that universal joints for the 100 k system operating at 30-40 rpm with 24000 ft.-lbs. of transmitted torque, and operative at angles greater than 6° are not standard and not available.

Based on the above results, a recommendation was made for top mounted equipment.

3.3 ROTORS

3.3.1 ROTOR TYPES

This study was limited to the "horizontal axis, propeller type rotor system." Although the study demonstrated obvious contrasts between helicopter and propeller blade technology applications, the helicopter-type blading was considered as a propeller type rotor system and was included in the conceptual design studies. The propeller and helicopter type blades differ basically in method of construction, in planform and in aerodynamic loading and balancing. Both performance and cost trade-offs favored propeller type blading rather than helicopter type blading.

The comparison between the helicopter and propeller blade was made on the basis of the following parameters:

- Cost
- Performance
- Reliability
- Development Risk
- Versatility

In examining the difference in cost, a number of helicopter rotor manufacturers were contacted. The consensus of opinion was that helicopter rotors would tend to cost more since additional weight and fabrication procedures would be required to achieve balancing ahead of the quarter chord point. The weight added to the quarter chord region would result in a tendency to reduce the weight of the afterbody section so as to minimize total rotor weight while obtaining the desired balancing. The light afterbody section would be more susceptible to damage and could incur higher maintenance costs than for the propeller approach.

Since the propeller concept makes more efficient use of the material weight by utilizing it for structure, as opposed to balance considerations, it was judged that capital and/or maintenance cost advantages were realized with this approach.

In terms of performance the helicopter and propeller would appear to offer the same aerodynamic efficiencies. However, other system considerations favor the use of the propeller approach. In general, the propeller is a stiffer structure in the flapwise as well as torsional direction and can be expected to offer smaller blade deflections. This is important in the consideration of tower overhang requirements and the effective diameter offered by the rotor under various operating conditions. In addition, the lower weight achievable with the propeller provides relief to the remaining system structural elements.

The reliability of these concepts cannot be addressed in quantitative terms since insufficient data is available for the propeller concept and the helicopter approach has not been utilized in a WTG application.

Due to the fact that previous WTG's have employed propeller design technology (Hutters' and Argand's machines) the development risk associated with this approach is assessed to be lower than with the helicopter concept. In addition, Hamilton Standard has developed sophisticated analytical techniques which permit a thorough understanding of propeller dynamics in the WTG operational configuration. Over a quarter of a million propellers have been produced over a 33 year period, including spar/shell construction using fiberglass, none of which were quarter chord balanced.

On the basis of the above, WTG's built with propeller technology will offer greater versatility in terms of power level and scope of applications. The potential to build propeller based rotor designs having diameters on the order of 200 feet provides the WTG with lower power generation costs than otherwise available. This factor should allow wind power systems in general to be competitive with a broad range of alternative energy sources.

Ducted and shrouded propeller type rotors were investigated, especially for multiple rotor applications. Although, shrouded and ducted rotors can produce from 50% to 100% more power and energy in a given wind, (Reference 3-2) a large diffuser structure is required in order to minimize aerodynamic losses. Figure 3-8 illustrates the effect of the duct length to height ratio on the observed velocity at the duct throat to the theoretically obtainable velocity. Since energy is proportional to the velocity cubed, a modest velocity loss can lead to significant energy loss. As shown in the figure a length to diameter ratio of about 12 is optimum, reduction of the length to diameter ratio to a value of 5 results in a 19% energy loss in the stream. Therefore, the size of the duct would have to be approximately 500 to 1000 feet long for the large systems being considered in order not to induce excessive aerodynamic losses. Problems associated with supporting such structures and changing azimuth position eliminated this concept from further consideration.

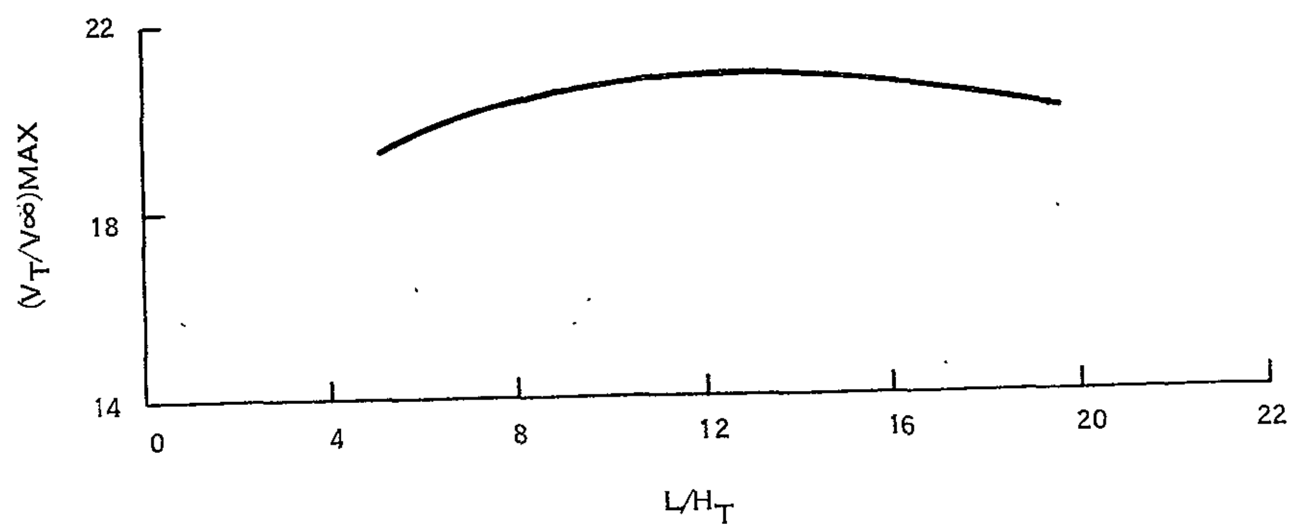


Figure 3-8. Effect of Diffuser Length to Height Ratio on Observed Stream Velocity

3.3.2 NUMBER OF BLADES

Selection of the number of blades per rotor was resolved basically by the economics of the system. One, two, three and four bladed rotors were considered in the conceptual design studies. Power characteristics for the number of blades per rotor are shown in Figure 3-9 which indicates there is not a significant power coefficient performance advantage with more than two blades. It is also noted in Figure 3-9 that the theoretical performance of the one-bladed rotor has not been accurately determined. The primary tradeoff with system economics is the aspect of system dynamics and control, which is relieved with an increasing number of blades. Therefore, with the dynamics and control problems of one-bladed rotors and the surplus blade costs of three and four bladed rotors, the two bladed rotor was selected for the preliminary design studies.

3.3.3 BLADE AIRFOIL SELECTIONS

Selection of the blade airfoil was based on a number of factors, including the existence of well proven aerodynamic performance characteristics, the appropriate high lift-to-drag ratios over a wide range in lift coefficients, and the susceptibility of the airfoil to structural design and fabrication capabilities. A number of airfoils were reviewed, as shown in Figure 3-10.

The selection of airfoil type is strongly influenced by operating lift coefficient and design Reynolds number. Although performance improves with high values of L/D, the corresponding high values of operating C_L rapidly reduce solidity below practical limits. Thus the selection of the airfoil type must be based on the maximum L/D consistent with practical blade geometries and performance levels. As previously pointed out, cost considerations of the step-up gear lead to high values of velocity ratio which require low Activity Factor blades. Thus from Figures 3-11 and 3-12, an operating lift coefficient of 1.0 appears to be a good choice for two bladed wind turbines. For higher blade numbers the operating C_L would need to be reduced. A single blade wind turbine appears to be required for an operating C_L of 1.50. As indicated in Figure 3-10, conventional airfoils like the NACA 230XX weries, the NACA 44XX and the NACA 6 Series show L/D in the 100-200 range at an operating C_L of 1.0. Advanced airfoils like the Wortmann design show L/D above 150 at an operating C_L of 1.50, whereas sail type airfoil data indicate unacceptably low L/D's (below 20).

The NACA 23000 Series airfoil was tentatively selected primarily due to low development risk and minimum development cost. Also, this airfoil has a reliably high lift-to-drag ratio, 120, at an operating lift coefficient of 1.0, for appropriate design Reynolds numbers of about 3×10^6, with the large two bladed rotor systems.

Other high performance airfoil selections such as the WORTMANN and LIEBECK Series, with lift-to-drag ratios above 150 at operating lift coefficients of 1.5 were also included in the parametric analysis phase to evaluate performance and cost trade-offs in cost effectiveness studies. With the limited experience for these airfoils and their reported sensitivity to surface conditions as well as the limitation on velocity ratio at the required high operating lift coefficient, it is concluded that these airfoils should not be used for first-generation wind rotors.

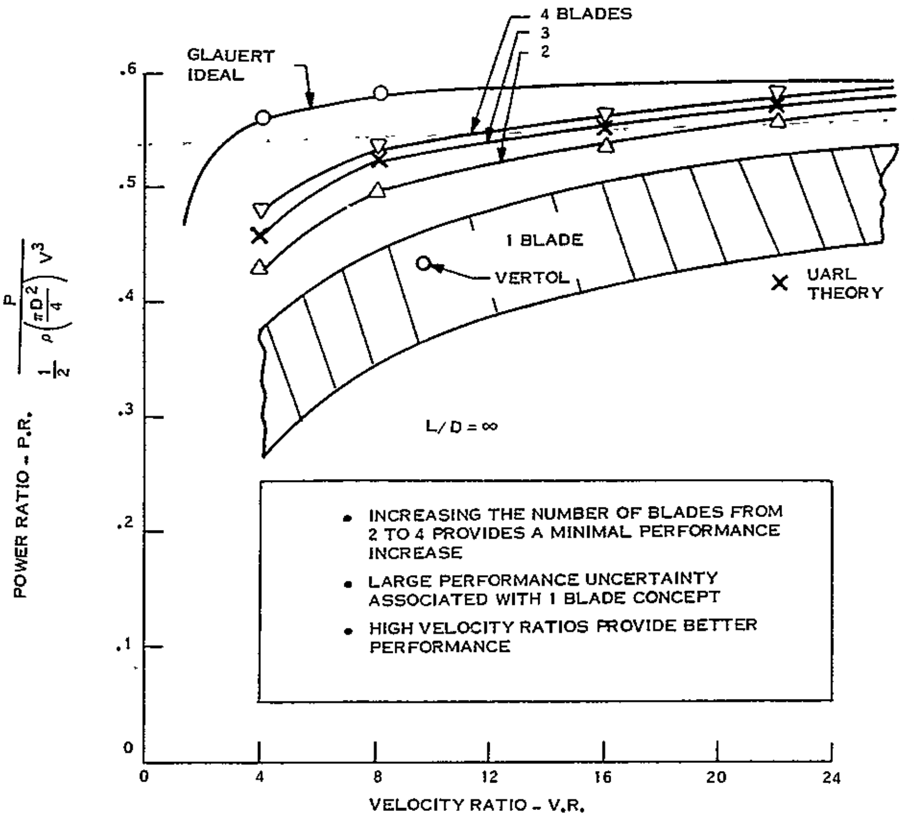


Figure 3-9. Effect of Number of Blades on Rotor Performance

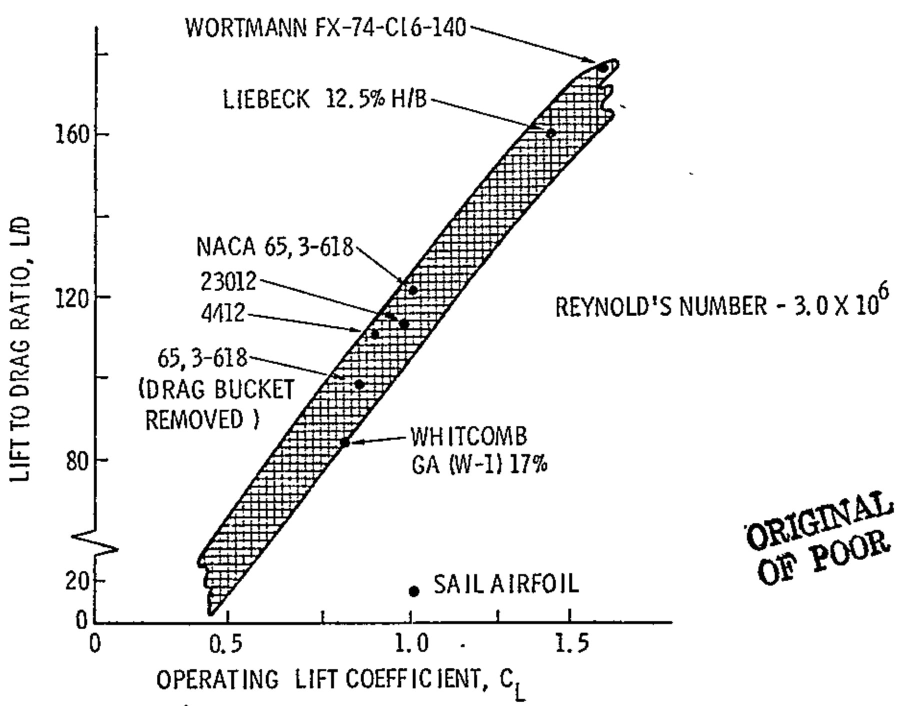


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Figure 3-10. General Trend of Lift-to-Drag Ratio with Operating Lift Coefficient for Various Thickness

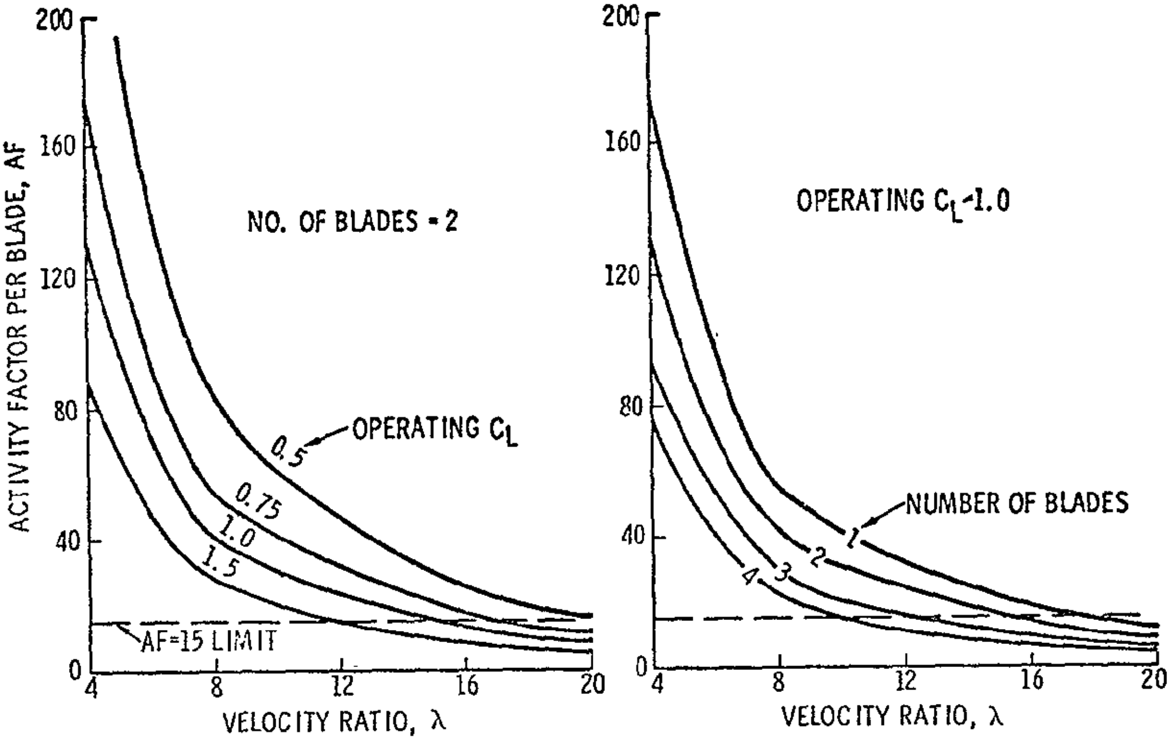

Figure 3-11. Effect of Velocity Ratio on Activity Factor

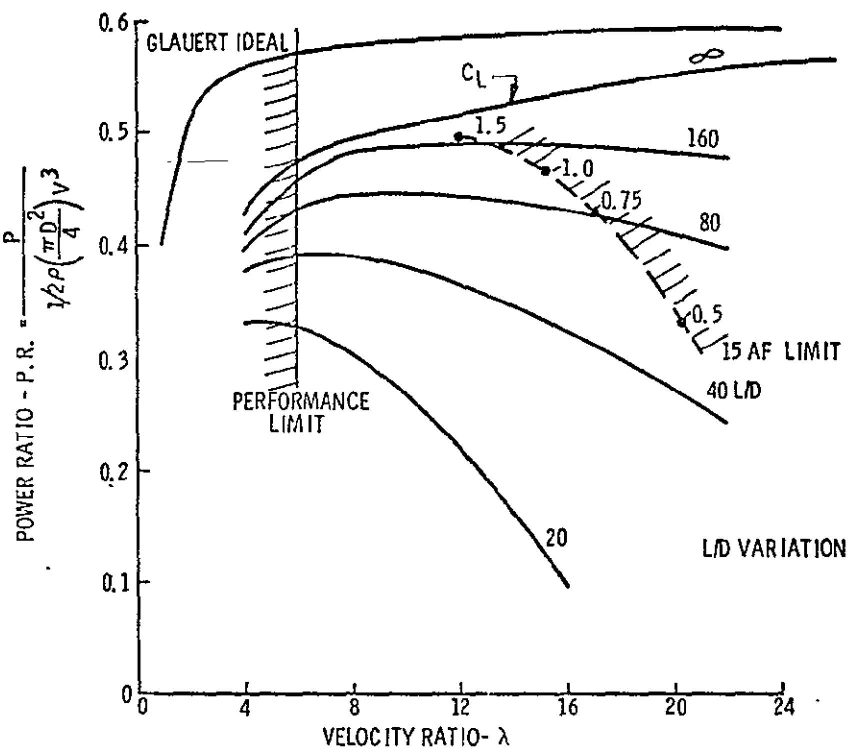

Figure 3-12. Performance and Activity Factor Limits on Performance Spectrum, 2-Bladed Wind Turbines

3.3.4 OPTIMUM BLADING SELECTION

This section discusses the selection of the blade geometry which results in the best overall WTG design.

The optimum blade configuration was determined by considering the effects on blade aerodynamic efficiency, design velocity ratio, lift-to-drag ratio, operating lift coefficient, activity factor (area-weighted solidity ratio), blade planform and taper, blade thickness ratio, blade twist, and blade root cut-out. In establishing the most cost effective blade configuration, it is also necessary to consider the structural and fabrication design aspects as well as the optimum aerodynamic design. On the basis of initial design analyses, it appears that no significant compromise of the aerodynamically optimum configuration is required.

The design velocity ratio combines the two most important variables affecting the rotor design, the diameter and the wind velocity, as well as the most important drive train variable, the shaft RPM. Both aerodynamic performance, including power, and geometric blading parameters are expressed in terms of velocity ratio. The considerations of system size and cost effectiveness tend to require high design velocity ratios, which in turn dictate low activity factors and high lift-to-drag ratios for two bladed rotors as shown in Figure 3-11.

The optimum design velocity ratio is limited on the high end by a structural design limit on activity factor as shown in Figure 3-12. Also shown in Figure 3-12 is how the blade airfoil selection, as defined by the lift-to-drag ratio (L/D) and the operating lift coefficient, affects the optimum design velocity ratio and peak power ratio.

An example of how the activity factor and design velocity ratio affect rotor RPM selection and annual energy capture is shown in Figure 3-12 for a given rotor size, a given wind duration curve, and a given rated wind speed. The significance of Figure 3-13 is that for any given activity factor blade and any characteristic wind environment, there is an optimum rotor RPM in terms of energy capture. From a systems standpoint, higher activity factor permits efficient operation at higher rotor RPM and results in lower transmission cost as shown in Figure 3-14 . It also demonstrates that for the given set of boundary conditions, such as the constants indicated, activity factor does not significantly affect maximum energy capture (as long as velocity ratio is changed accordingly), although it represents very significant variations in blade geometry. The tapered blade geometry which is characteristic of a low activity factor offers lighter blades resulting in lower material and fabrication costs.

Optimum blade planform and taper is derived from the activity factor, the airfoil, and the airfoil thickness ratio. The effects of changes in the blade airfoil thickness ratio is shown in Figure 3-15, in which the performance indicated is for a blade with constant thickness ratio along the blade radius. In practice, the thickness ratio varies with the blade radius, being greater at the inboard end because of structural reasons and because the effects of increasing thickness is much less significant on the aerodynamic performance at the inboard sections than at the tip. The thickness ratio (and therefore the planform

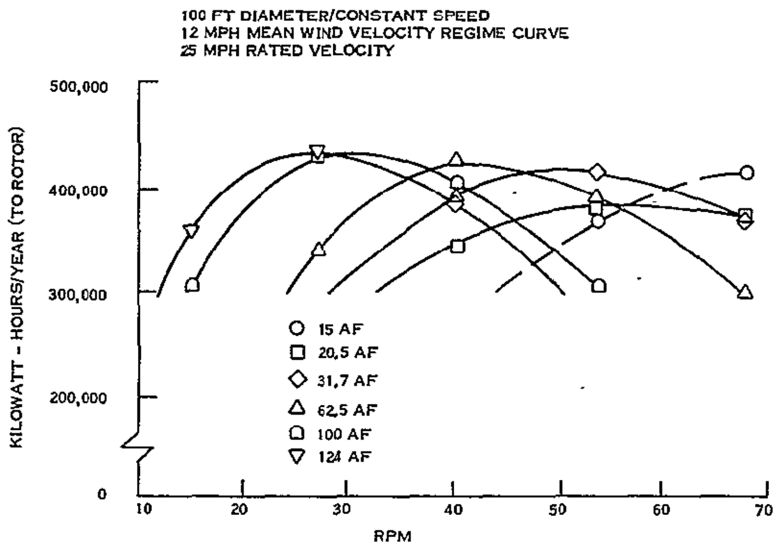

Figure 3-13. Effect of Activity Factor and Rated Velocity Ratio Upon Optimum RPM

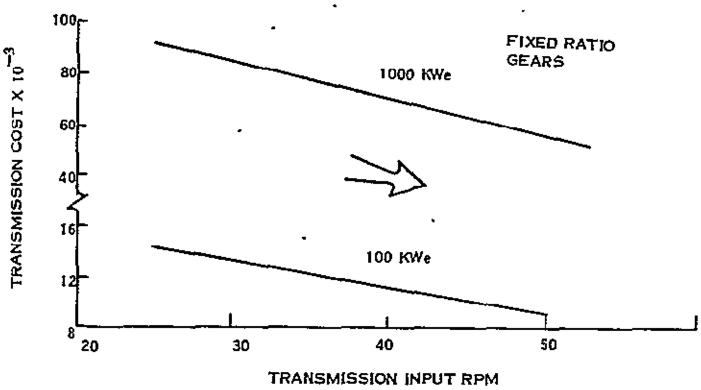

Figure 3-14. Effect of Transmission Input RPM on Transmission Cost

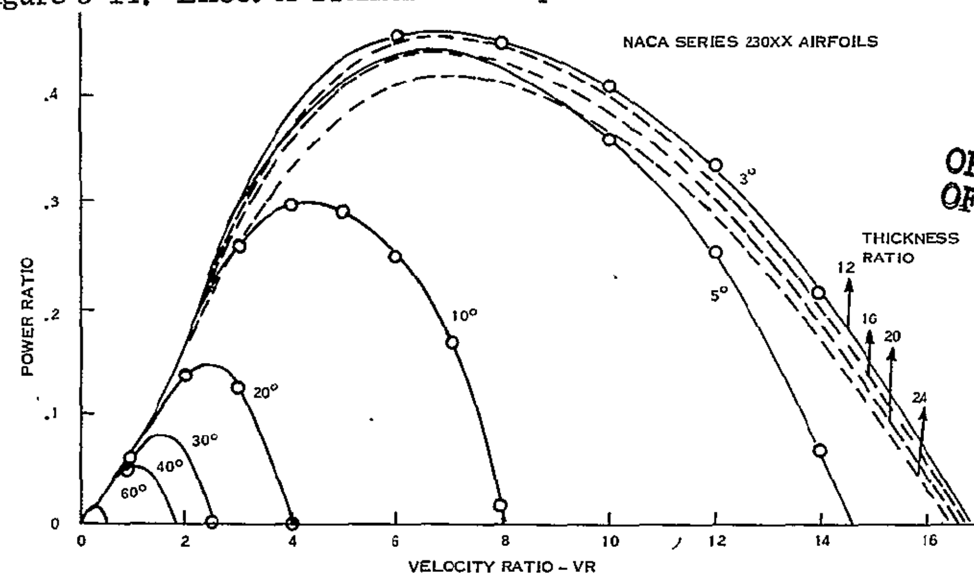

Figure 3-15. Effect of Thickness Ratio on Performance

and taper) is a powerful parameter in the blade structural design in reducing blade weight and cost and must be carefully traded off to produce the most cost effective design. It is noted here that the thickness ratio was increased during the Preliminary Design, see Figure 5-39.

Another consideration in the conceptual design of the blades was the economic advantages of partial removal of inboard blade sections, which are not very effective aerodynamically. The performance degradation is shown in Figure 3-16 as a function of percent cut-off at the blade root and reflects both the effects of reduction in airfoil area and the increased drag of an exposed elliptical blade spar. Analyses indicated that any economic advantages of shortening the blade airfoil sections is offset by the performance degradation and that the concept is no more cost effective than a nominal 15% cut-off, which is about a practical minimum limit as required by the rotor hub.

The optimum planform, taper, and twist for blades with a 23012 series airfoil and an activity factor of 30 is resolved in Figure 3-17, which presents chord length, blade thickness, and blade twist as a function of percent of blade length. A total twist of about 15° is indicated from the root of the blade to the tip with a zero-twist reference at the 0.75 blade radius. A thickness ratio of 0.2 is maintained from 0.50 blade radius to the tip and increases to about 0.40 at the blade root. The chord length varies non-linearly from the blade root to blade tip.

In terms of the aerodynamically optimum blading, the various geometric options may be resolved based on economic considerations and conceptual design requirements. It is noted that the recommended aerodynamically optimum blade concept is not necessarily the most cost effective design, and additional tradeoffs with structural design and fabrication techniques are required throughout the rotor design development.

3.3.5 BLADE STRUCTURAL DESIGN/FABRICATION

Conceptual design studies on the blades included structural considerations for the basic load, frequency, and dynamic stability requirements, and the impact of these considerations on the aerodynamic blade design and on the fabrication techniques and materials. In order to satisfy the requirements for high reliability and long life, detailed analyses were performed to assess the effects (1) of limit, intermittent and continuous loads, (2) of flatwise, edgewise and torsional stiffness, exciting frequencies and bending modes, and (3) of mode coupling on dynamic stability (flutter) and on the need for blade balancing.

Manufacturing studies for the conceptual design included assessments (1) of realistic limits on aerodynamic blading parameters such as activity factor, rotor RPM, and rotor size, (2) of the fabrication techniques and materials, and (3) of the rotor blade construction costs as a function of size. The primary element in WTG manufacturing cost is the blades and, accordingly, a broad investigation was made of the blade concepts, materials, fabrication methods and facilities. The concepts given the most consideration were:

1. Metal Spar and shell construction
2. Metal Spar with ribs and aluminum skin

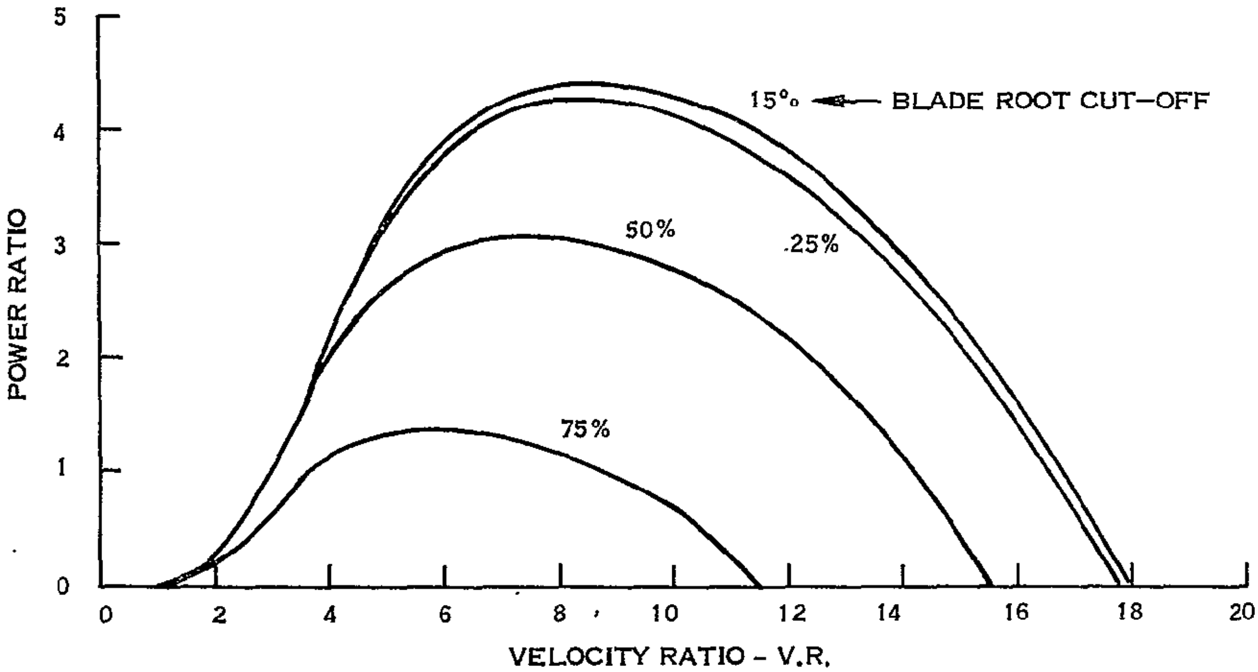

Figure 3-16. Effect of Blade Root Cutoff on Performance

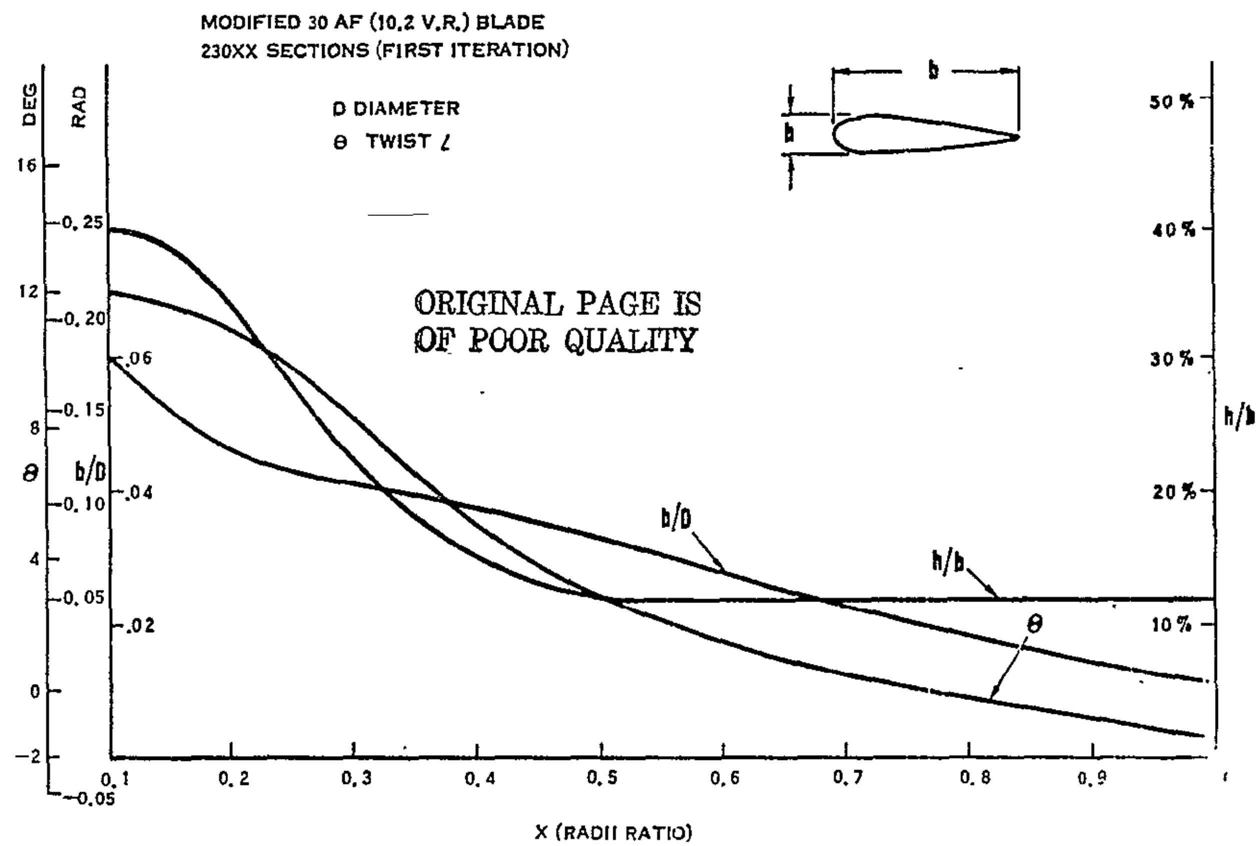

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Figure 3-17. Blade Shape, Conceptual Design

3. Extended aluminum blades
4. Metal Spar, foam and fiberglass skin
5. Molded fiberglass monocoque
6. Fiberglass spar and shell prepreg layup
7. Fiberglass monocoque layup
8. Filament wound spar and shell
9. Filament wound monocoque
10. Monocoque blade with rovings

In all but the lowest activity factor blades, the spar/shell blade construction was found the most efficient. For the low activity factors, the monocoque structural concept is also an attractive method of construction. With the fabrication of either spar/shell or monocoque designs, the filament winding process using fiberglass/epoxy materials was found to be more cost effective than the other shell fabrication techniques, because of the potential for automatic processing and relatively low tooling costs.

3.3.6 ROTOR HUB DESIGN CONCEPTS

Hub considerations that were included in the conceptual design studies were the variable pitch mechanism, the aspects of teetering and coning, and the hub fabrication methods. The selected concept for the hub assembly is shown in Figures 3-18 and 3-19. The basic hub structure as a weldment of simple steel forgings, standard commercial steel plate, and standard steel tubing. Forged hub structures were eliminated because of the high forging and machining costs. The pitch change mechanism concept is based entirely on the use of low-cost industrial catalogue components which meet the conceptual design requirements for high reliability, low fabrication/maintenance costs, and power only during pitch angle changes. Effects of hub teetering and blade coning to reduce structural requirements were reviewed during conceptual design studies. The aspect of a teetered hub was also included in analyses on the rotor-tower dynamics during the system preliminary design studies.

3.3.7 ROTOR RECOMMENDATIONS

The recommendations for the rotor design that evolved from the conceptual design studies are as follows:

- 2-bladed, propeller-type rotor
- NACA 23012 Series airfoil
- 30 Activity Factor Blade (.0365 Solidity)
- Design velocity ratio of 10
- Minimum blade inboard cut-off
- Spar/shell or monocoque construction
- Fiberglass/filament wound fabrication

3.4 TRANSMISSIONS

Wind turbine rotors are characteristically low rpm, high torque machines. Rpm values are typically 20 to 40 with torques varying between 20,000 and 400,000 ft. lbs. Generators, on the other hand, are high speed, low torque devices. It is a requirement of the power transmission system to be compatible with the

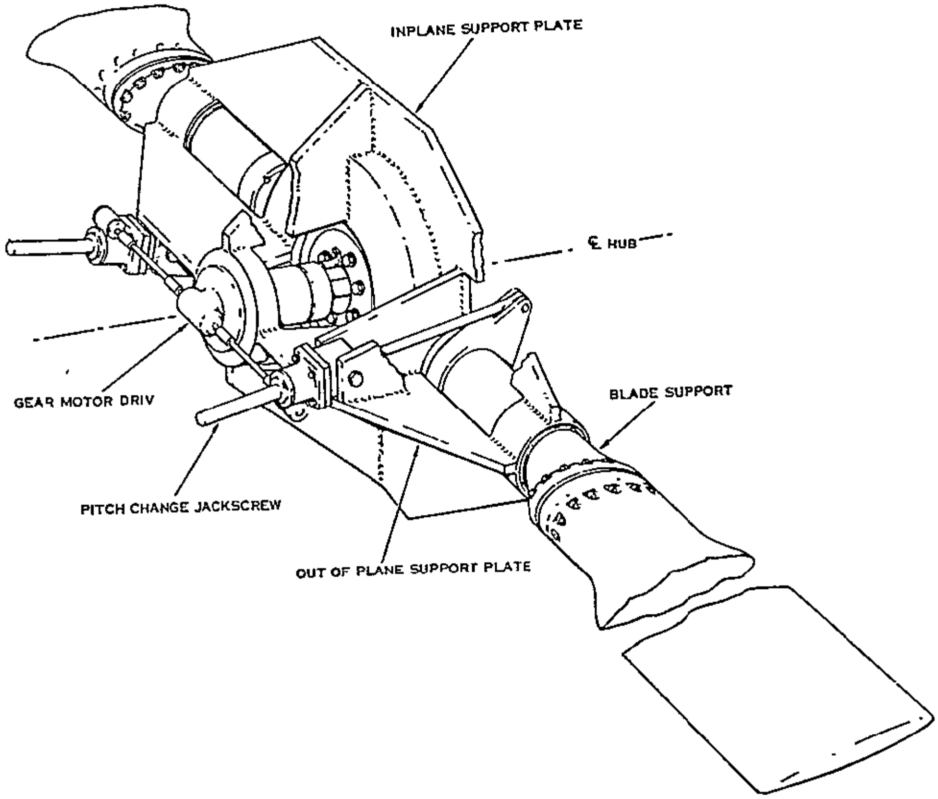

Figure 3-18. Conceptual Design Wind Turbine Hub

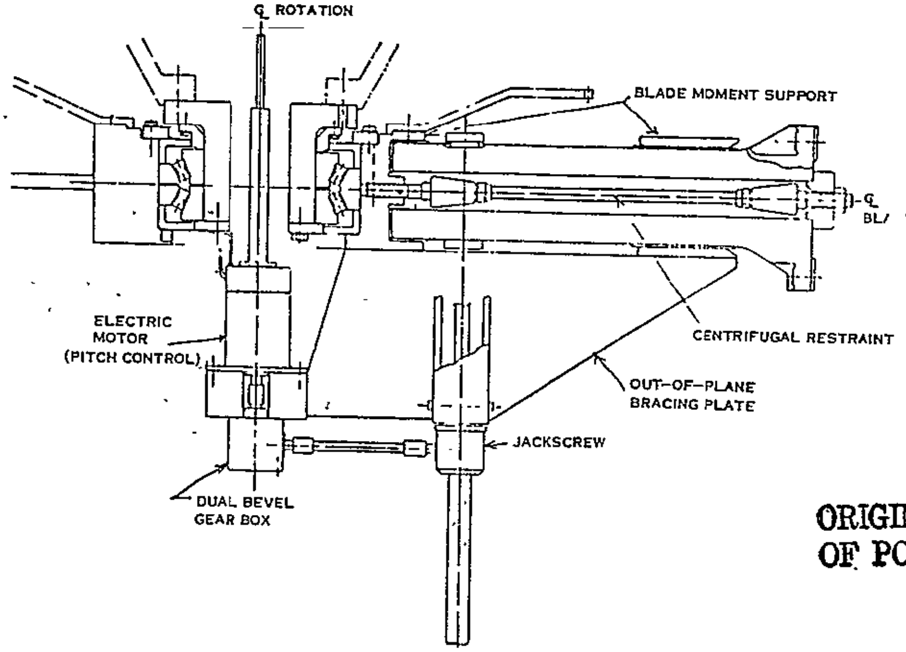

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Figure 3-19. Conceptual Design Hub Assembly Crossection operating characteristic of both the turbine rotor and generator.

Cost and performance of various transmission candidates were obtained to provide the necessary input to make the proper selection. This section will describe the various transmission candidates, their cost and pertinent operating characteristics. Table 3-3 shows the various transmission candidates that were consistent with the rotor and generator options.

3.4.1 FIXED RATIO SPEED INCREASERS

Fixed ratio gear speed increasers were examined because of their high efficiency (96% - 98%), high reliability and compactness of design; Figure 3-20 illustrates the size and appearance of a 1 MW unit. Most of these systems have their own lubrication system and are commercially available. Table 3-4 shows the variation of cost for fixed ratio gear speed increasers as a function of power level and rpm. It is important to note the cost savings associated with increases in rotor rpm. This is explained by the fact that rotor input torque is inversely proportional to rpm and speed increaser size is determined by torque level. It is also important to note that variation in speed increaser output rpm has very little effect on cost for a given input rpm. This fact was used in a generator-transmission cost study for constant rpm systems which cross plotted the cost vs. rpm for synchronous and induction generators and transmission. The results of this effort showed the most cost effective combination occurs at 1800 rpm.

Chain speed increasers were also examined because of their potential low cost. They are particularly suited for low speed, high torque application, however, they are larger and heavier than corresponding gear boxes.

TABLE 3-3. CANDIDATE TRANSMISSION CONCEPTS

<table>
<tr><th>ROTOR OPTION</th><th>GENERATOR OPTION</th><th colspan="3">TRANSMISSION CANDIDATES</th></tr>
<tr><td rowspan="5">CONSTANT RPM</td><td rowspan="5">CONSTANT RPM</td><td rowspan="5">FIXED RATIO</td><td rowspan="4">MECHANICAL</td><td>GEAR BOX</td></tr>
<tr><td>BELTS</td></tr>
<tr><td>CHAINS</td></tr>
<tr><td>COMBINATIONS</td></tr>
<tr><td>HYDRAULIC</td><td>FLUID PUMP & MOTOR</td></tr>
<tr><td rowspan="7">CONSTANT VELOCITY RATIO</td><td rowspan="3">CONSTANT RPM</td><td rowspan="3">VARIABLE SPEED</td><td rowspan="2">HYDRAULIC</td><td>FLUID PUMP & MOTOR</td></tr>
<tr><td>HYDRO-VISCOUS</td></tr>
<tr><td>ELECTRICAL</td><td>DC/DC/AC</td></tr>
<tr><td rowspan="4">VARIABLE RPM</td><td rowspan="4">FIXED RATIO</td><td rowspan="4">MECHANICAL</td><td>FIXED GEARS</td></tr>
<tr><td>BELTS</td></tr>
<tr><td>CHAINS</td></tr>
<tr><td>COMBINATIONS</td></tr>
</table>

TABLE 3-4. COST TREND OF FIXED RATIO GEAR BOXES (1800 RPM GENERATOR

POWER LEVEL	100 kW	500 kW	1000 kW
COST (LOW RPM ROTOR)	$14,000	$55,000	$≈ 85,000
COST (HIGH RPM ROTOR)	$ 8,000	$30,000	$≈ 57,000

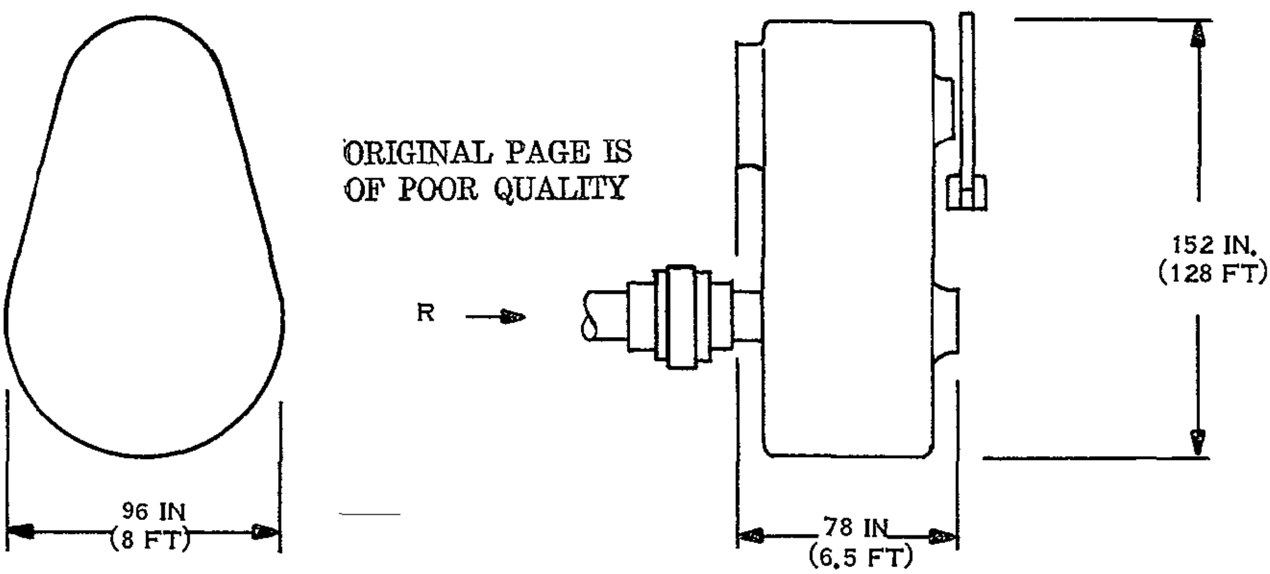

Figure 3-20. 1 MW Gearbox Unit

TABLE 3-5. COST TREND OF FIXED RATIO CHAIN TRANSMISSIONS (1800 RPM GENERATOR)

POWER LEVEL	100 KW	250 KW	1000 KW
COST (LOW RPM ROTOR)	$10,410	$20,680	$112,360
COST (HIGH RPM ROTOR)	$ 7,150	$12,120	$ 78,100

Chain speed increasers are custom manufactured and designed and as such, will require special bearing support housings and lubrication systems. Table 3-5 shows the cost of chain speed increasers as a function of power level and rpm while Figure 3-21 gives a typical geometry for one proposed chain system.

Combinations of chains and gear boxes were also examined for the conceptual design task. These systems were studied because of the potential for lower cost than gear boxes above and higher reliability than chains. Typical cost and geometry are shown in Table 3-6 and Figure 3-22.

TABLE 3-6. COST TREND OF FIXED RATIO GEAR BOX WITH CHAINS (1800 RPM GENERATOR)

POWER LEVEL	100 KW	250 KW	1000 KW
COST (LOW RPM ROTOR)	\$9,120	\$18,320	\$77,740
COST (HIGH RPM ROTOR)	\$5,860	\$ 9,750	\$43,470

Another speed increaser system considered was the combination of chains with belts. Chains and belts seemed a natural combination to produce a low cost speed increaser. Belts can be used at the high speed low torque side of the speed increaser and have many desirable features. Belts are inexpensive but are short lived and would, therefore, have to be periodically replaced. Belts can absorb torque shocks and through the use of an idler can be designed to offer preset torque limiting potential. Belts also provide electrical insulation and can allow for mounting misalignments. They are, however, quite large and require bearing supports and enclosures which make them more attractive for lower power systems.

TABLE 3-7. COST TREND OF CHAIN AND BELT TRANSMISSION (1800 RPM GENERATOR)

POWER LEVEL	100 KW	250 KW	1000 KW
COST (LOW RPM ROTOR)	\$ 6,790	\$ 16,310	\$ 71,510
COST (HIGH RPM ROTOR)	\$ 3,520	\$ 7,750	\$ 37,240

Table 3-7 shows the cost of a chain and belt speed increaser as a function of power level and rpm. As with most systems, there is a marked decrease in cost as rotor rpm is increased. Figure 3-23 shows the typical geometry for a chain and belt system, take particular note to the dimensions.

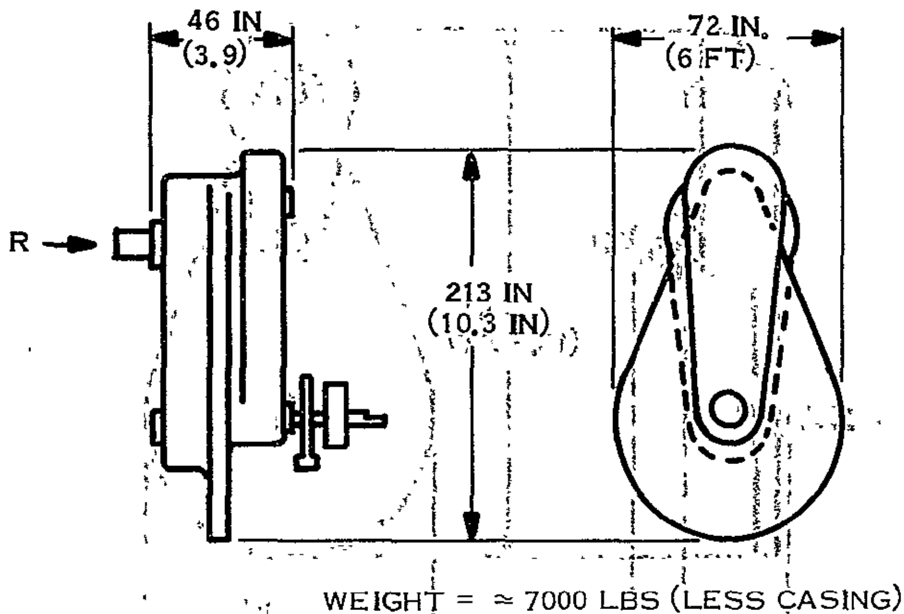

Figure 3-21. Fixed Ratio Chain Transmission Concept
(100 kW, Low RPM Rotor, 1800 RPM Generator)

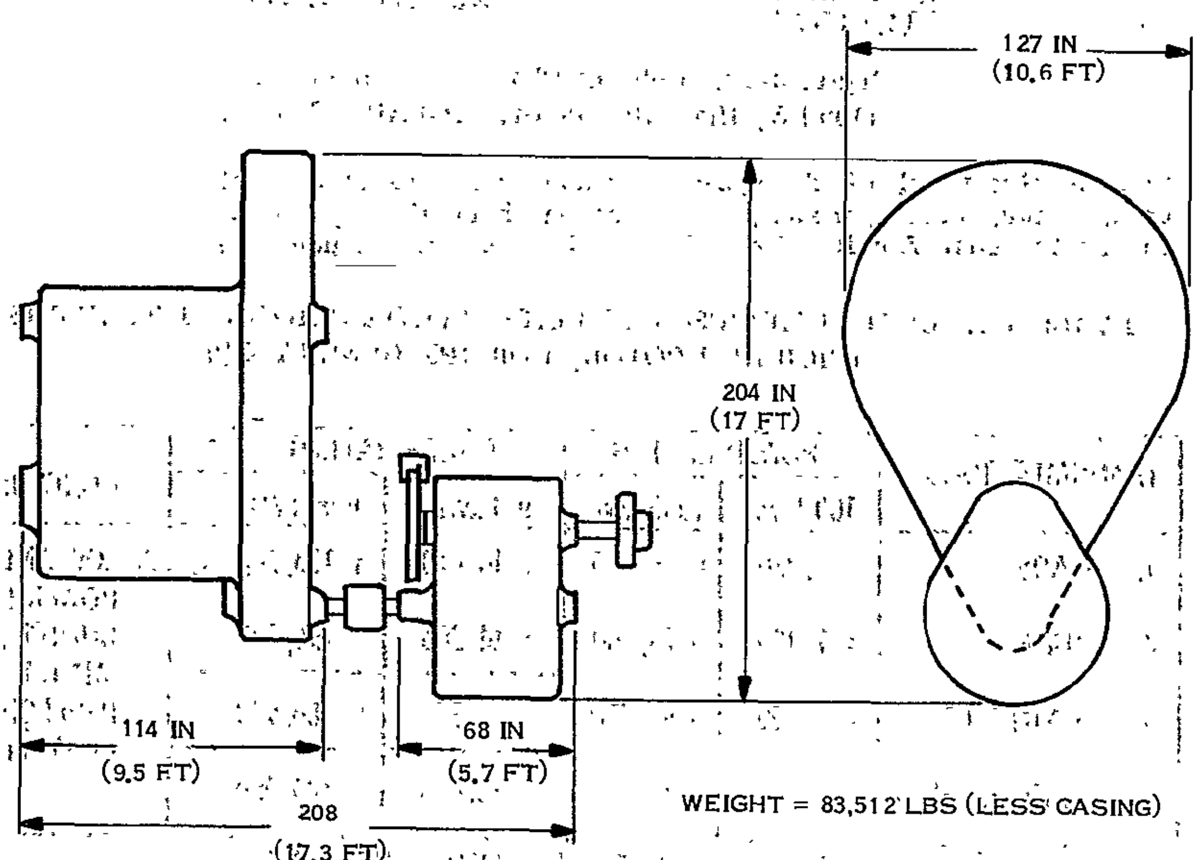

Figure 3-22. Fixed Ratio Gear Box with Chains Concept
(1000 kW, High Velocity Ratio, 1800 RPM Generator)

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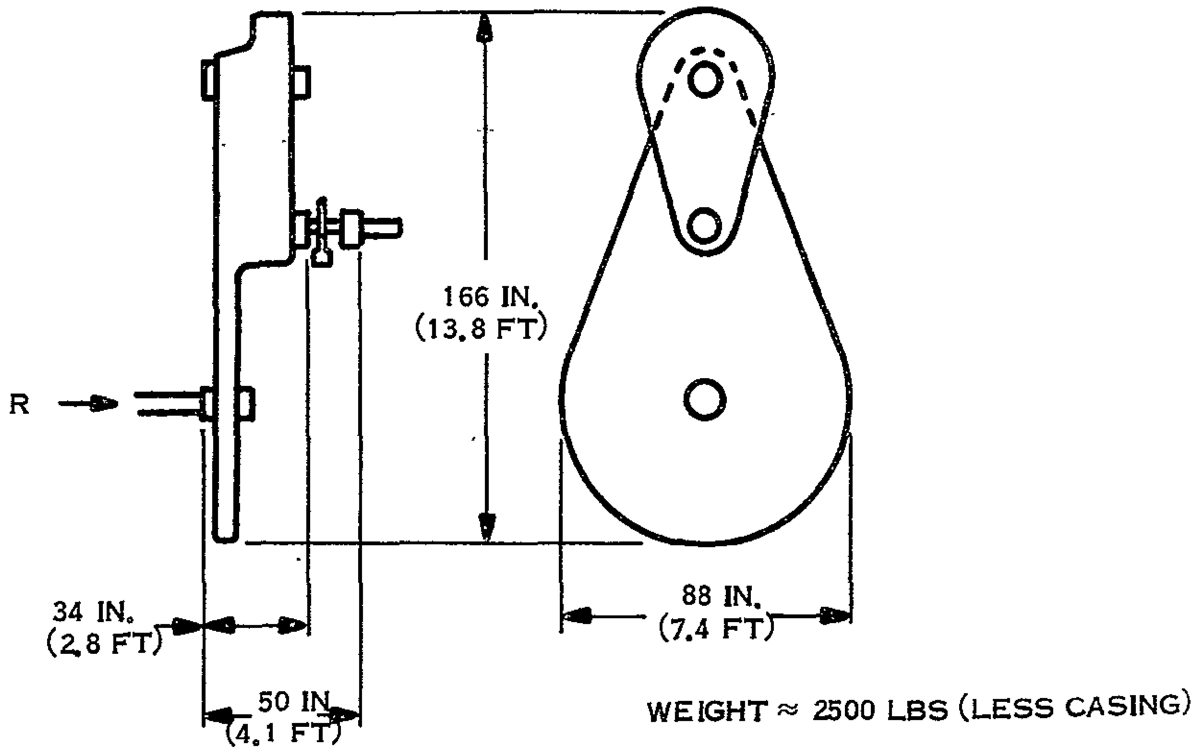

Figure 3-23. Belt and Chain Transmission Concept
(100 kW, High RPM Rotor, 1800 RPM Generator)

To determine which of the speed increaser concepts offered the most cost effective/minimum rist approach, a comparison chart (Table 3-8) was developed comparing system cost for 100 KWe and 1000 KWe with and without maintenance.

TABLE 3-8. COST COMPARISON OF FIXED RATIO MECHANICAL TRANSMISSIONS
(HIGH RPM ROTOR, 1800 RPM GENERATOR)

TRANSMISSIONS	POWER RATING		POWER RATING		CONCLUSIONS
	100 kWe	1000 kWe	100 kWe	1000 kWe	
1. GEARS	$ 8,320	$ 56,900	$ 16,640	$ 113,200	LOW AND HIGH POWER LEVEL CHOICE FOR MINIMUM DEVELOPMENT COST RISK
2. CHAIN	$ 7,150	$ 78,700	$ 14,300	$ 156,200	
3. CHAIN & GEAR	$ 5,860	$ 43,470	$ 11,720	$ 86,940	
4. CHAIN & BELTS	$ 3,520	$ 37,240	$ 12,090	$ 124,450	
	NO MAINTENANCE COST INCLUDED		MAINTENANCE COST INCLUDED		

Referring to the table, chains and belts, without maintenance or bearing supports and enclosures, appear to be the most cost effective system. When maintenance costs are included the choice shifts to chains and gears. However, when the expense to provide the enclosure, lubrication system and bearing supports necessary to accommodate the increased size are included, the choice for the conceptual design must be all gears as the minimum cost and performance risk approach. Clearly, chains became less competitive at the higher power levels because of the rapid escalation in the lubrication and housing costs associated with this concept.

3.4.2 OTHER TRANSMISSION CONCEPTS

Other transmission concepts examined included variable speed transmissions which consisted of a high step-up ratio gear box which drove a hydroviscous driver. The high gear box would step up the rotor rpm to greater than the required generator rpm. The hydroviscous unit would then provide the amount of slip required to produce the right generator speed. The slip in the hydroviscous unit would generate large quantities of heat which would be dissipated by a heat exchanger. Table 3-9 shows the cost of some typical variable speed transmissions utilizing the fixed ratio gear box in conjunction with the hydroviscous drive. Even without the cost of the heat exchanger, these systems are not cost competitive and were, therefore, not recommended for the conceptual design.

TABLE 3-9. COST OF VARIABLE SPEED TRANSMISSION
(FIXED GEAR BOX WITH HYDRO-VISCOUS DRIVER)
(1800 RPM GENERATOR)

POWER LEVEL	100 KW	250 KW	1000 KW
COST OF DRIVER	\$ 16,890	\$ 25,870	\$ 89,400
COST OF HIGH RATIO GEAR BOX	≈\$ 14,000	≈ \$ 28,000	≈ \$ 85,000
TOTAL TRANSMISSION COST	\$ 31,000	\$ 54,000	\$ 174,000

Other transmission systems considered were; hydraulic pump/motor drives, cycloidal and harmonic drives, roller traction drives, DC/DC/AC units. These systems were either not available in the performance range required or too costly.

Hydraulic pump/motor transmissions capable of operating at RPM's below 400 are not commercially available. In addition, they would require large capacity, high pressure reservoirs which would add cost and complexity as compared to the gearbox concept.

Cycloidal and harmonic drives capable of providing a 40 to 60 step-up ratio are not available in the power ranges of interest to this study. Similarly, roller traction drives were found to be commercially unavailable.

Finally, DC/DC/AC transmissions were found to have several serious drawbacks including: weight (95 lbs/KW), cost ($300/KW) and low efficiency (86%).

3.5 GENERATORS

In the conceptual design of wind turbine generator systems for electric utility applications, a number of electrical generator options are available to convert the mechanical power into electrical power. The electrical power generated must be compatible with and regulated to the requirements of existing public utility networks, as reviewed earlier in section 2.0. Therefore, the generator selection depends primarily on the cost to meet the utility requirements and on the machine efficiency in the conversion process. The generator assembly weight and volumetric capacity are secondary considerations in generator selections and do not influence the option trade-offs.

In section 3.1 on system operational trade-offs, it was learned that wind turbines can be designed and operated to provide mechanical power either at a variable or at a constant shaft speed (RPM). With a variable speed rotor, it is possible to extract more power from the wind as discussed in section 3.1, but, a variable speed transmission or a variable speed generator or both would be needed to meet utility requirements. A constant speed rotor requires a more elaborate control system, but, permits the use of both a fixed gear ratio transmission and a constant speed generator. Therefore, both constant and variable speed generators were considered in the conceptual design studies.

3.5.1 GENERATOR OPTIONS

Table 3-10 lists six candidate variable speed generators and four candidate constant speed generators, which can be compared in terms of performance and cost. It becomes immediately obvious that variable speed generator systems are not cost effective for large scale Wind Turbine Generators in electric utility applications. It also is very apparent that only two constant speed generators, the synchronous and induction, are logical candidates, both of which are almost equivalent in terms of performance and cost. The synchronous generator is more widely used and accepted by utilities and provides more versatility of operating modes and applications. Although the synchronous generator appears to have some distinct technical and economic advantages, either concept can be employed in a WTG without significant system cost or performance differences.

3.5.2 INDUCTION VS. SYNCHRONOUS

The trade-offs between the induction and synchronous generator options during the conceptual design studies included the cost comparisons, as shown in Figure 3-24, at 100 KW and 1000 KW levels as a function of generator shaft speed. At the low power level, costs are equivalent, however the induction generator operates at a lower power factor with a significantly lower efficiency. For the higher power levels, the induction generator appears more cost effective at the higher RPM's.

TABLE 3-10. PERFORMANCE CHARACTERISTICS OF CANDIDATE GENERATOR CONCEPTS

SYSTEM CHARACTERISTICS	GENERATOR TYPE	FT^3/KW	LBS/KW	$/KW	POWER FACTOR, PERCENT	EFFICIENCY, PERCENT
CONSTANT SPEED, 1000 KW, 1800 RPM	SYNCHRONOUS	0.07	7.6	34	100	95
	INDUCTION	0.05	5.0	22	88	94
	HYDROELECTRIC	-	70	77	100	95
CONSTANT SPEED 100 KW, 900 RPM	SYNCHRONOUS	0.18	1.5	53	100	94
	INDUCTION	0.12	2.5	53	88	90
	PERMANENT MAGNET	0.085	18	1500	-	-
CONSTANT VELOCITY RATIO, 100 KW	FIELD MODULATED GENERATOR SYSTEM	0.31	14	250	-	80
	ROESEL	1.8	100	1500	-	60
	DC/DC/AC	7.5	97	300	-	80
	ACYCLIC STATIC INVERTER	0.4	15	230	-	80
	VARIABLE SPEED CONSTANT FREQUENCY	0.1	7	500	-	74
	AC/RECT/CYCLO	0.4	29	450	-	90

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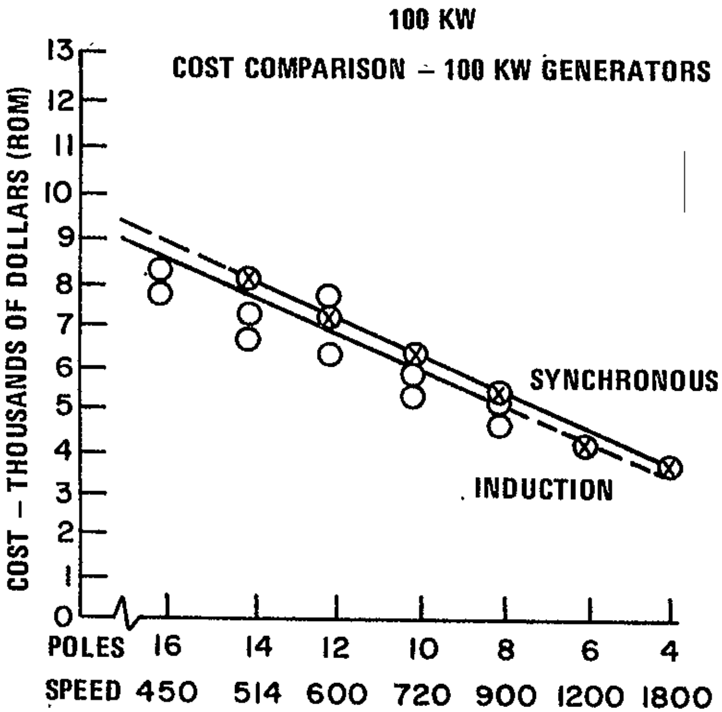
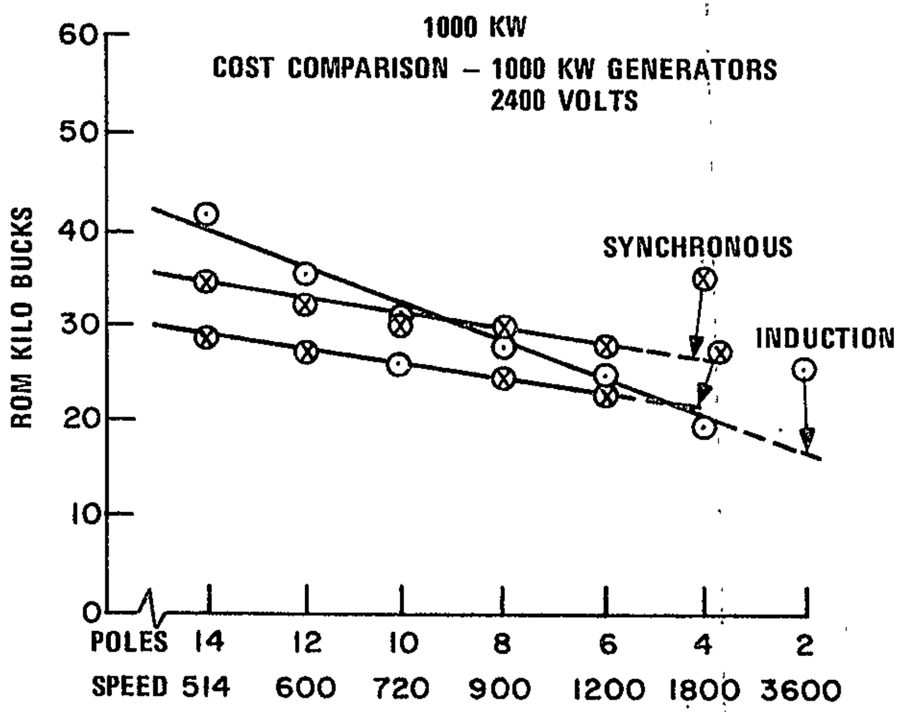




- KVA $= \dfrac{Bg\,(d^2\ell)\ \mathrm{RPM}\ A}{90 \times 10^7} \sim d^2\ell\ \mathrm{RPM}$
- COST ~ WEIGHT ~ VOLUME
- COST BASED PRICE STRUCTURE (APPLICABLE TO SYNCHRONOUS)

Figure 3-24. Cost Comparison of Synchronous and Induction Generators

System trade-offs were also conducted during the conceptual design studies on the generator shaft RPM. As shown in Figure 3-24, the costs of both induction and synchronous generators decrease with increasing shaft speed which is dependent on the number of poles in the machine. Four and two pole machines require special design and fabrication methods which are reflected in the higher cost data for the 2-pole induction generator and the 4-pole synchronous generator (NOTE: More up-to-date 1975 catalogue data indicated a significant cost reduction for the 4-pole synchronous generator, which, by the end of the conceptual design studies made the 1800-RPM synchronous generator more cost effective than the 1200-RPM machine and made the synchronous generator more competitive with the induction generator). Generator shaft speed trade-offs with the effect of step-up ratio on transmission costs indicated it was more cost effective to utilize the highest generator shaft speed available. This conclusion results from the fact that the transmission cost is primarily a function of the low speed shaft input torque for large numerical gear ratios.

3.5.3 GENERATOR PROTECTION

Available options for generator protection were also evaluated in the conceptual design studies. An adequate system of components for standard generator protection is outlined in Figure 3-25 for a 100 KW system. Cost estimates indicate approximately $10,000 in protective relaying and switchgear equipment is required to protect a $4,000 generator, which does not warrant such expensive protection. It is recommended that electric circuitry, which can be combined with the control electronics, be developed as a substitute for protective equipment. Once the electric circuitry is developed, it can cost effectively serve any size generator.

3.5.4 GENERATOR RECOMMENDATIONS

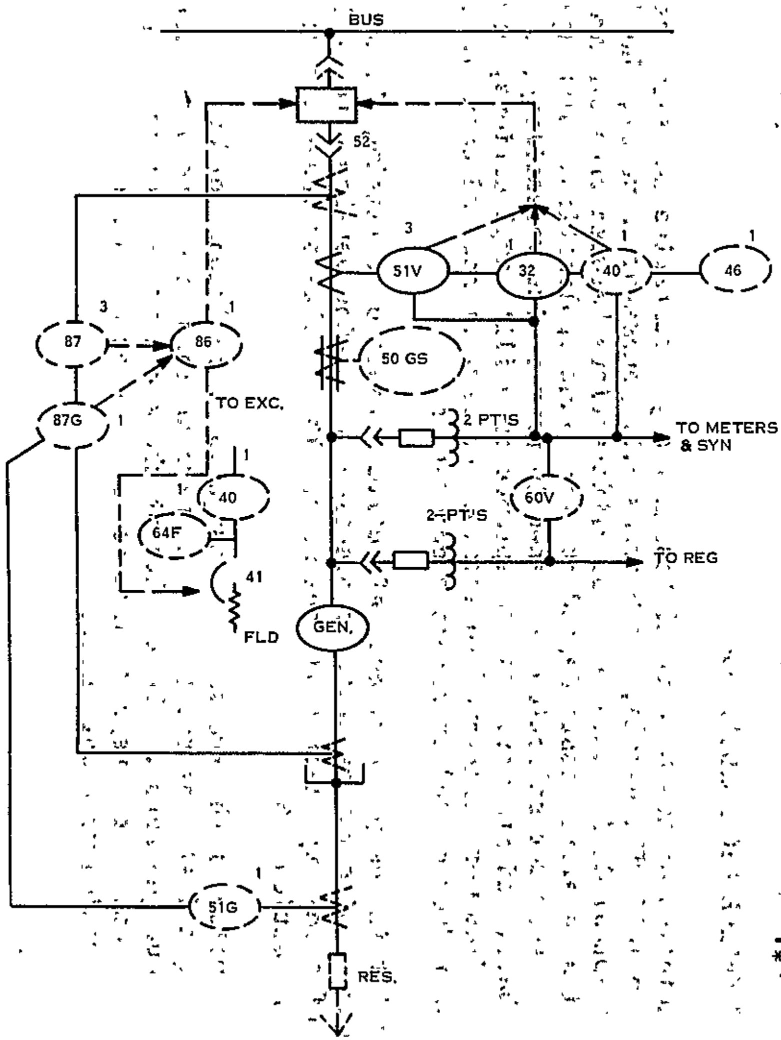
At the conclusion of the Conceptual Design Task, a tentative decision was made to select an 1800 RPM, 4-pole, synchronous A.C. generator, however, the depth of the conceptual design study did not differentiate between true synchronous and induction generator costs dependent upon plant application. Although the synchronous generator selection was based in part on its versatility, its extensive data base, and its universal acceptance in utility applications, more advanced studies were continued to examine the performance and cost trade-offs between the induction and synchronous generators.

At this point in the program, cost effective methods of utilizing the system control electronics for generator protection were being developed to minimize the cost of accessory electrical equipment. The results of these efforts are detailed in Section 5.3.

At the conclusion of the Preliminary Design Task, additional effort was given to developing a design of the 500 KW WTG with an induction generator replacing the synchronous generator. The results of these efforts are discussed throughout the remainder of the report where comparisons of synchronous and induction generator operation were warranted.

3.6 CONTROLS

Conceptual design studies for a control subsystem were focused primarily on defining the control functions and modes and designing the least expensive



		ROM
52	CIRCUIT BREAKER	$1800
51V	VOLTAGE RESTRAINED OVERCURRENT RELAY (3)	900
51G	GROUND PROTECTION RELAY	375
32	ANTI-MOTORING PROTECTION	623
40	LOSS OF EXCITATION	771
46	PHASE UNBALANCE - ROTOR OVERHEATING	100
64F	GENERATOR FIELD GROUND	450
60V	LOSS OF POTENTIAL TRANSFORMERS (PT)	450
86	LOCK OUT (OPERATED BY 87)	350
87	DIFFERENCE CURRENT (DUE TO SHORT CIRCUITS) (3)	1000
86G	GROUND DIFFERENTIAL (NEUTRAL GROUNDED THRU CURRENT LIMITING IMPEDANCE)	400
	*AUTOMATIC SYNCHRONIZER (3 LINES)	2000
	ROM TOTAL	10,000

*ADDED FOR AUTONOMOUS OPERATION

Figure 3-25. One Line Diagram - Standard Generator Protection systems to satisfy the control requirements. In order to achieve this the control functions were divided into three basic groups: sensors, actuators and deciders. It was learned early in the conceptual design studies that the control functions were rather extensive, that the control modes were complicated, and that WTG control and power regulation in a variety of wind conditions would require a large number of complicated logic processes to be built into a control subsystem. Therefore, the "decider" was recognized as the key component in the controls design.

3.6.1 CONTROL FUNCTIONS AND MODES

Basic control functions and modes were defined in all aspects of the conceptual design studies in terms of the operational trade-offs and the system configuration concepts as well as the rotor, transmission, generator and tower options. The control functions considered in the conceptual design were:

1. Start-up control and generator cut-in
2. Azimuth control (wind direction orientation)
3. Rotor control/power regulation
4. Generator output monitoring
5. Wind/WTG status, data computation, storage, transmission
6. Generator cut-out and shut-down control
7. Generator, WTG, and utility protection
8. Auxiliary/Emergency Power
9. Control aspects of a maintenance/non-operating mode

The control modes, which are outlined schematically in Figure 3-26, included each of the operational modes associated with start-up, operating, shut-down, non-operational protection and maintenance. All of the secondary control modes associated with wind conditions, power regulation, system faults, and emergency power were also considered in the conceptual design studies.

In defining the control system functions and modes, wind conditions had the most significant effects on the conceptual design. The problems of how to handle rotor start-up and speed control with automatic generator cut-in, how to regulate power during gusting winds, and how to maintain control during marginally high and low wind speeds to prevent repeated generator cut-in and cut-out cycling were addressed.

Requirements for the control system to handle the various wind conditions emphasized the need for a good logic and decision element. It was established that cut-in would depend on a "start decision" based on pre-start data monitoring and a minimum of 6 minutes at cut-in wind speeds, during which time preparations would be made to yaw the rotor into the wind and to bring the rotor up to synchronous speeds with blade control. The various control functions were determined for marginally high and low wind conditions. Control functions and modes were also studied in detail for gusting conditions during the conceptual design. Using the available gust model (See appendix 8.3) and a specified maximum rate of change in wind velocity of 200 MPH/SEC, either positive or negative, control system requirements were evaluated in terms of power regulation and control system response using generator excitation and pitch/yaw control. Rotor yaw control was found to be totally inadequate for power regulation in gusting wind conditions.

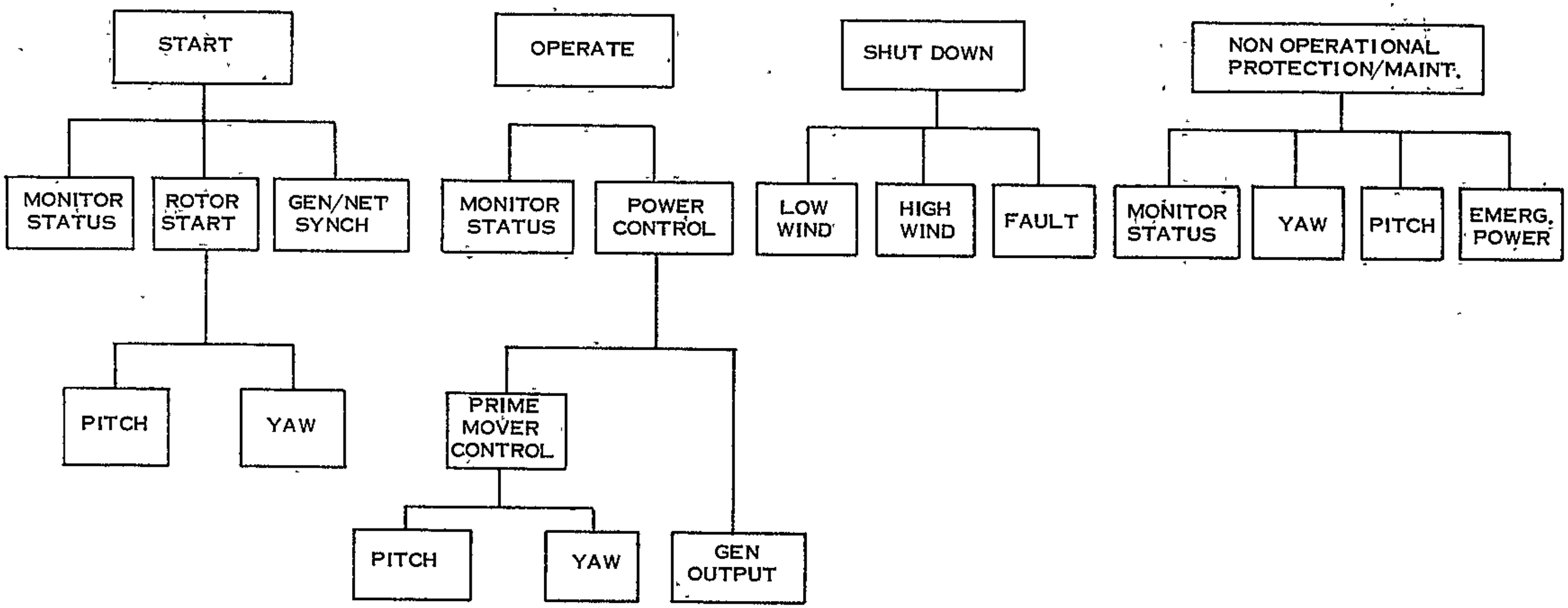


Figure 3-26. Control Modes (Use of Control Functions)

3.6.2 COMPONENT OPTIONS

The control system concepts were broken down into three basic elements; sensors, deciders, and actuators. The component options available for each one of these elements were covered in preliminary design studies and included:

A) SENSORS: either mechanical, electronic (transducers), pneumatic, or relay

B) ACTUATORS: either hydraulic, electric pneumatic or relay

C) DECIDERS: either relay, logic module, analog circuitry, micro computer, fluidics or mechanical

Each of the component options were evaluated in terms of control applications, system compatibility and cost. Most of the effort was concerned with the "decider" to provide an adequate logic and decision element at a reasonable cost. It was most apparent that the microcomputer satisfied both considerations. The microcomputer is a special purpose computer with a self-conditioned syscem of memory, control, arithmetic/logic, input/output, and utilizes a permanent program. Also, it is noted that the microcomputer utilizes standard components that can be configured for specific applications. The power requirements for the microcomputer are less than 10 watts with input/output signal conditioning. Sizes can vary with from 2 to 16 integrated circuits and unit costs vary from $100 to $400.

3.6.3 SENSORS AND ACTUATORS

Primary sensors and actuators were itemized for each WTG subsystem function. Those associated with control of the tower include wind speed and direction sensors, an azimuth position sensor, and a yaw (azimuth orientation) drive motor. For rotor control, sensors are required for pitch angle indication, rotor lock status, and rotor imbalance indication along with actuators for blade pitch control and lock. With the transmission, sensors are required for shaft speed, shaft torque, shaft vibration, lubrication temperature, and brake status, and an actuator is required for the brake. Sensors for the generator subsystem include current and voltage sensors for each phase, generator temperature sensors, vibration sensors, field (volts and amps) frequency and a neutral current sensor. A field control actuator is also required for the generator. Generator output for network distribution also requires sensors to evaluate power output and synchronization, and requires a contractor or circuit breaker. A system of sensors and circuit breakers are also required for any auxiliary power needed from the network.

3.6.4 CANDIDATE CONTROL SYSTEMS

Three basic control systems were analyzed as candidate concepts for functional requirements and cost. These three systems are shown in Figures 3-27, 3-28, and 3-29, respectively. All three utilize electronic transducers for sensors. Three different deciders were considered including the microcomputer, the analog circuitry, and the logic circuitry. Hydraulic actuators and switch gear and relays were used with the analog circuitry. Electric actuators were used with both the logic circuitry and the microcomputer, but, with the microcomputer,

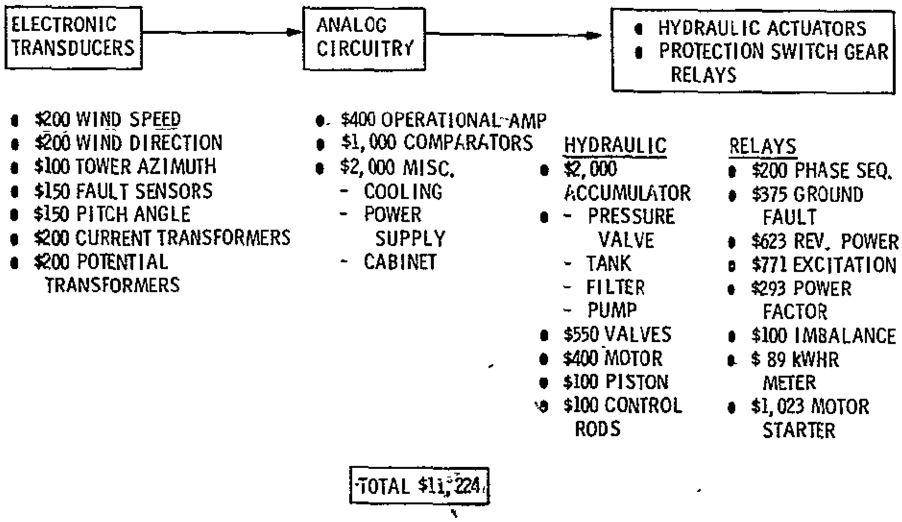


Figure 3-27. Candidate Control System No. 1

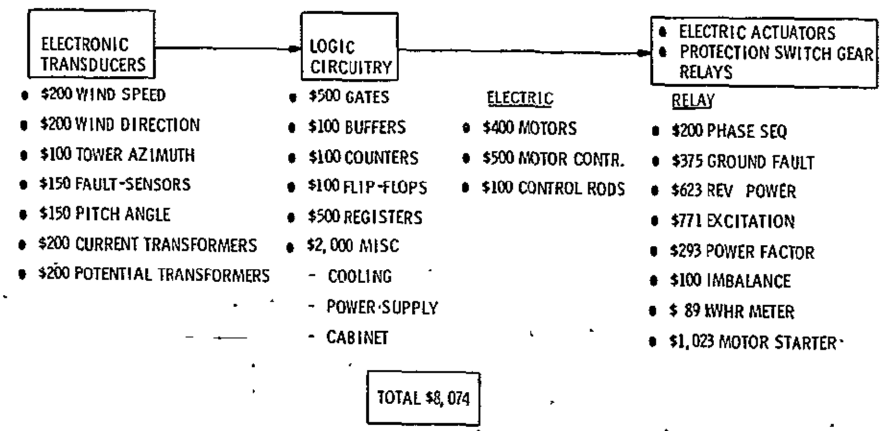


Figure 3-28. Candidate Control System No. 2

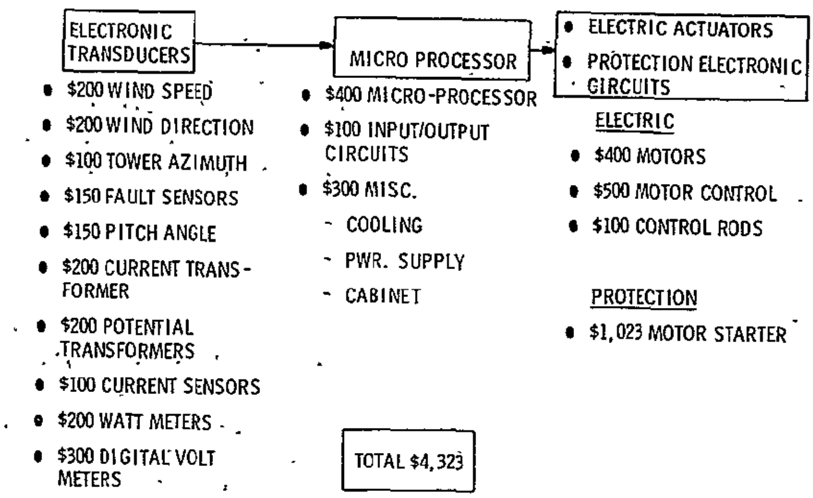


Figure 3-29. Candidate Control System No. 3

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electronic circuitry was used to replace expensive protective relays.

3.6.5 CONTROL SYSTEM RECOMMENDATIONS

The control system recommendation is based on estimated cost and viability with the WTG control requirements. The cost advantage of the microcomputer based system is only partly evident in Figures 3-25, 3-28 and 3-29 since this system also provides protection with electronic circuitry as opposed to expensive protective relays. Therefore, the electrical subsystem costs are also reduced by the inclusion of a microcomputer in the system.

3.7 TOWER

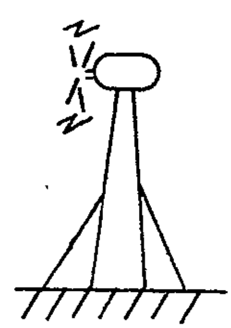
This section describes the work conducted in selecting the most promising tower concepts for use in the WTG System. While cost was the major consideration in this analysis the importance of aesthetics was not overlooked.

3.7.1 CANDIDATE DESIGN CONCEPTS

Many candidate tower design concepts were considered to be potentially attractive, for low and high power WTG's. All the concepts were evaluated within the framework of the environmental design requirements and especially the fifty year life requirement. The overall philosophy was to utilize common standard structural materials and fabrication techniques. Ten concepts were chosen for preliminary evaluation, all being vertically standing structures with a horizontal rotor axis. The ten initial concepts, some of which were variations of the same theme, were considered without regard to their present availability. The initial ten concepts are pictured and briefly described below.

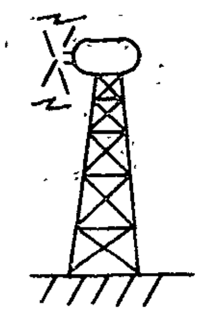
Concept "A" - Reinforced Concrete

Concrete structure, precast or slip-formed in sections, with standard imbedded steel rods or post tensioning cables.



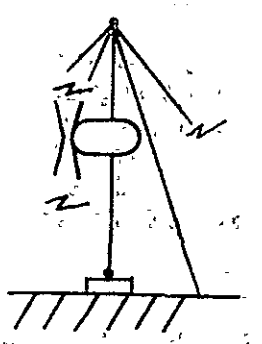
Concept "B"
Reinforced Concrete with Guys

Basic concrete structure, perhaps prestressed, with some form of reinforcing and utilizing guy wires for load sharing.



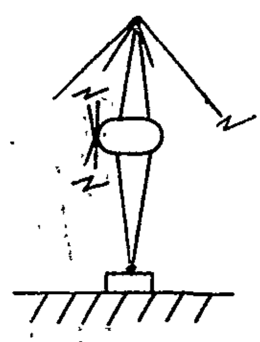
Concept "C" - Truss Tower

Tower built-up from commercially available standard structural shapes.



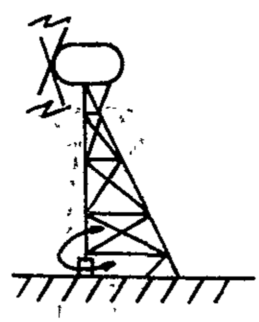
Concept "D" - Guyed Rotating Mast

Central tubular or built-up support structure with pivoted base, midpoint nacelle, and full-height guys.



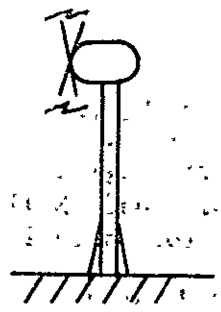
Concept "E"
Guyed Rotating Wing Structure

Similar to Concept "D" with airfoil shaped central support structure.



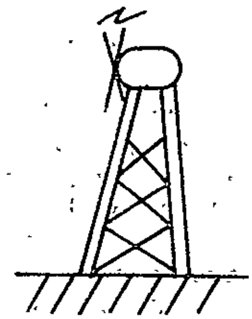
Concept "F" - Rotating Tower

Azimuth change through tower turning. Efficient establishment of load paths.



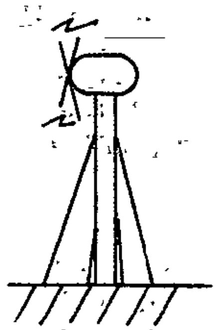
Concept "G" - Pole Tower

Primary utilization of pipe sections with auxiliary base cone.



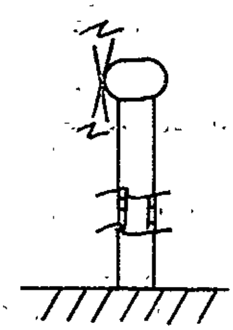
Concept "H" - Multiple Pole

Inclined utilization of commercially available pipe sections.



Concept "I" - Ball Joint Pole with Guys

Primary pipe structure with guys for load sharing and ball joint for moment reductions.



Concept "J" - Built-up Tube Shell

Central support section built-up from individually fabricated structural shapes.

The ten initial concepts collectively comprised the thinking and variations initially thought relevant to tower design. It was intended that further evaluation would possibly result in a synthesis of several good concepts for the WTG applications.

3.7.2 CONCEPT EVALUATION

Before proceeding with an analysis to size towers for the various power levels, a qualitative evaluation was utilized to determine the most attractive candidate towers. The process, although simple, had the benefit through a weighted score approach, of supplying relative concept comparisons. Table 3-11 illustrates the approach. The results highlighted six concepts for analytical consideration. In an effort to make the load analysis even more manageable, concepts "G" and "H" were considered to be secondary variations of "I" and "J". At some size, for instance, it would be necessary from a stress point of view to expand the largest commercially available pipe sections into a built-up tube section or to employ guying for load sharing. Thus concepts "A", "C", "I" and "J" were carried through to sizing and costing during the preliminary design phase.

3.7.3 DESIGN ASSUMPTIONS

Following the initial screening process outlined in the previous section, further efforts were made to select the best tower designs from the remaining 4 candidates, A, C, I and J. The following assumptions were made in performing a relative costing of the preferred concepts:

- Transmission and generator equipment located aloft
- Rotor blade ground clearance height of 20 feet
- Use of an equivalent thrust (thrust at rated power times a factor of 1.5)
- Use of a nominal foundation reflecting good soil conditions

Included in the tower cost estimates were materials, transportation of materials, fabrication, and erection costs. The early estimates did not include a systems contractors fee but did include nominal 10% supplier profits.

3.7.4 TOWER CONCEPTUAL DESIGN RECOMMENDATIONS

Figure 3-30 summarizes the relative costs of the 4 preferred concepts. As shown in the figure, the best overall choices were reinforced concrete and truss towers. In general, guying was not considered particularly effective due to the rotor imposed geometric constraints and the questionable dynamic effectiveness. As previously mentioned, pole towers generally grew into built-up shell tube sections in the higher power ranges. Reinforced concrete was attractive because of the low materials cost while truss towers are entrenched as a competitive and versatile approach to many tower applications. Shell tubes shared less extensive industry usage (liquid storage towers) and were primarily attractive for the large WTG systems. The final recommendation prior to the preliminary design phase was to pursue the reinforced concrete and truss concepts while omitting further efforts on shell tubes and pole towers primarily due to their specialized construction and potentially high costs for the WTG application.

TABLE 3-11. COMPARISON OF TOWER CONCEPTS

	RATE	"A" S	"A" WS	"B" S	"B" WS	"C" S	"C" WS	"D" S	"D" WS	"E" S	"E" WS	"F" S	"F" WS	"G" S	"G" WS	"H" S	"H" WS	"I" S	"I" WS	"J" S	"J" WS
FACTORY CONSTRUCTION	10	5	50	5	50	8	80	5	50	5	50	7	70	8	80	8	80	8	80	8	80
MINIMUM SITE PREPARATION	10	5	50	4	40	6	60	5	50	5	50	1	10	6	60	3	30	3	30	6	60
MAXIMUM EASE OF TRANSPORTATION	8	5	40	5	40	6	48	5	40	5	40	4	32	6	48	6	40	5	48	5	40
MAXIMUM EASE OF ERECTION	8	3	24	2	16	4	32	1	8	1	8	2	16	4	32	3	16	2	24	3	24
MINIMUM MAINTENANCE	7	10	70	9	63	10	70	7	49	7	49	3	21	10	70	7	56	8	49	10	70
MAXIMUM EQUIPMENT ACCESS.	6	5	30	4	24	5	30	3	18	3	18	5	30	5	30	5	30	5	30	7	42
VISUALLY ATTRACTIVE	5	5	25	4	20	5	25	2	10	2	10	4	20	4	20	4	20	4	20	4	20
MAXIMUM STIFFNESS	10	8	80	8	80	9	90	5	50	5	50	10	100	8	80	6	90	9	60	9	90
			$\boxed{369}$		333		$\boxed{435}$		275		275		299		$\boxed{420}$		$\boxed{362}$		$\boxed{341}$		$\boxed{426}$

KEY: R = RATE (RELATIVE WORTH)
S = SCORE (10 MAXIMUM)
WS = WEIGHTED SCORE

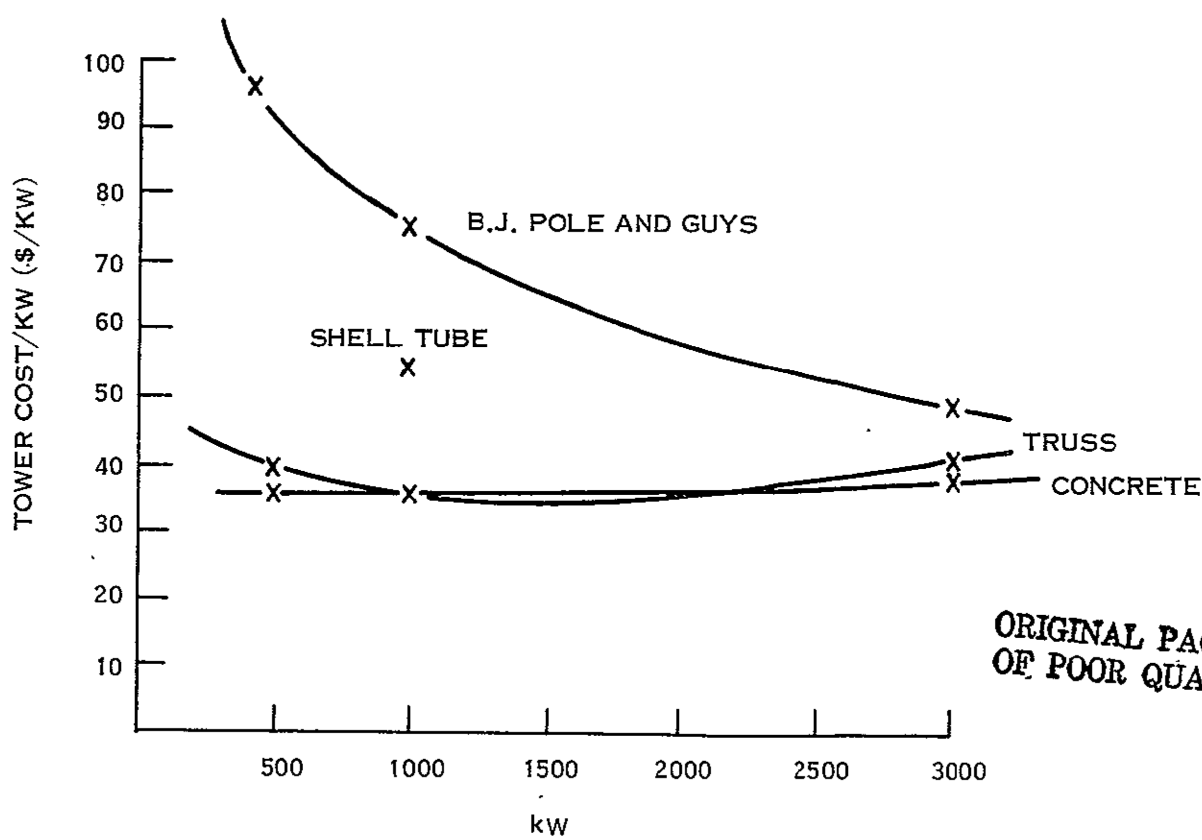


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Figure 3-30. Conceptual Design Cost Comparisons of Preferred Tower Concepts

3.8 SYSTEM SELECTION

At the end of the Conceptual Design Task, specific design approaches were selected, where possible, in order to formulate the parametric design model. Certain decisions were deferred, however, until later in the program if a clear advantage for a particular concept could not be discerned. A synopsis of the conclusions made at this point in the study are listed below.

WTG Configuration

- The optimum configuration consisted of placing the power transmission/generator equipment atop the tower. Efforts to place this equipment on the ground were precluded by the high cost of right angle drives (in the case of mechanical systems) and the large, expensive pumping requirements of hydraulic transmissions.

- Placing the rotor downwind from the tower is favored due to the inherent stability of this configuration to changes in wind direction. Also of importance is that this configuration minimizes the rotor overhang required to accommodate rotor blade deflections.

- The use of multiple rotors per tower was also considered during the Conceptual Design Phase. In comparing a 100KW system having one rotor to a 100KW system having 3 rotors atop a single tower, it was estimated that the cost of the 3 rotor system would be 1.5 times that of the single rotor system.

Operating Mode

- Systems operating at constant RPM and at constant velocity ratio were considered. The constant RPM system was preferred because it resulted in lower tower and blade loads as well as a lower cost transmission or generator. The single advantage of a constant velocity ratio system is the potential for higher energy capture, however, in the investigations performed, the increase in energy capture (3 percent) did not offset the higher system capital costs.

- Both variable pitch and fixed pitch systems were considered. A variable pitched system using blade rotation was selected due to its excellent response characteristics under changing wind conditions.

Rotor Subsystem

- A two bladed system was selected, as opposed to three, on the basis of lower cost. The technical acceptability of this decision was verified later in the program.

- A rigid as opposed to a teetered hub was also selected on a tentative basis at this point in the study. A dynamic analysis of rotor/tower interactions verified the use of a rigid hub in conjunction with 2 blades; this analysis was performed during the Preliminary Design portion of the study.

- Filament wound blades were selected as the lowest cost approach. This concept is amenable to an efficient fabrication technique, provides a lightweight structure, and requires minimum maintenance when formed in a propeller-type structure.

  The use of a helicopter type blade, balanced at the quarter chord point, was unattractive due primarily to the following reasons:

  - higher fabrication costs
  - heavier in weight
  - higher maintenance costs

  In addition, the propeller concept had been demonstrated in European Windmills and was judged to have less development risk in the size range of interest.

Tower

Many tower concepts including truss, concrete, tube shell and pole designs were evaluated. The truss and concrete approaches appeared to be least cost with the truss having the advantage of better cost predictability while the concrete offers a more aesthetic design.

The importance of selecting the proper rotor/tower natural frequencies in order to arrive at a low cost, acceptable technical solution was also recognized. As shown in Figure 3-31, unnecessary increases in the tower stiffness will aggravate the tower cost.

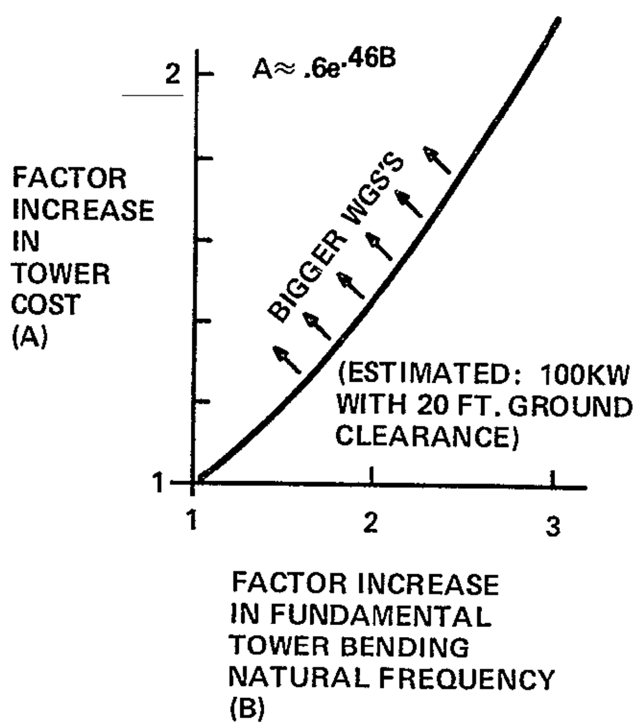

Figure 3-31. Effect of Tower Stiffness on Cost

Transmission Subsystem

- Mechanical, hydraulic and electrical type transmissions were considered. The mechanical concepts including: gearbox, belts, chains and combinations of the above were judged to be the least cost approach; of these the gearbox was preferred when maintenance aspects were addressed.

Generator

- Either synchronous or induction generators can be accommodated in the WTG, however, the synchronous generator is preferred due to its higher power factor, efficiency and the fact that it is more familiar to utilities.

Control Subsystem

- A microcomputer was selected as the main control element in the system. The microcomputer offers an inexpensive approach to handle the many operating situations which can occur. Design emphasis on the control system permits a relaxation of the design requirements on other system components.

The configuration, operating mode, and subsystem selections made in this task were included in the parametric design model. The next section discusses how this model was used to select WTG's offering low power generation costs.

SECTION 4.0

PARAMETRIC ANALYSIS/OPTIMIZATION

SECTION 4.0

PARAMETRIC ANALYSIS/OPTIMIZATION

The large quantity of independent and dependent performance and cost variables associated with the design of wind turbine generators necessitates a detailed parametric analysis with sophisticated optimization methods to resolve the most cost effective system. Mathematical relationships between system performance and costs have been incorporated into a computer program, which will search for the minimum cost per kilowatt-hour by selecting the appropriate WTG operating conditions for the assumed site. Assumptions used in the analytical model and the results of the parametric analysis are presented in the following sections.

4.1 COMPUTER PROGRAM DESCRIPTION

The design optimization computer program described in this section served as an effective system design tool, by characterizing system component requirements, operational conditions, and costs as a function of any specific site/wind environment. The functional objectives of the computer code are to determine the minimum cost per kilowatt-hr, to identify the system, and to evaluate the system cost sensitivity to a wide range of performance and design variables. Appendix 8.5 provides a listing of the WTG-OPT computer code input, output and calculational procedure.

The overall computer program structure is composed of an analytical model with ten major subroutines which interacts with an optimization code with one major subroutine to define the minimum ¢/KW-HR. A generalized outline of the overall program structure is shown schematically in Figure 4-1. The analytical model operates on a given set of base case independent variables and calculates the cost per kilowatt-hour. The optimization code assesses the cost sensitivity to each independent variable and simultaneously selects new base case independent variables for the analytical model. The procedure continues, using the method of steepest ascent, until the optimization code determines that the optimum has been reached.

The primary linkages of the major subroutines in the analytical model are shown by the flow chart in Figure 4-2. Major inputs to the program include the assumed wind site statistics, rotor blade aerodynamic performance characteristics, and system component cost/weight equations as a function of appropriate system performance variables. Major program outputs are minimum ¢/KW-HR, subsystem cost and weight breakdowns, optimum performance and design parameters, a breakdown of the annual power and energy production, and a chronological summary of the economic investment analysis.

4.2 COMPUTER CODE ASSUMPTIONS

Inputs to the code are based on a number of performance and economic assumptions, which make up the subroutines of the analytical model. The input assumptions can easily be changed or varied depending on the desired options or objectives to be reflected in the optimization. The performance assumptions include the environ-

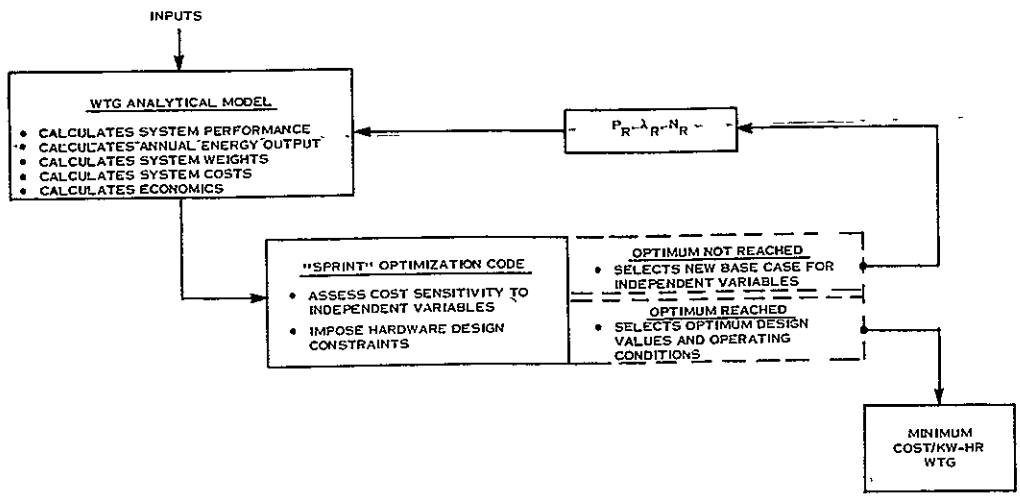

Figure 4-1. WTG-OPT Computer Code Model

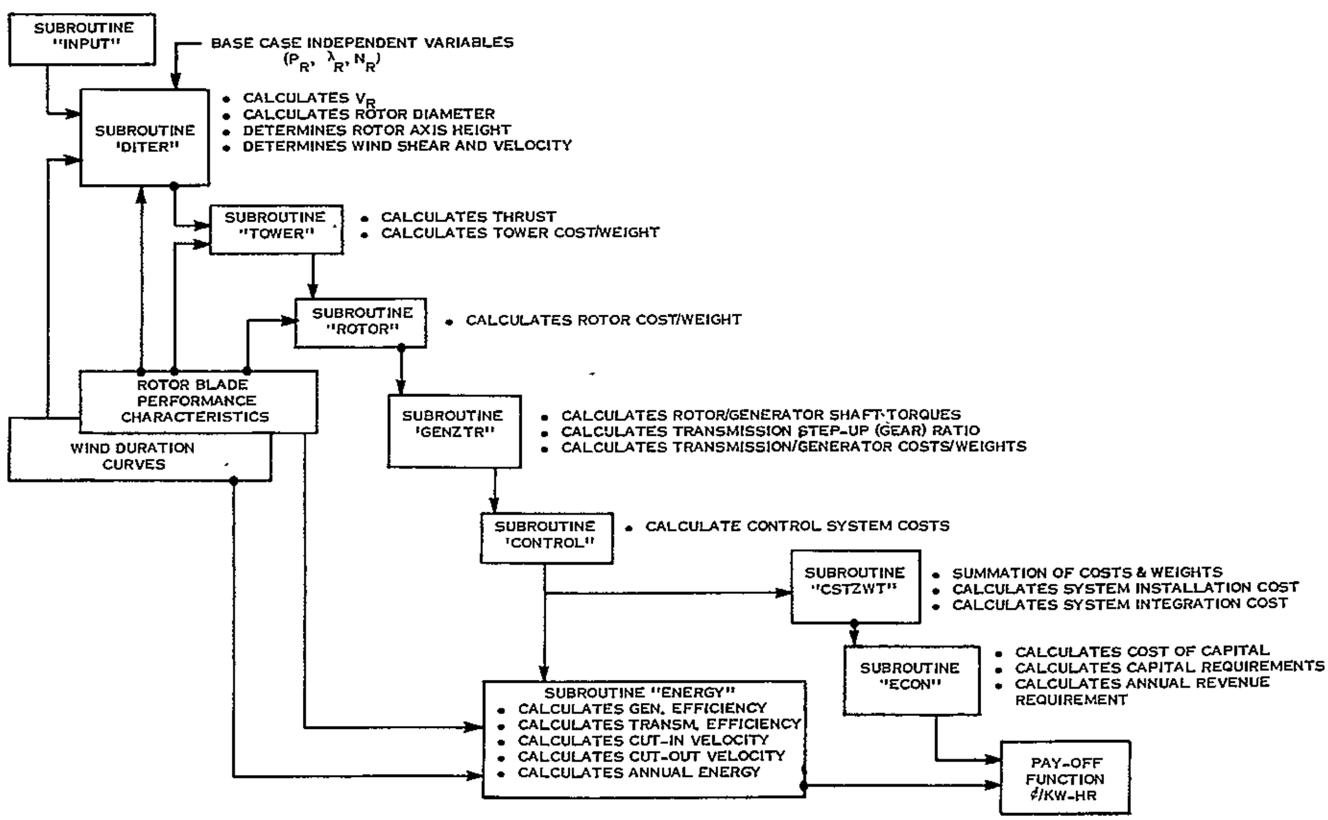

Figure 4-2. WTG Analytical Model

mental aspects, rotor aerodynamic characteristics, mechanical efficiencies of power transmission, and the electrical efficiencies of power regulation and energy conversion. The economic assumptions encompass the equipment capital costs, the methods and costs of capitalization, costs of systems integration, installation, operation and maintenance, and the effects of system life and annual production rate on costs. A third category of input assumptions is that of subsystem component weight, which indirectly affects both the performance and economics of the system optimization.

4.2.1 PERFORMANCE

4.2.1.1 Environmental

The environmental assumptions include a wind duration curve with a characteristic annual median wind speed, a wind shear profile which is dependent on the site terrain, and a characteristic site air density which is dependent on altitude and annual median relative humidity.

The wind duration curves assumed in this study and modeled in the computer code were shown earlier in Figure 2-1 and were provided as part of the study ground rules.

The wind shear profile represents the increase in wind speed with height above the ground, and is modeled in the computer code in equation form as an exponential variation with height. The exponential constant used was 0.167, which is representative of level terrain with no ground interference such as trees and buildings.

The air density assumed ACAO US standard sea level atmosphere, 1.219 kilograms per cubic meter. System performance will vary with atmospheric density, due to the altitude elevation and mean relative humidity which are characteristics of specific site locations.

4.2.1.2 Aerodynamics

Non-dimensional aerodynamic characteristics for the rotor blading in the form of power and thrust coefficients as a function of velocity ratio, are modeled in the computer program in tabular form for any given rotor-blade design option. By introducing the scaling parameters for diameter, rpm and wind conditions, the aerodynamic coefficients are translated directly into mechanical power, radial torque, and axial thrust on the main shaft. A large number of aerodynamic variations were used as computer program inputs during the parametric analysis; these represented variations in the number of blades, different airfoil shapes, different blade activity factors, and blade twist.

4.2.1.3 Mechanical

The computer analytical model assumes that the axial thrust interacts with the tower on the rotor axis through a radial thrust bearing and that the mechanical power is transmitted to a speed-increaser (gear box) in the form of radial torque at a constant rotor shaft speed.

In terms of mechanical assumptions the rotor shaft torque, rotor shaft speed, and generator shaft speed define the gear box requirements (size, speed, step-up ratio, and number of stages) and efficiency. Nominally, the mechanical power transmission efficiency of gear box speed-increasers varies between 96% and 99% at rated torque depending on the number of stages and other unknown variables. The computer code assumes a mechanical power transmission efficiency of 96% at rated torque.

4.2.1.4 Electrical

The electrical input assumptions to the computer code include the generator efficiency variations and the limiting generator operating conditions involved in the electrical power output and net energy conversion.

The efficiency characteristics of generators vary considerably with both the type and the size of the machines, as well as from one design to the next. The computer code assumes a synchronous A.C. generator, operating at a unity power factor. The code utilizes a computational procedure based upon specific designs to determine the variation in efficiency with percent of rated power and includes efficiency corrections for variations in rated power.

Other electrical input assumptions in the computer code are:

1. The minimum power requirements for generator cut-in as a function of rated power, which is used to determine the minimum wind velocity to overcome generator losses at rated RPM.

2. A minimum power coefficient required to maintain operational stability at rated power to determine generator cut-out wind speeds. This computation can be automatically bypassed by the maximum statistical wind speed in the wind duration curve or can be bypassed by imposing a cut-out wind speed directly on the wind duration curve.

3. Calculation and integration of the electrical power duration curve in 100 hour increments from cut-in to cut-out wind speeds to determine the annual accumulated energy generated.

4.2.2 ECONOMICS

A significant portion of the analytical model of the computer program is devoted to the calculation, integration, and summation of cost information. Cost assumptions are included throughout the analytical model in equation form; these expressions relate cost data with the appropriate system environmental, performance and economic variables.

The derived cost information is categorically summed for selected inputs to the Economic Model which determines the annual revenue requirement for any projected system life. The resultant annual revenue requirement is then combined with the resultant annual accumulated energy to determine the cost per kilowatt-hour.

4.2.2.1 Equipment Capital Costs

The costs of equipment and materials were derived from vendor quotations and 1974-5 catalogue data. In most cases, multiple sources of cost information were obtained, and the lower costs were favored in curve fits of the cost data. Table 4-1 lists the cost expressions used during the parametric analyses.

The capital cost relationship having the most uncertainty was the rotor subsystem expression. During this phase of the program the rotor cost was related to the blade diameter, however, as a result of the Preliminary Design work it became apparent that a more complex relationship would be useful in expediting the optimization procedure. Consequently, further work is recommended to determine the effect of specific system operating parameters, such as RPM, and cut-out velocity, on rotor cost.

Statistical data were also obtained from vendors on component life, replacement and maintenance costs, which are assumed representative of recurring costs and included in the economic model.

Equipment crating and delivery costs are also assumed in addition to quoted equipment costs. Actual crating costs were solicited from vendors for the rotor hub and blades, the gear box, and the generator. Equipment delivery costs for all equipment except the tower, are based on a standard shipping rate of $5 per 100 lbs. weight, which is representative for accessible sites within a 1000 mi. radius.

The influence of production quantity on component cost was solicited from vendors and the available information was included. Other miscellaneous equipment was assumed to be discounted at 5, 10 and 15% based on production ratio of 10, 100 and 1000 units, respectively, which is representative of the electrical industry.

4.2.2.2 Systems Engineering, Integration and Installation Costs

Contractual system engineering, integration and administration is assumed as a capital cost, which is included in the economic model. The impact of this contractor cost on system cost is assumed to depend heavily on the production quantity of the contract, and is expressed as follows:

$$C_{ENG} = [55{,}000n^{0.5} + 150{,}000]\,\frac{1.07}{n} + 0.07CC$$

This cost is considered a representative figure to cover engineering design, technical liaison, purchasing, contract management and administration through system operational check-out and delivery and a 7% systems engineering fee (see section 5.1.5 for further discussion).

Installation costs are represented in the computer code by the following expression:

$$C_{INST} = 12{,}000n^{-0.069} + 44\,(P_R)\,(n)^{-0.29}$$

which also reflects a strong dependence on production quantity. Large production

Table 4-1. Capital Cost Expressions Used in WTG-OPT

Subsystem/Component	Capital Cost Expression Used During Parametric Analysis, $
Rotor	$C = (33.77\ D^{2.22}/n^{0.11}) + 20,000$
Gearbox	$C = 2.987\ Q_R^{0.795}$
Mainshaft	$C = 0.01835\ Q_R^{1.097}$
Flexible Couplings	$C = 0.0324\ Q_R^{1.030}$
Main Bearings	$C = 0.150\ Q_R^{0.8645}$
Primary Brake	$C = 145.7\ P_R^{0.300}$
Generator (Synchronous)	$C = 84.12\ P_R^{0.835}$
Generator Accessories	$C = 169.1\ P_R^{0.446}$
Controls	$C = 3636 + 9.2\ P_R$
Truss Tower ($T_e h_T < 2 \times 10^6$ nm)	$C = (8078. + 4.24 \times 10^{-3}\ T_e h_T)\ 1.56/n^{0.084}$
($T_e h_T > 2 \times 10^6$ nm)	$C = 11,894 + 2.40 \times 10^{-3}\ T_e h_T)\ 1.56/n^{0.084}$
Concrete Tower ($T_e h_T < 2.4 \times 10^6$ nm)	$C = 0.65\ [9397. + 2.33 \times 10^{-3}\ T_e h_T]/1.13^{\frac{n-1}{n}}$
($T_e h_T > 2.4 \times 10^6$ nm)	$C = 0.45\ [5053 + 4.41 \times 10^{-3}\ T_e h_T]/1.13^{\frac{n-1}{n}}$

NOTE: These are capital cost expressions only, intended to show which system variables affect a given cost and to what degree. They cannot be employed to inter-relate the system variables. For system design the full WTG-OPT computer code (Section 8.5.3) must be used.

quantities assumes lower installation procedures and cost sharing of installation equipment for multi-unit installations. Installation costs also assume the use of pre-assembled components, accessible sites, transportation and operation of appropriate sized cranes, an appropriate union-scale labor crew and a skilled service crew.

4.2.2.3 Methods and Costs of Capitalization

The program analytical model utilizes an economic subroutine which assumes a capital investment of I dollars for a system life of n years and determines the annual revenue requirement based on a constant amount required each year to:

1. pay fixed expenses
2. pay taxes
3. pay prorated interest on debt
4. pay prorated return to stockholders
5. build a capital fund that will allow for periodic component replacement or overhead as well as retirement of prorated debt after n years.

The analytical model assumes a profit-making, tax paying public corporation, such as a public utility. Two basic assumptions of the program are:

1. The cost of capital is based on the weighted average after tax cost for a typical utility. Therefore, the proposed WTG investment is a part of the overall corporate capital structure and not the incremental cost of the proposed investment alone.
2. The capital fund will grow at an interest rate corresponding to the after tax cost of capital, ie, it can be re-invested in other equivalent projects.

Economic options in the program include either Straight Line, Sum-of-year-digits, or Double-Declining-Balance depreciation methods and a variable System Life. For most of the parametric analysis studies the economic subroutine uses the following economic assumptions:

1. 30 year life
2. Straight-Line Depreciation
3. 50% Debt fraction (bonds)
4. 50% Equity fraction (equity)
5. 9% Debt interest
6. 11.5% Return on equity

7. 48% Corporate tax rate

8. 0% Investment tax credit

Using this economic model, the annual revenue requirement varies from 16 to 17% of the total initial capital cost.

4.2.3 COMPONENT WEIGHTS

Calculations of component weights are made in the analytical model using equations relating component weights to the various system/component performance parametrics. Weight data were obtained from vendors, vendor catalogues, and engineering assessments and curve-fitted to provide equations for the analytical model. Component weight outputs are used for component specifications and for determining shipping costs. A listing of the expressions used in the code are given in Appendix 8.5.

4.3 PARAMETRIC ANALYSIS RESULTS

Parametric analyses were performed to determine the optimum (minimum cost per KW-HR) system design and operating conditions and to determine the system cost sensitivity to some of the environmental, performance, and design variables. The scope of the parametric analysis included rated power levels from 50KW to 3000KW and mean annual wind speeds from 9 mph to 21 mph. Variations in the blade cost assumption, blade characteristics, site characteristics, and economic conditions were also investigated. Design constraints used in the parametric analysis included only standard hardware, state-of-the-art technology and an operational tip speed of less than 107 m/s. This latter constraint was used in order to simplify the blade dynamic analysis. It should be noted that tip speeds of commercially designed propellers are typically 214 m/s. The following results of the parametric analysis were used to establish system design specifications.

4.3.1 OPTIMIZATION STUDIES

The optimization process in the parametric analysis assumes three independent variables which are optimized and from which other system specifications are determined as dependent variables. The independent variables are:

(1) The rated power level,

(2) Rotor RPM, and

(3) Velocity ratio at rated wind speed.

These three independent variables totally define the system performance by relating the wind statistics, the rotor aerodynamic characteristics, and the energy conversion techniques. Optimization of these independent variables define the ideal system specifications for minimum energy costs.

4.3.1.1 Power Level

Variations in energy costs with rated power level for any given annual median wind speed is shown in Figure 4-3. The results indicate that energy costs continue to decrease with increasing power level up to 3000KW, however, a transmission design constraint imposes an upper limit on the rated power level specification. This transmission constraint reflects non-standard gear boxes, which are torque rated above 4×10^6 n-meters and which will require some nominal engineering development. The results clearly demonstrate that the higher units are the most cost effective and that energy costs will escalate rapidly with units rated below 500KW.

4.3.1.2 Rotor RPM

Rotor speed is a key design variable in the optimization procedure, and by optimizing RPM as an independent variable, a maximum energy capture is achieved. The variations in optimum rotor speed with rated power is shown in Figure 4-4, which indicates that the ideal rotor speed decreases with increasing rated power level and increases with the annual median wind speed for any particular site. During the parametric analyses the following expression was used to calculate rotor cost:

$$C = [33.77(D)^{2.22} / n^{0.11}] + 20,000$$

Since experience in previous work had shown that blade cost is also sensitive to tip speed, as higher tip speeds are reached, it was decided to utilize a constraint of 107 meters/sec. This had the effect of maintaining the rotor cost on an equal basis for all of the parametric work. Later, during the Preliminary Design, it was found advantageous to exceed this constraint on the 1500 KW unit. This was due primarily to the fact that the blade ground clearance was raised from 6 - 15 meters, thereby, placing the rotor in a higher wind.

4.3.1.3 Rated Velocity Ratio/Rated Wind Speed

The velocity ratio, which is the blade tip speed-to-wind speed ratio, is optimized as an independent variable to determine the most cost effective rated wind speed. The optimum rated velocity ratio is resolved from the most cost effective use of the rotor power coefficient curve, which in effect is the rotor aerodynamic efficiency variation with wind speed at constant RPM. Therefore, optimizing for rated velocity ratio provides the highest average rotor aerodynamic efficiency over the total operational wind spectrum. The resultant optimum rated wind speeds are expressed as a rated-to-mean velocity ratio at a 9.14m reference height in Figure 4-5 as a function of rated power level and site annual median wind speed. Results demonstrate the rated-to-mean wind speed ratio increases with power at low (< 500KW) power levels, becoming essentially constant above 500KW, and decreases with increasing annual median wind speed at all power levels.

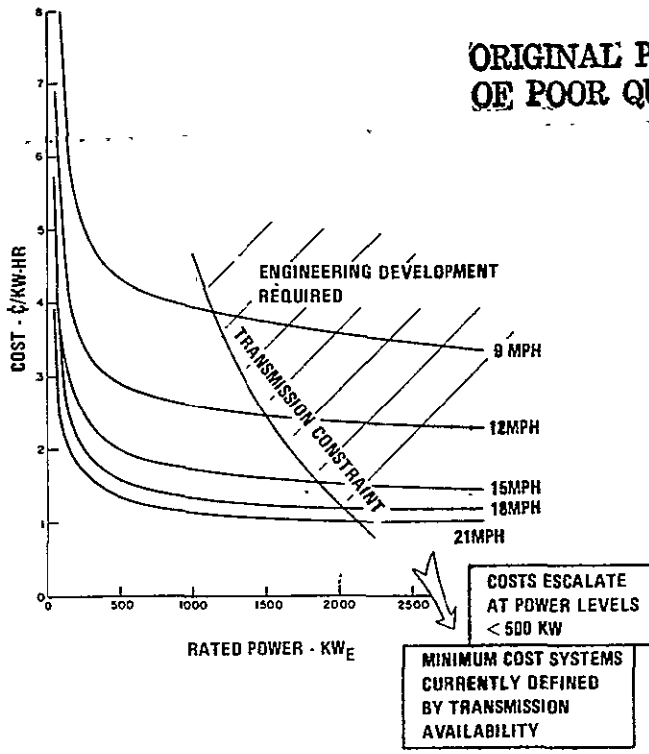

Figure 4-3. Effect of Rated Power on Power Generation Costs

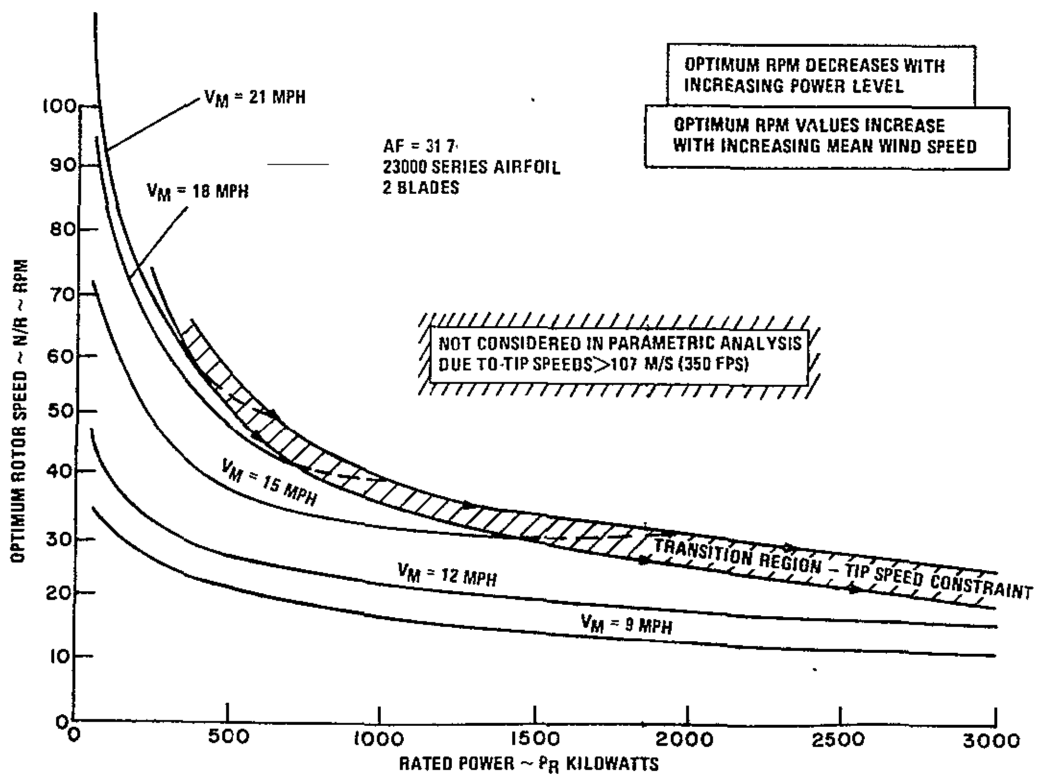

Figure 4-4. Optimum Rotor Speed (RPM) vs. Rated Power

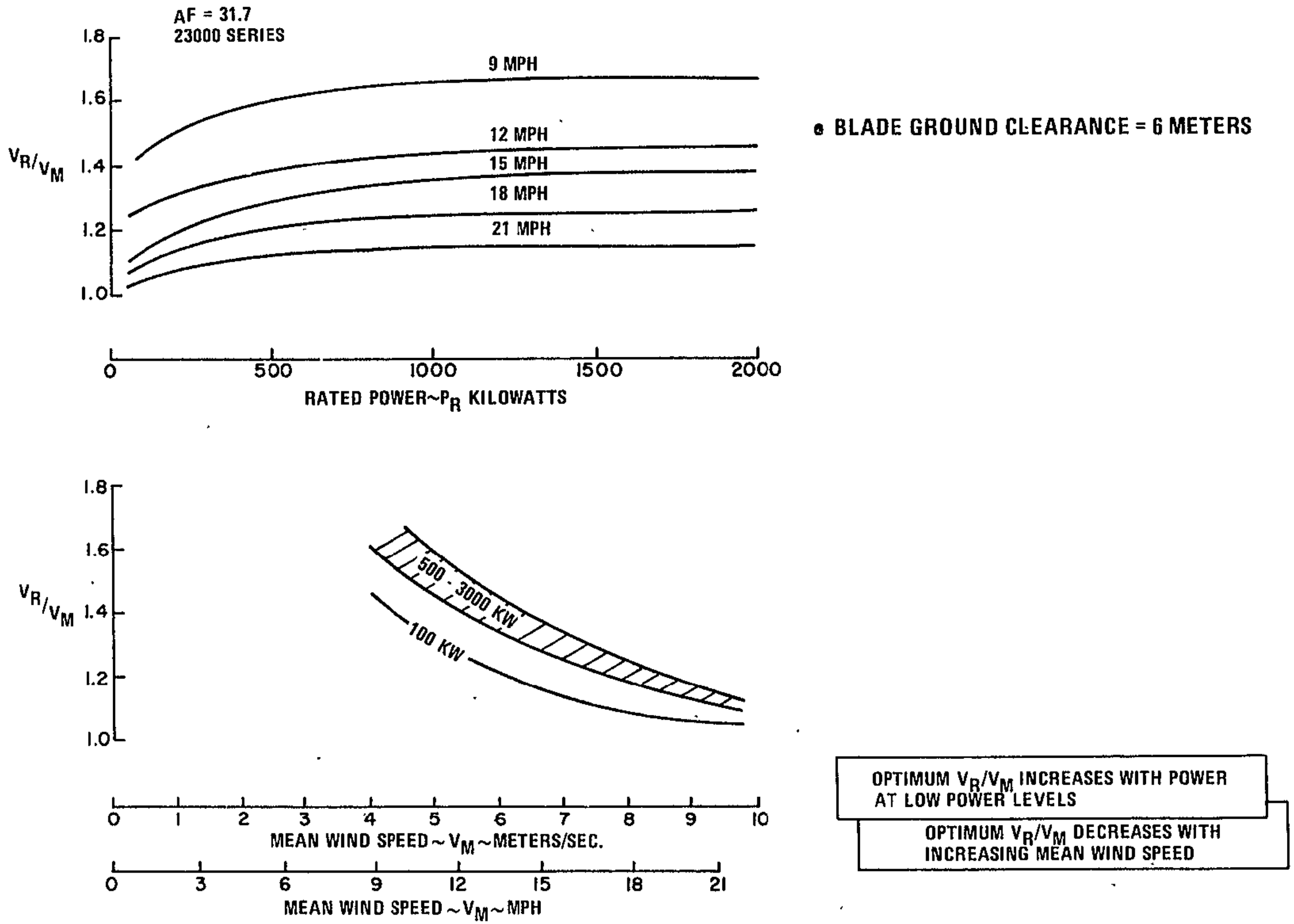

Figure 4-5. Optimum Rated-to-Median Wind Speed Ratio

4.3.1.4 Rotor Diameter

The most important dependent variable is the resultant optimum rotor diameter that satisfied the requirements for optimum rated power at the optimum rated RPM and rated λ. The variations in optimum system rotor diameter with rated power are shown in Figure 4-6 as a function of the site annual median wind speed. Results show that high power systems can be designed and be cost effective at the lower mean wind speed sites if the rotor systems are large. However, results also demonstrate that for any given power requirement, rotor diameters can be reduced significantly by utilizing the higher mean wind speed sites. Figure 4-6 also shows the effect of the tip speed constraint, as mentioned earlier, on increasing rotor diameter while decreasing rotor RPM to maintain a given power level.

4.3.1.5 Cost Sensitivity Studies

Effects of a variety of system variables on cost were covered in the parametric analysis and are reviewed in Section 6.0. Studies included cost sensitivity to the following parameters:

- ENVIRONMENTAL

  1. Annual Median Wind Speed
  2. Wind shear characteristics
  3. Altitude

- DESIGN

  4. Tower height/rotor clearance
  5. Tower-type selection
  6. Airfoil selection
  7. Blade activity factor (Solidity, Taper, Twist)
  8. Rotor diameter

- OPERATIONAL

  9. Rotor RPM
  10. Cut-in/Cut-out velocities
  11. Rated wind speed

- ECONOMIC

  12. System Design Life
  13. Method of depreciation

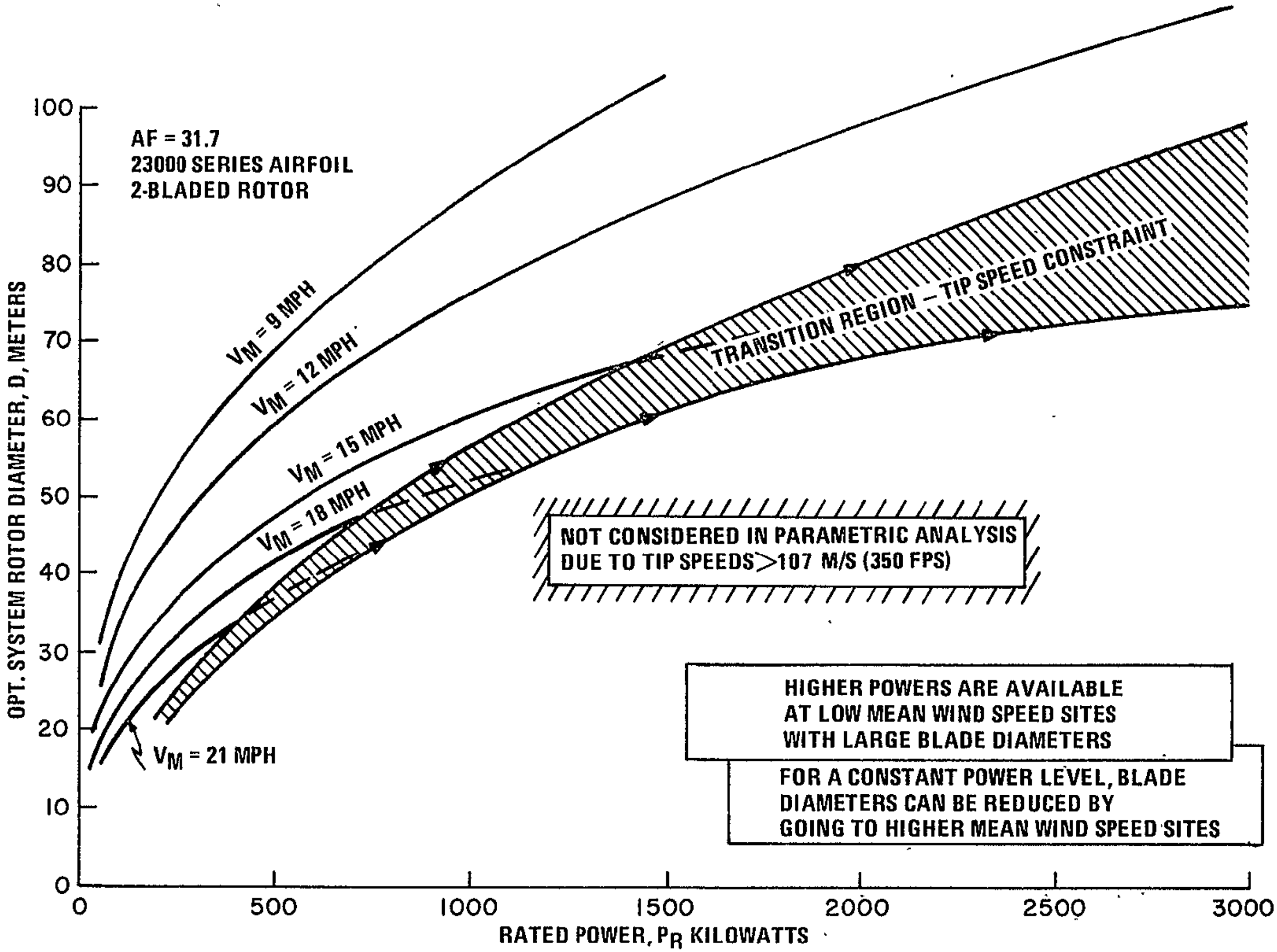

Figure 4–6. Optimum System Rotor Diameter vs. Rated Power

14. Utility Bond rate

15. System production quantity

Some of the most significant results are shown in Figures 4-7 through 4-11. The most important parameter affecting power generation cost is median annual wind speed. Figure 4-7 illustrates the necessity of having high wind speeds, regardless of power level, to obtain low power generation cost.

The effects of rotor speed on energy cost is shown in Figure 4-8 as a function of median wind speed and power level. While RPM affects the cost of the rotor and transmission, the primary influence of rotor speed on system cost effectiveness is maximum energy capture. Figure 4-8 demonstrates that for any given median wind speed it is cost effective to reduce RPM with increasing rated power level, and that for any given rated power level, it is cost effective to increase the RPM for sites with higher annual median wind speeds.

The sensitivity of system cost effectiveness to rotor diameter is shown in Figure 4-9, as a function of rated power level and annual median wind speed. Significan observations from these results are that large rotor diameters are cost effective and that higher median wind speeds allow the use of smaller rotors for any given rated power level.

The sensitivity of power generation cost with rotor diameter is shown to be greatest at power levels less than 500 KW at all wind sites. This result provide some insight into the previously stated result that larger power levels are the most economical.

In terms of the economics of system production, the cost effectiveness is improve with increased production rates. The system cost sensitivity to production quantity in terms of both capital cost and energy cost is shown in Figure 4-10. It is noted that the cost effectiveness for a small number of developmental or experimental systems is marginal and more uncertain than high quantity production and the capital and energy costs can be reduced significantly with a large number of systems which are basically similar in scale (i.e. fixed rotor size).

Summarizing the other cost sensitivity studies, the results indicated the following conclusions.

In terms of rotor blade design, system cost effectiveness is relatively insensitiveto both activity factor and airfoil selection. Blades with low activity factors are slightly more cost effective at all power levels. High and low performance airfoils indicate only slight differences in cost effectiveness. The prime airfoil selection, 23000 series, with an activity factor of 30 was the most cost effective at all power levels for an 18 MPH annual median wind speed site.

In terms of tower design, the concrete tower was shown to have a significant cost advantage as compared to the truss tower over a range of power levels (see Sectio 5.3.5). Effects of tower height indicate it is slightly more cost effective from the system standpoint to get the rotor up into the higher winds by increasing the tower height and rotor ground clearance, especially at the lower annual median wind speed sites.

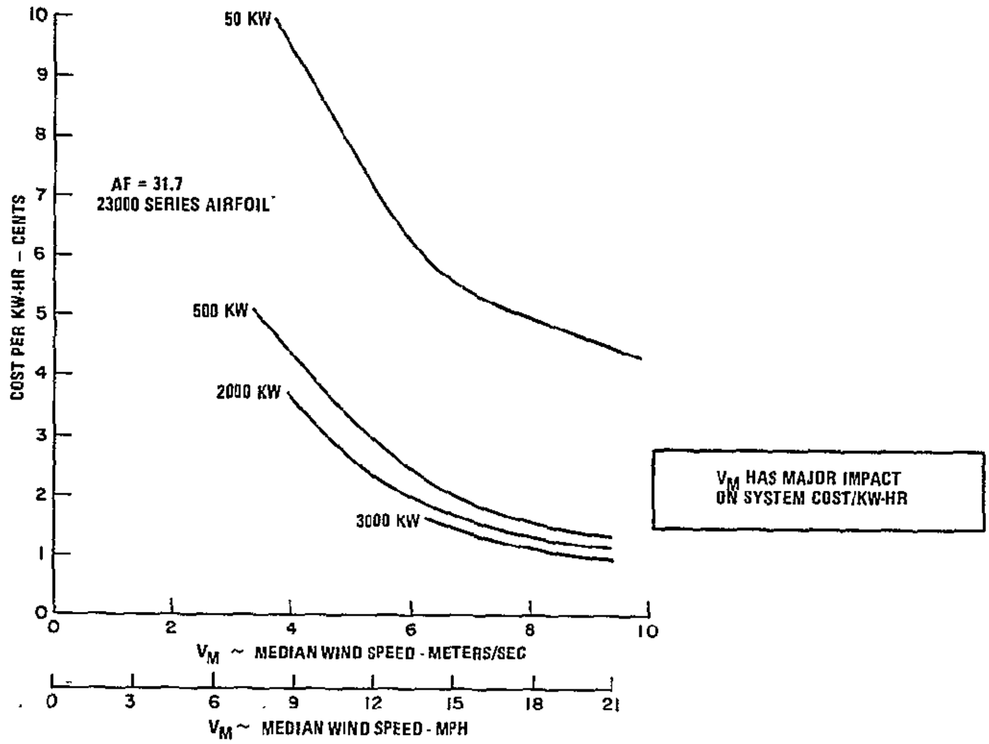

Figure 4-7. Cost Sensitivity to Median Wind Speed

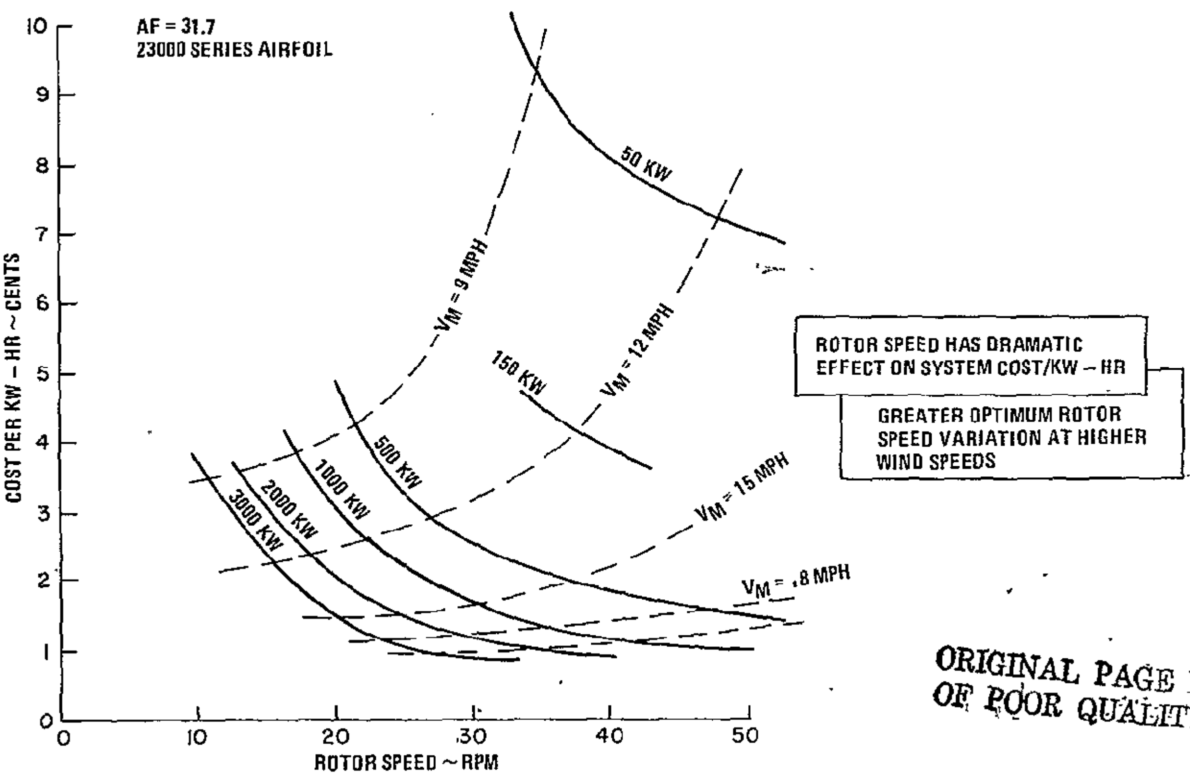

Figure 4-8. Cost Sensitivity to Rotor Speed

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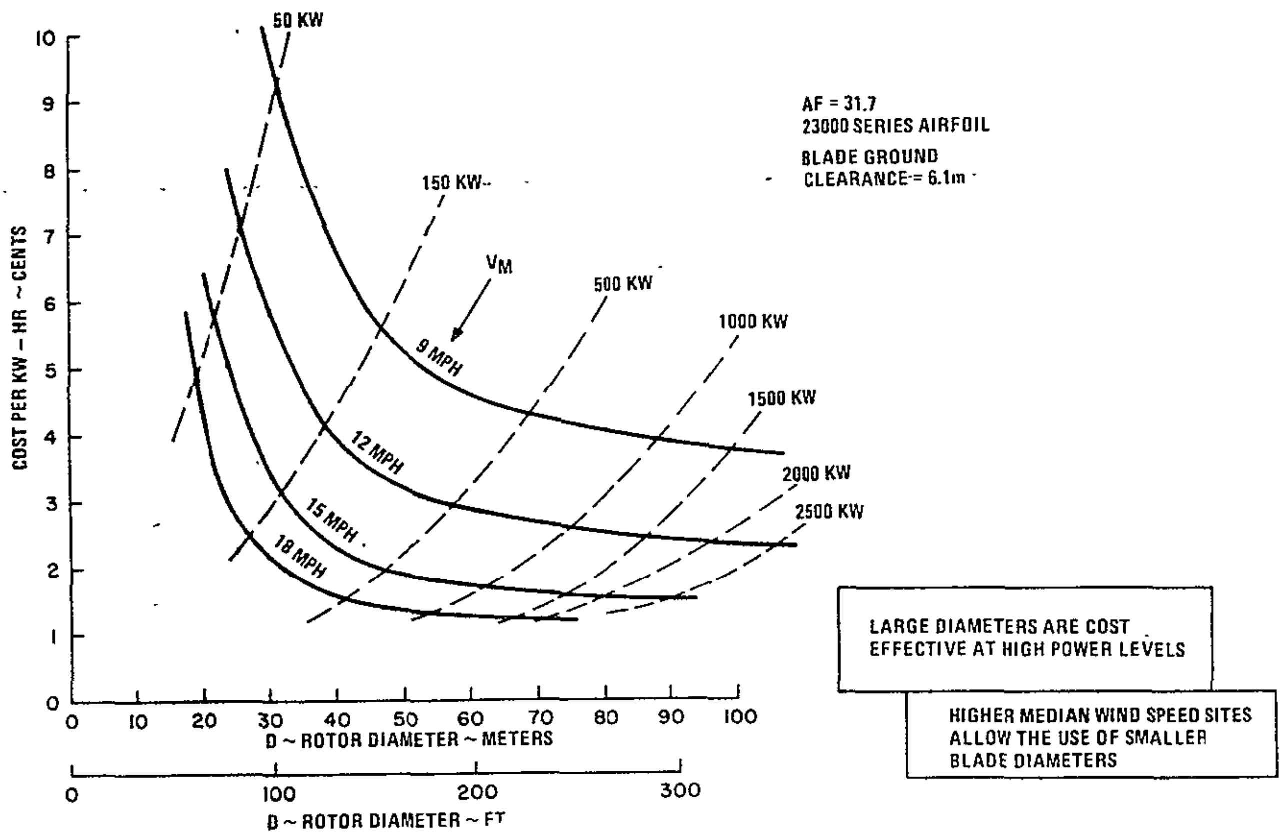


Figure 4-9. Cost Sensitivity to Rotor Diameter

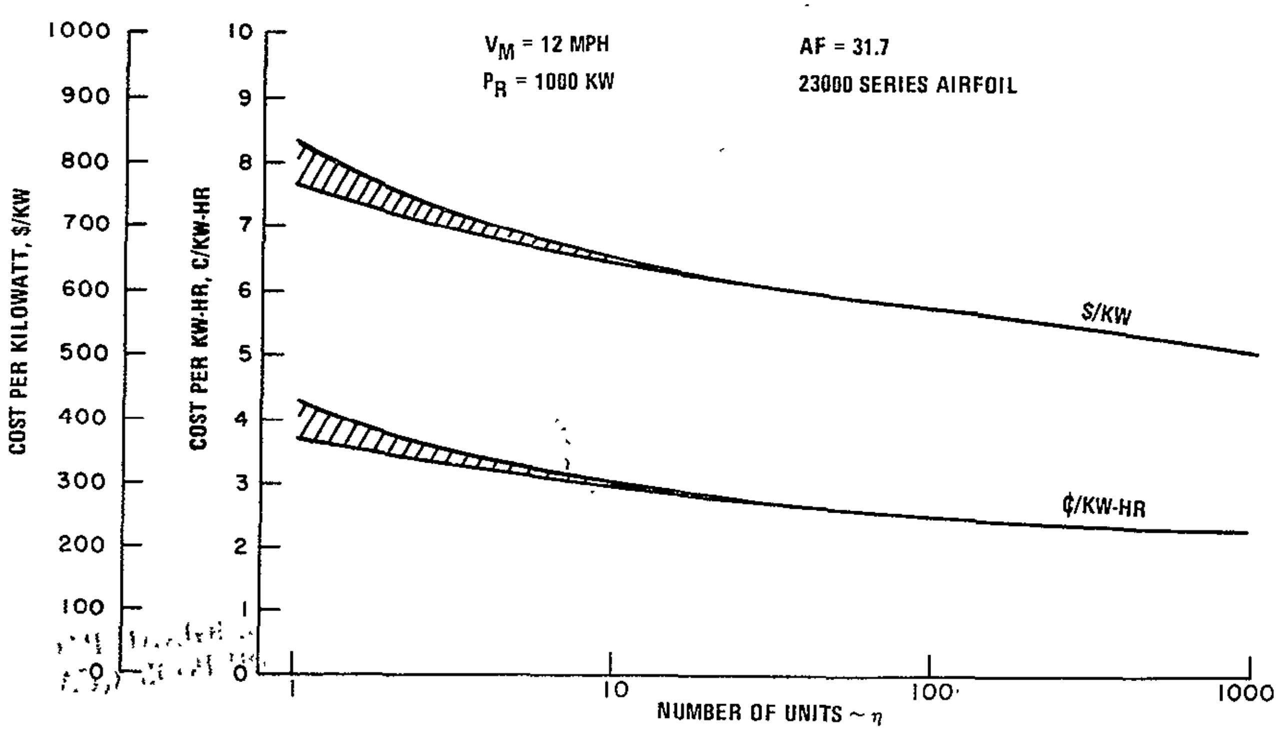


Figure 4-10. System Cost Sensitivity to Quantity

Cost effectiveness studies of other operational parameters indicated that the selection of cut-in and cut-out velocities is not critical. To raise the cut-in velocity slightly to insure adequate power generation output and prevent motoring in light winds will not significantly compromise cost effectiveness or annual energy output. The same is true for cut-out, because high winds are so infrequent.

Studies of altitude effects indicate that the variation in air density has a minor impact on system cost effectiveness. Although systems sited at sea level show a slight cost advantage because of the higher air density it is important to recognize that higher winds occur more frequently at higher altitudes. This aspect was not considered in this study, but, is being addressed by ERDA in other activities.

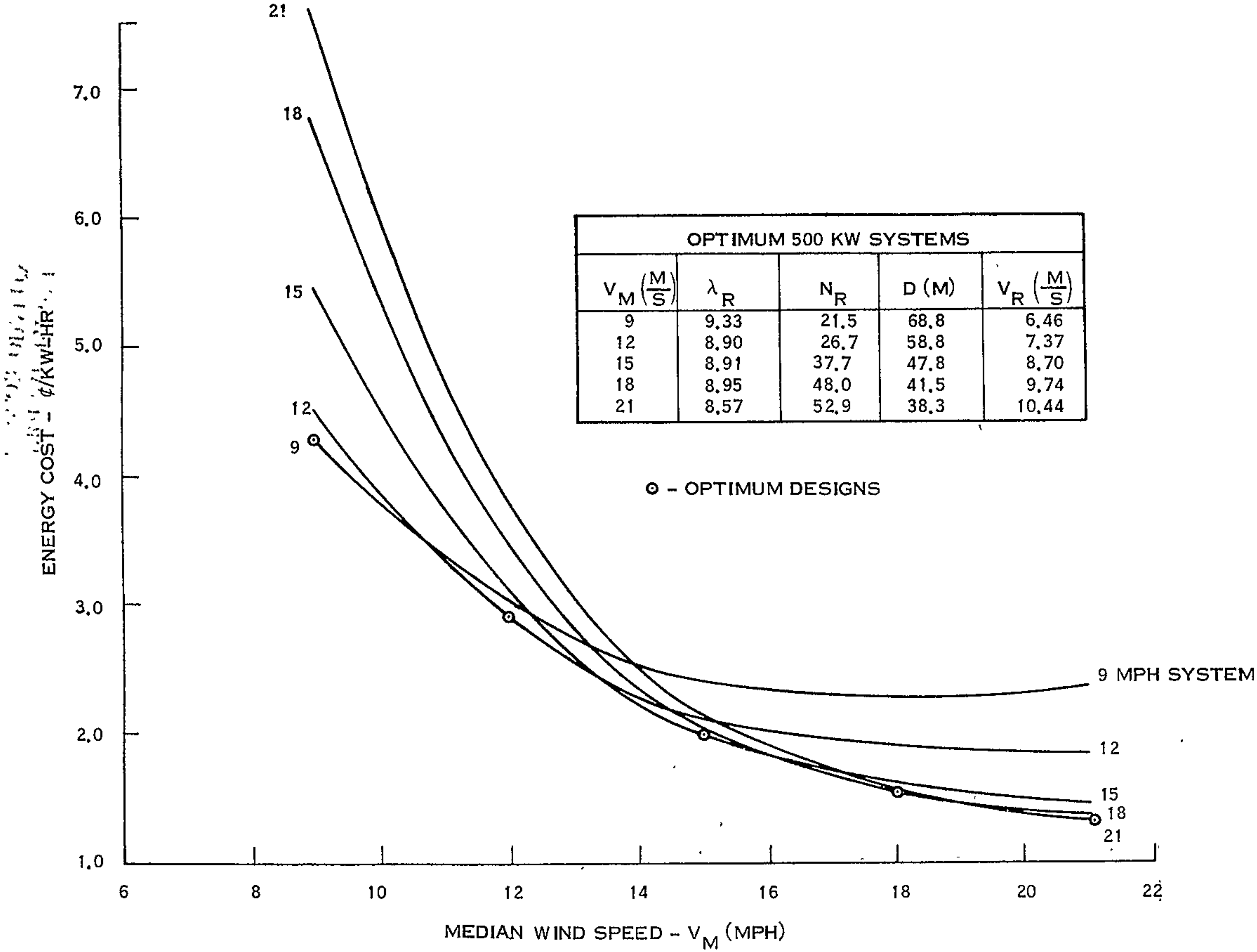
A study of the economic variables indicated that it was slightly more cost effective to use accelerated depreciation methods such as "sum-of-years digits" or "double-declining-balance" instead of straight-line-depreciation, if allowable. Bond interest rates will obviously affect system cost effectiveness but not significantly. System life is observed to have a slight effect on cost effectiveness, and indicates that the shorter the life (from 10 to 50 years) the most cost effective the system. This is true because of the manner in which the economics are determined, and if the system can be written-off in a shorter period of time, there are tax advantages and less capital amortization required. However, the length of time in which the system can be writtn-off must be accepted by the IRS.

4.3.2 OFF-OPTIMUM PERFORMANCE

The effects of environmental, operational and design constraints on the energy capture and the energy cost were evaluated in separate studies for off-optimum conditions. These studies are particularly significant in the practical application of large production orders of standardized WTG systems for utility needs for a variety of undefined or poorly defined wind sites. Also, off-optimum studies were used to perform in-depth rotor system cost tradeoffs, and to develop the effects of blade/WTG system dynamics on rotor design cost effectiveness.

4.3.2.1 Off-Optimum Operation of Optimized WTG Systems

In order to assess the WTG System performance in wind regions other than that for which they were designed, several computer runs were made by inputting the design characteristics of optimized systems as constants and by varying the wind regime inputs. The results of the study indicate that moderate off-design operation of the WTG systems incurs small energy cost penalties, but, that the energy cost penalties become much larger for systems, which are poorly mismatched between the actual and the design annual mean wind speed. An example of the effect of system mismatch with the windsite environment is shown in Figure 4-11 for the 500 KW system. Similar results for the 1000 KW system are shown in Figure 4-12. These results indicate that energy cost penalties caused by operating in off-design wind environments are more significant at lower power levels than at the higher power levels.



OPTIMUM 500 KW SYSTEMS

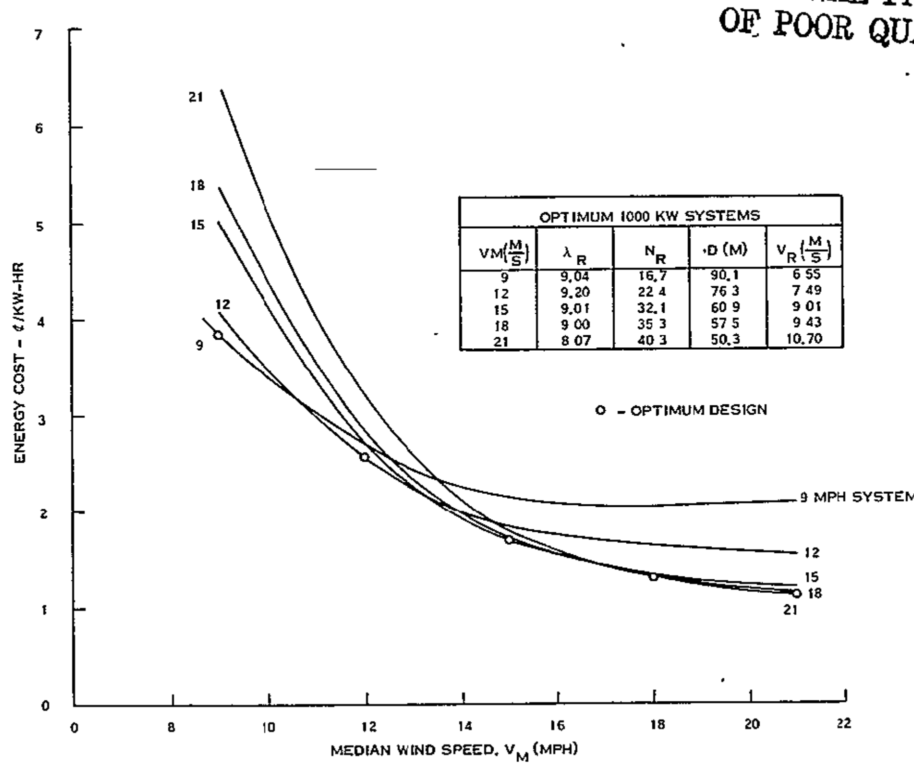
$V_M \left(\frac{M}{S}\right)$	λ_R	N_R	D (M)	$V_R \left(\frac{M}{S}\right)$
9	9.33	21.5	68.8	6.46
12	8.90	26.7	58.8	7.37
15	8.91	37.7	47.8	8.70
18	8.95	48.0	41.5	9.74
21	8.57	52.9	38.3	10.44

Figure 4-11. Effect of Median Wind Speed on Optimum Design Cases, 500 kW

4.3.2.2 Off-Design Optimization of WTG Systems

The previous section assessed the impact of using the identical WTG in various sites on power generation cost. However, since transmission and generators are catalog items, the major cost benefit from standardization is in the rotor.

To assess the use of a particular rotor blade design in several different wind regimes (to minimize tooling and design costs) the WTG optimization code was modified to hold blade diameter constant and search for the optimum power level (generator) and RPM (transmission) for operation in any given input wind regime. An example of these results are shown in Figures 4-13 and 4-14 for a 55.6 meter rotor diameter, which is an optimized rotor size for a 500KW system and a 12 MPH annual median wind speed. A continual decrease in energy cost with increasing annual median wind speed is observed for this rotor as shown in Figure 4-13, however, the energy costs also reflect an optimized power rating for the different annual median wind speeds as shown in Figure 4-14. It is interesting to note in Figure 4-14 that the optimum power rating for 55.6 m-diameter rotor at 12 MPH is not 500 KW, although the optimum diameter for a 500 KW system at 12 MPH is 55.6m. This seemingly contradictive statement is accurate but is difficult to comprehend due to the multi-dimensional aspects of optimization, which are explained graphically in Figure 4-15. As shown in the figure, D_2 is the optimum rotor diameter for power level P_{R1}, but, P_{R1} is not the optimum power level for D_2 (PR_2 is the optimum for D_2).



$V_M (\frac{M}{S})$	λ_R	N_R	D (M)	$V_R (\frac{M}{S})$
9	9.04	16.7	90.1	6 55
12	9.20	22 4	76 3	7 49
15	9.01	32.1	60 9	9 01
18	9 00	35 3	57 5	9 43
21	8 07	40 3	50.3	10.70

Figure 4-12. Effect of Median Wind Speed on Optimum Design Cases, 1000 kW

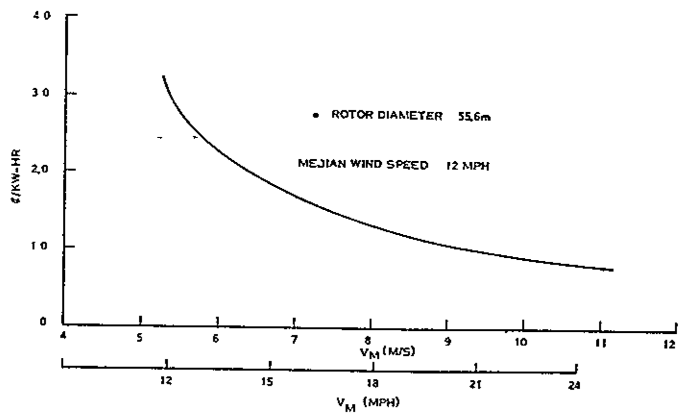


Figure 4-13. Power Generation Cost vs. Median Speed for Optimum, Constant Rotor Systems

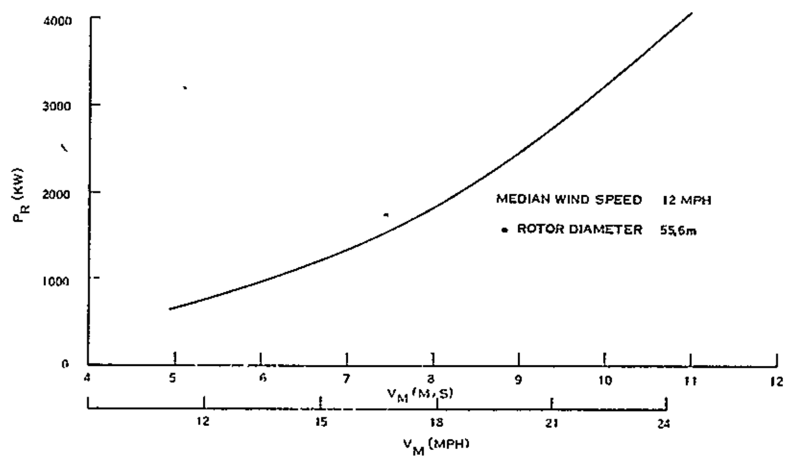


Figure 4-14. Optimized Power Level vs. Median Wind Speed for Constant Rotor Systems

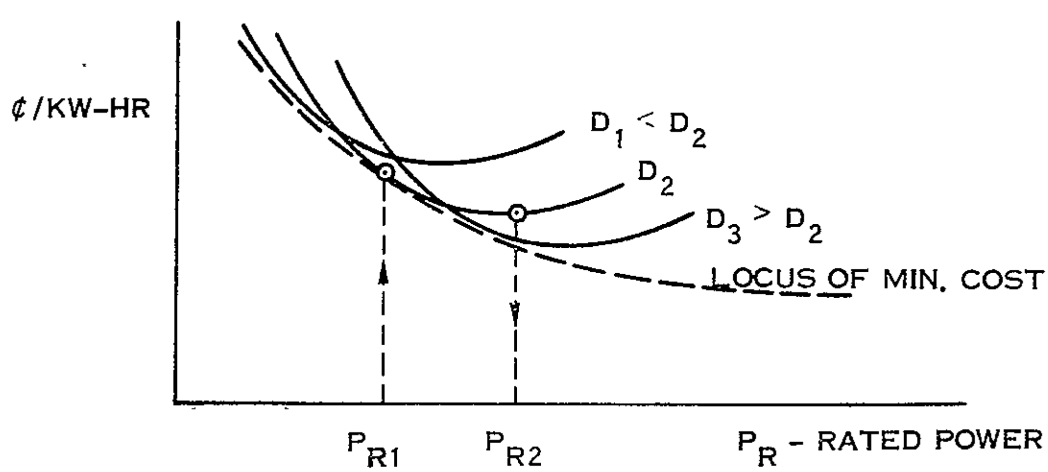


Figure 4-15. Effect of Power Level Upon Optimum Rotor Diameter

SECTION 5.0

PRELIMINARY DESIGN

SECTION 5.0

PRELIMINARY DESIGN

This section describes the two Preliminary Designs in their entirety. Section 5.1 discusses the system design philosophy, 5.2, the system operation, and 5.3 provides a detailed description of the WTG subsystems.

The primary emphasis in developing these designs was placed upon:

- minimizing capital costs
- minimizing maintenance costs
- maximizing system energy capture

Efforts to minimize capital costs were concentrated primarily in two areas: the utilization of commercially available hardware and the use of a control system which has the capability to avoid undesirably high wind loading conditions upon the WTG.

The overall system design, material selections and equipment locations have been formulated with the idea of permitting maintenance in an efficient, low cost manner. Examples of some key maintenance features are the use of bearings with split housings, the vertical mounting of the over-under parallel shaft gearbox speed increaser, the use of flexible couplings with removable center sections and the inclusion of a rotor locking flange which permits access to the complete mechanical power transmission without removal of the rotor subsystem.

The third point, maximization of energy capture, begins with the selection of the optimum design operating conditions for the assumed wind conditions. Thereafter, a control system which ensures mechanical and electrical stability under steady state and gusting wind conditions (within the limits of cut-in and cut-out) is required in order to obtain the most energy possible at that site. The control system described in Section 5.3 is designed to perform these tasks.

5.1 SYSTEM DESIGN DESCRIPTION

The section describes the rationale which resulted in the development of the two Preliminary Designs and an overall description of these systems. A detailed discussion of each of the subsystems within the WTG's is presented in Section 5.3.

The first step in the Preliminary Design task was the establishment of the system ground rules. Using these assumptions, the design optimization code (Appendix 8.5) was run to select the operating conditions for minimum power generation cost (¢/KW-HR). The results of the system optimization provided the basic parameters from which to initiate the Preliminary Design: RPM, blade diameter and rated velocity ratio.

Next, the results of the code were used to determine the transmission and generator requirements. The rotor subsystem was also designed based on the aero/structural requirements consistent with the computer code design point. Design of the rotor subsystem is not straightforward and requires close integration with other subsystem design considerations. Effects of wind shear, tower shadowing, rotor/tower dynamic coupling as well as maintenance problems have a direct impact upon the rotor subsystem design. In addition, the variable pitch mechanism, which is critical to the performance and safety of the WTG must be integrated within this design.

Still another trade-off to be performed between the rotor and the remainder of the system is the amount of blade coning and the degree of inclination on the hub/main shaft assembly.

After the required iterations, the proper trade-offs between the various subsystems can be made. Throughout this entire process the control system requirements must also be determined. The result of these interactions, previously depicted in Figure 1-2, are the Preliminary Designs discussed below.

5.1.1 ASSUMPTIONS

The basis for the Preliminary Designs was provided by the results of the Conceptual Design and Parametric Analyses Tasks as interpreted by both NASA and General Electric. The assumptions made in terms of the design conditions and component selection for both the 12 MPH and 18 MPH systems are shown in Table 5-1. The wind duration curves associated with the 12 and 18 MPH median wind sites have been presented in Section 2.2.

In performing the system calculations a tower shadowing model for a truss tower was used in all cases. At this time there is no analytical expression available to define the shadowing effect of geometries representative of concrete towers. However, it should be noted that the cyclic loading associated with wind shear dominates the impact of the tower shadowing effect of truss towers for the cases studied.

5.1.2 SYSTEM DESCRIPTION

The first step in defining the Preliminary Designs was to run the design optimization code to determine the minimum cost operating conditions. Results of this effort are shown in Table 5-2. The design conditions selected are a trade-off between minimizing capital and operating costs, and maximizing energy capture. Both designs utilize rotor diameters approaching 200 feet in diameter which is consistent with results of the Parametric Analyses. Large blade diameters allow lower rated velocities and higher energy capture. The capacity factors for the two machines are 0.42 and 0.51 for the 500 kW and 1500 kW Systems, respectively. As shown in the table, the rated velocity ratios are near the design value of 10. Another factor which affects both capital costs and energy is the selection of the cut-out velocity. Higher cut-out velocities require more expensive blades, however, the annual electrical energy production also increases. The cut-in velocity is determined by the mechanical and electrical losses in the system. The design conditions selected provide the least cost power generation within the framework of the economic and wind site model constructed.

TABLE 5-1. PRELIMINARY DESIGN ASSUMPTIONS

MEAN WIND, MPH	12.	18
POWER LEVEL, KW	500.	1500.
BLADE CLEARANCE HEIGHT, M	15.2	15.2
TERRAIN FACTOR	0.167	0.167
AIR DENSITY, kg/m^3	1.22	1.22
CONFIGURATION		
ROTOR LOCATION	DOWNWIND	
EQUIPMENT LOCATION	ATOP TOWER	
NUMBER OF ROTORS	1	
NUMBER OF BLADES	2	
OPERATING MODE	CONSTANT RPM	
POWER REGULATION TECHNIQUE	BLADE PITCH CONTROL	
SUBSYSTEM SELECTION		
ROTOR, AIRFOIL	PROPELLER, 23012	
TRANSMISSION	GEARBOX	
GENERATOR, RPM	SYNCHRONOUS, 1800	
TOWER	CONCRETE OR TRUSS	
CONTROLS	MICROCOMPUTER	

TABLE 5-2. PRELIMINARY DESIGN SYSTEM CHARACTERISTICS

POWER RATING, KWE	500	1500
ANNUAL ENERGY, KW-HRS	1.88×10^6	6.62×10^6
CAPACITY FACTOR	0.42	0.51
ROTOR DIAMETER, M (FT)	55.8 (183)	57.9 (190)
RPM	29	40
RATED VELOCITY, M/S (MPH)	7.27 (16.3)	10.1 (22.5)
λ RATED (λ DESIGN = 10)	9.0	9.5
TIP SPEED, M/S (FT/SEC)	84.7 (278)	121 (398)
CUT-IN VELOCITY, M/S (MPH)	3.54 (7.92)	5.11 (11.4)
λ CUT-IN	18.5	18.4
CUT-OUT VELOCITY, M/S (MPH)	17.9 (40)	22.3 (50)
λ CUT-OUT	3.67	4.18
HOURS ABOVE VCI	6257	6568
HOURS AT RATED	2067	2718

An exterior view of the upper portion of the 1500 kW preliminary design is shown in Figure 5-1. In the illustration the rotor subsystem can be seen at the left. The hub portion is about 25 feet in diameter while the blade chord and blade root thickness are 12 and 5 feet, respectively. The mechanical power transmission and generator are housed within the nacelle; at the right of the nacelle is an instrumentation boom which points into the wind. The entire nacelle is mounted atop the pintle which contains the azimuth drive mechanism. As seen in the illustration, the main shaft is inclined 6 degrees to the horizontal in order to reduce the rotor overhang distance required for tower clearance. Also of importance is the fact that the blades are coned at 3 degrees which lowers the steady state shank bending stresses and assists in maintaining the desired blade/tower clearance.

The angle of inclination and the cone angle have been selected so as to achieve the objectives stated without seriously compromising the effective blade diameter. The design of the nacelle permits access to all of the critical components by means of hinged panels. A walkway around 3 sides of the nacelle permits adequate working space for servicing. In cold climates, the walkway would be replaced by a larger nacelle so that maintenance procedures could be conducted in a controlled environment.

A view of the 1500 kW WTG with the nacelle cover removed is provided in Figure 5-2. In this illustration the main shaft, supported by two bearings, is shown. Separating the main shaft from the gearbox transmission is a flexible coupling. The coupling will compensate for small misalignments between the main shaft and the gear box which are present at assembly or developed over the life of the unit. Each of the bearings has a split housing which permits ease of disassembly; in addition, the flexible coupling has a removable center section. These design features allow complete disassembly of the main shaft and its associated parts. It is noted that the inclusion of a rotor locking flange permits this to be done without the need to remove the rotor subsystem.

The transmission is oriented such that the high speed shaft is on top where it is accessible for maintenance. Attached to the high speed shaft, at the left of the transmission, is a hydraulic disc brake which is capable of bringing the system to an emergency stop in 20 seconds.

Another flexible coupling joins the transmission high speed shaft to the generator shaft. Located on the generator shaft is a torque sensor. The role of the torque sensor in the control system is described in Section 3.2.

Vertical orientation of the transmission results in a favorable position for the generator. In the design shown, the generator and associated electrical leads and auxiliaries are easily accessible. The generator selected is a 1800 RPM, 4160 V., synchronous unit*, equipped with an exciter and voltage regulator.

* Additional preliminary design was performed with a 1800 RPM, 4160 V, 500 kW, induction unit equipped with a 300 KVAR capacitor bank and 4160 volt capacitor bank contractor.

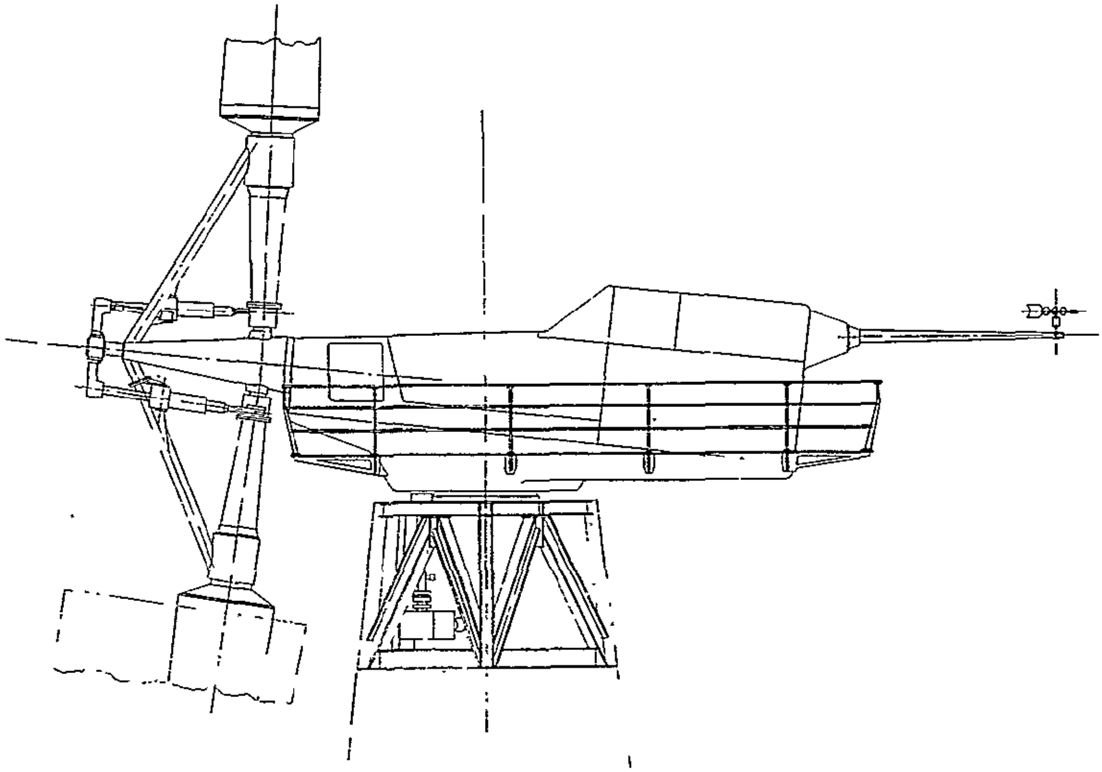


Figure 5-1. Exterior View of 1500 kW Unit

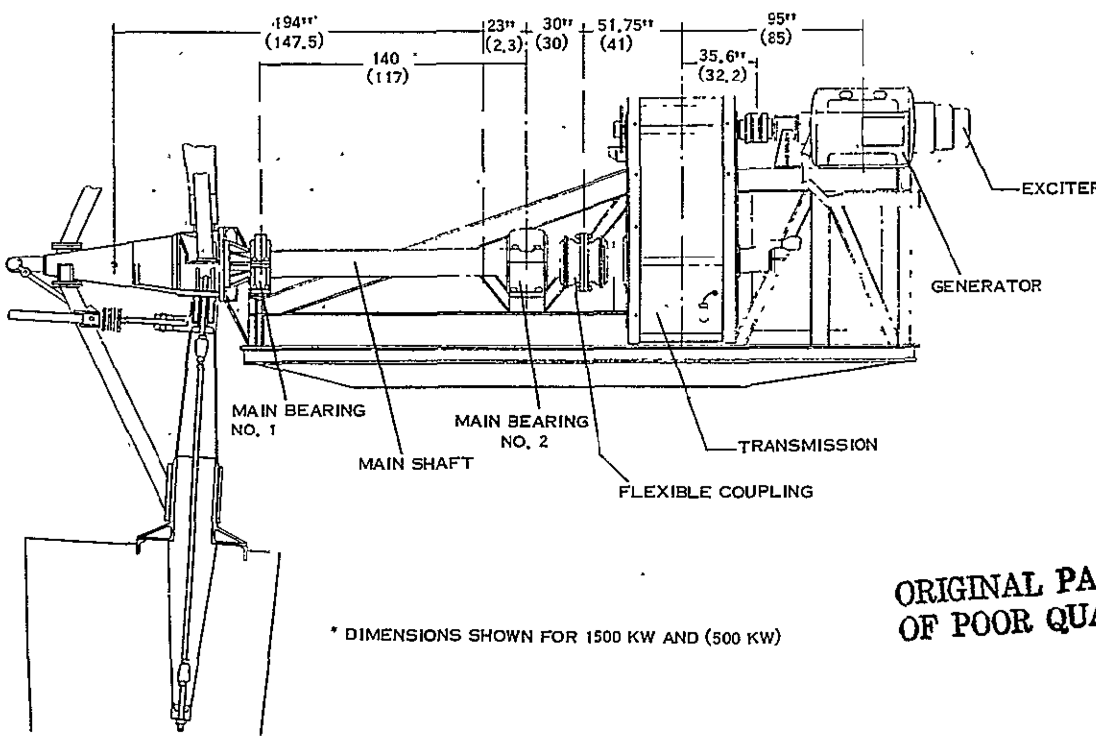


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Figure 5-2. Preliminary Design Layout

The rotor, mechanical power transmission and generator are rigidly attached to a bedplate which transfers the loads into the pintle structure. The layout of these subsystems attempts to reduce the bending moments about the pintle due to the gravity and aerodynamic loads.

Another aspect of the design shown in Figure 5-2 is the variable pitch change mechanism. The worm gear actuators are mounted in a pivot bracket on the hub out-of-plane brace and the output shaft is connected to the blade trunnion arm on the blade. Each of the actuators are connected by two angle gearboxes and a shaft with universal joints to the T gearbox mounted to the forward end of the hub. The pitch change input shaft extends from the T box through the hub, the main drive shaft and the gearbox shaft to the pitch change power unit which is shown mounted to the gear box housing. Operation of this high reliability, fail-safe mechanism is described in detail in Section 5.3.1.

The control system for the WTG is shown schematically in Figure 5-3. This system is comprised of a microcomputer, sensors and actuators; the microcomputer and the isolated sensors are powered by an uninterrupted power supply (UPS). The UPS system, which consists of a battery coupled to a trickle charger, and the use of a redundant microcomputer provide the necessary reliability to the control system. The principal functions of the control system are to provide for system start-up, shutdown, nacelle turning, power regulation, mutual protection of the network and the WTG, data computation and control of the system during maintenance and emergency modes, and communications with a remote headquarter site.

All instrumentation and electronics that are not required to be aloft are protected in a structure at the base of the WTG tower. The master control for the operational logic is provided by a microcomputer in conjunction with data acquisition electronics.

The computed performance and status data obtained from the microcomputer is suitable for transmission over a leased phone line. The computer also constructs a coded signal as an input to an automatic call unit which will automatically dial up a predetermined telephone number when conditions warrant outside attention to the WTG. Conversely, data can be obtained from the WTG by dialing the modem at the WTG site and upon achieving proper contact, can acoustically couple his phone set to a teletypewriter. Once the hook-up is completed, the agent may, from his remote headquarters site, command and interrogate the WTG to determine the status of the WTG and the nature of maintenance, if any, required. A maintenance crew arriving at the WTG site may acoustically couple a portable teletypewriter to the microcomputer and in this manner exercise the WTG and determine additional information on the condition of the WTG. Of course, it is also possible to connect the WTG microcomputer via phone line, to a larger general purpose computer such that the output of several WTG's can be integrated into the utility network dispatcher's decisions relative to the most efficient manner of regulating supply to demand.

The 500 kW and 1500 kW systems can be supported by either concrete or truss towers. Reinforced concrete towers offer more aesthetic appeal than truss towers, however, their cost is more sensitive to the site consitions and location. It is recommended that concrete towers constructed of precast sections be used where normal conditions and site accessibility exists.

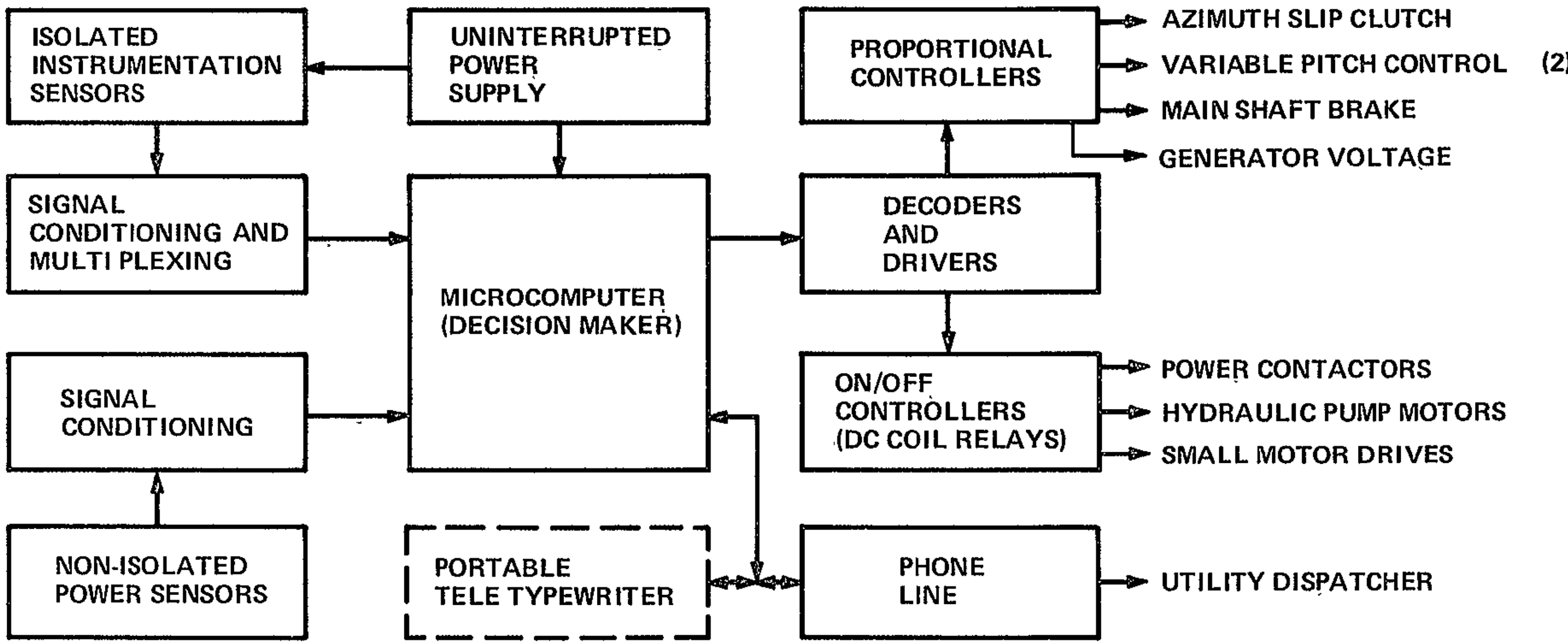

Figure 5-3. Control Subsystem Schematic

A considerable effort has been made to design the towers on a total system basis in order to minimize the rotor/tower dynamic load factors. It is recommended that this design approach be pursued in the Final Design so as to reduce the tower and rotor blade costs. The justification for treating the rotor/tower design jointly is supported by Reference 5-1 in which arbitrary stiffening of the tower was shown to be an inadequate approach.

5.1.3 SYSTEM WEIGHT SUMMARY

In a very basic sense, the weight of the system is important since, as for most industrial products, it will directly affect cost. For a system such as the WTG, the weight of individual subsystems is even more important since undesirable dynamic interactions (related to weight) can occur between subsystems.

For the reasons cited above it is desirable to minimize the overall system weight. Specifically, the rotor/tower dynamic analysis performed showed that it was advantageous to reduce the nacelle weight since higher nacelle weights caused a reduction in the tower natural frequency. Also of importance is the effect of rotor weight upon the nacelle and tower designs. Higher rotor weights result in a necessarily stronger, heavier and more costly mechanical power transmission to accommodate the increased load. As a minimum, higher rotor weights will also decrease the tower natural frequency.

Table 5-3 provides a weight summary for both the 500 kW and 1500 kW WTG systems. Total weight aloft for the 500 kW and 1500 kW systems is approximately 40 tons and 62 tons, respectively. The rotor/nacelle configuration was designed such that the static loading provided a zero moment about the azimuth bearing.

The rotor subsystem weights include: hub, blades, and pitch change mechanism. The higher weight of the 1500 kW rotor subsystem is directly attributable to the larger loads upon the blades of this unit. Further refinements of the blade design could result in some weight reduction.

Some conservatism may may also be present in the weight of the bedplate/pintle subassembly. The weights for these structures were calculated on a preliminary basis. During a Final Design effort, a detailed computer analysis would probably allow some relaxation in the safety factors assumed.

5.1.4 SYSTEM LOAD SUMMARY

The mechanical system components were sized according to the rotor induced loadings, which were a consequence of several overall system decisions. Despite the relative dynamic advantages of a three-bladed rotor, or a teetered hub/two-bladed rotor over the rigid hub/two-bladed rotor, the latter approach was chosen on the basis of cost. This was made possible by the use of a control system and other design accommodations, which limited otherwise severe loadings for the rigid hub/two-bladed rotor subsystem. Analysis and a clearer definition of operating philosophy lead to a definition of load criteria. Three main criteria were established:

TABLE 5-3. WEIGHT SUMMARY

POWER LEVEL	500 KW	1500 KW
ROTOR	27,000	35,000
GEAR BOX	24,000	46,000
GENERATOR, SYNCHRONOUS (GENERATOR, INDUCTION)	4,500 (4,700)	11,200
MAIN SHAFT	4,000	5,000
M.S. BEARINGS	2,300	3,500
COUPLINGS	2,700	5,200
BEDPLATE/PINTLE	15,000	24,000
SUB TOTAL	79,500 (79,700)	129,900
TOWER -		
TRUSS (CONCRETE)	86,000 (450,000)	118,000 (650,000)
TOTAL	165,500 (529,500) LBS	247,900 (779,900) LBS

(1) Continuous loads at the rated wind velocity, continuous being defined as: steady or cyclic loadings over a long period of operation.

(2) Intermittent loads due to an instantaneous doubling of rated wind velocity. Those loads were assumed to occur on an occasional basis and were based on a preliminary gust model.

(3) Limit loads which were a consequence of a 120 mph storm condition. In this case the blades are stowed in the 3-9 o'clock position and the primary loading is structural drag.

These loads are shown in Figure 5-4 and tabulated in Table 5-4.

Other than the rotor, the mechanically loaded component groups would be the mainshaft and transmission, pintle structure, and tower. Each of those subsystems experiences some loadings not common to the others. The mechanical power transmission system, for instance, was designed around the loads shown in Table 5-5. The main shaft and gearbox shaft designs are more sensitive to the cyclic nature of the loadings so that the ranges are shown in parenthesis. Shock factors of 1.4 were applied to applicable loadings in Table 5-5, however, it is anticipated that this is a conservative assumption, since the control system will limit the magnitude of loads due to wind gusting. The pintle structure and tower were also designed based on the load schedule in Table 5-4. Some of the moments at the

TABLE 5-4. SYSTEM LOAD SCHEDULE

LOAD SCHEDULE

UNITS: # & FT-#
FX1, FZ2, WT = f(TOWER DESIGN)

	T	MZ1	FZ1	MY1	FY1	MZ2	T1	We	MZ3
CONTINUOUS LOADS									
500KW	19^k	400^k	1.9^k	19^k	1.9^k	19^k	134^k	$.73^k$	100^k
1500KW	45^k	400^k	4.5^k	45^k	4.5^k	45^k	290^k	121^k	134^k
INTERMITTENT LOADS									
500KW	32^k	600^k	3.2^k	32^k	3.2^k	32^k	425^k	73^k	100^k
1500KW	75^k	600^k	7.5^k	75^k	7.5^k	75^k	1000^k	121^k	134^k
LIMIT LOADS									
500KW	*6.7^k	0	$\varphi 18^k$	75^k	0	0	0	73^k	0
1500KW	*6.7^k	0	$\varphi 18^k$	75^k	0	0	0	121^k	0

* DLF OF X 1.35 DOES NOT APPLY
φ WIND LOADING ON NACELLE

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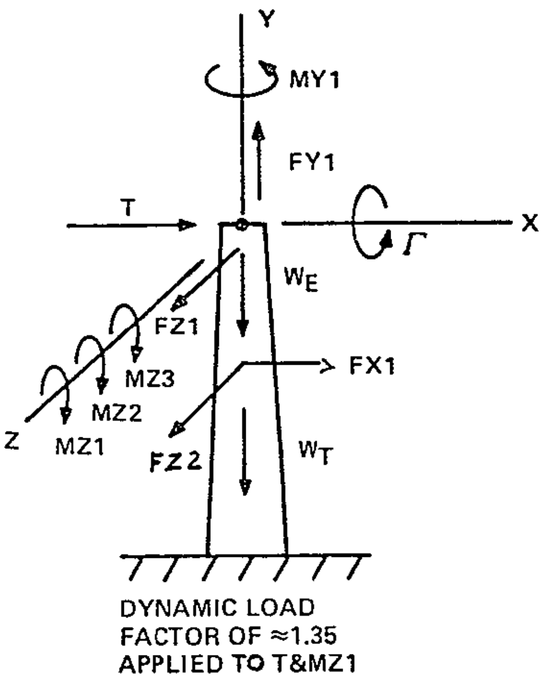

T – THRUST
MZ1 – MOMENT (PART OF THRUST FORCE-COUPLE
FZ1 – SIDE LOAD
MY1 – MOMENT (PART OF FZ1 FORCE-COUPLE)
FY1 – VERTICAL LOAD
MZ2 – MOMENT (PART OF FY1 FORCE-COUPLE)
Γ – TRANSMITTED TORQUE
FX1 – TOWER WIND LOADING
FZ2 – TOWER WIND LOADING
W_E – UPPER EQUIPMENT WEIGHT
W_T – ABOVE TOWER STRUCTURE WEIGHT
MZ3 – GYRO MOMENT DUE TO YAW

Figure 5-4. Illustration of System Loads

TABLE 5-5. MECHANICAL POWER TRANSMISSION LOADS

POWER LEVEL	500KW	1500 KW	
INPUT TORQUE	134(1 ± 1)	317(1 ± 1)	KLB-FT
FORCES - G LOAD	27.6	35.1	KLB
WIND SHEAR	1.9(1 ± 0.7)	4.5(1 ± 0.7)	KLB
THRUST	19.0(1.4 ± 0.7)	44.6(1.4 ± 0.7)	KLB
MOMENTS - AZIMUTH CHANGE	100	134	KLB-FT
- WIND SHEAR	400(1.4 ± 0.5)	400(1.4 ± 0.5)	KLB-FT
OUTPUT TORQUE	2.1(1 ± 1)	6.2(1 ± 1)	KLB-FT
GEARBOX SPEED INCREASE RATIO	61.9	44.3	

tower center line do not apply directly for the design of the pintle structure. It is noted that a dynamic load factor of 1.35 was assumed applicable to T and MZI, the primary lateral bending loads (See Figure 5-4). This load factor results from dynamic considerations and could be improved with further analysis and dynamic tuning during the Final Design effort.

5.1.5 SYSTEM COST SUMMARY

This section describes the anticipated costs of the Wind Turbine Generators developed during the Preliminary Design Task. In estimating the sales price of these units it has been assumed that WTG's are a mature commercial product; accordingly no development or "learning factor" expenses have been included. The scenario used to calculate the WTG costs are:

- A systems contractor is responsible for the complete design, fabrication and check-out of the WTG's
- An order for 100 identical units has been received with specifications similar to units built in the past
- The 100 units will be installed in a 3 year time period in one geographic locale
- The energy generation costs are for power delivered at the tower base - no power distribution costs are included
- Costs are in 1975 dollars

Unit costs for the 500 kW and 1500 kW WTG's are shown in Table 5-6. Program Management, Final Design and Systems Integration costs have been amortized over 100 units; the costs shown for these categories include labor and overhead where the labor is assumed to be equal to the overhead cost.

Responsibility for the technical direction, program schedule, purchasing and financial control are included under Program Management. The Final Design cost reflects the assumption that some effort will be required for any production run. While in some cases this may entail only a limited task to modify an existing design, significantly different site conditions may require substantial deviations from a previous design. The systems integration cost includes expenses related to the establishment of the system requirements, configuration control, development of the site erection plan and interfacing with the utility company.

Equipment costs, including the systems contractor cost (but not fee) represent vendor quotations for established commercial products except for the rotor. The rotor cost was estimated by Hamilton Standard and resulted, in part, from their discussions with Hercules, a manufacturer of filament wound products. As shown in the table, the rotor and transmission are by far the major system cost contributors; together they comprise 50 to 60 percent of the system cost.

TABLE 5-6. SYSTEM COST SUMMARY

SYSTEM TASK	500 kW, WTG 12 MPH MEDIAN WIND	1500 kW, WTG 18 MPH MEDIAN WIND
PROGRAM MANAGEMENT	2,000	2,000
FINAL DESIGN	1,500	1,500
SYSTEMS INTEGRATION	2,500	2,500
EQUIPMENT COSTS		
ROTOR	154,000	185,000
TRANSMISSION	100,000	162,000
ELECTRICAL	37,400	60,700
CONTROLS	13,600	13,600
TOWER (concrete)	39,300	55,900
BEDPLATE/PINTLE	19,100	36,400
SUBTOTAL	363,400	513,600
ASSEMBLY COSTS		
NACELLE	8,600	8,600
SITE DELIVERY	5,600	9,300
ERECTION	14,700	14,700
SYSTEM CHECK-OUT	2,500	2,500
SUBTOTAL	31,400	35,100
SITE COSTS		
LAND	1,500	1,500
CONTROL BUILDING	3,500	3,500
TOTAL DIRECT LABOR & MATERIAL	405,800	559,700
GROSS OPERATING MARGIN	60,870	83,955
TOTAL COST	466,670	643,655

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First Unit Costs

The costs of Table 5-6 reflect estimates for a "mature commercial product" environment. Costs of early units will be substantially higher due to many factors. Recently, General Electric contracted with NASA-Lewis to construct and install two 1500 kW WTG power systems (Mod-1). The engineering time and purchased parts quotations of the second unit, devoid of recurring costs, was employed to generate Table 5-6a, which is a representative first unit cost summary for the 1500 kW WTG system. Note that aircraft industry average hourly, overhead, general and administrative and fee rates were used in Table 5-6a rather than those particular to the specific contractors. Table 5-6b presents a corresponding first unit cost estimate for the 500 kW WTG.

The total cost of Table 5-6 (for the 1500 kW unit) can be related to the $2.633 million first unit cost by means of the learning curve to establish the cumulative quantity of units necessary to achieve the lower cost. A 90 percent learning curve is quite conservative when compared with historical data on items with similar technological complexity. This means that for each doubling of cumulative production the average unit cost (for all units) will be reduced to 90 percent of the previous value. Thus if the first unit costs one hundred dollars, the first two will average ninety dollars, the first four will average 81 dollars etc.

Solving for the average cost of 100 unit production runs for a first unit cost of $2.633 million and 90 percent learning yields:

Units	1500 kW WTG Average Cost
1 - 100	$1308 K
101 - 200	1046
201 - 300	966
↓	↓
3401 - 3500	650
3501 - 3600	647
3601 - 3700	644

Thus, if a 90 percent learning rate is achieved the total cost of Table 5-6 will be reached when cumulative 1500 kW unit production is approximately 3600 to 3700 units. This is consistent with the mature commercial product assumption and well within potential WTG implementation possibilities.

There are numerous factors that drive the learning curve, including:

1. Production automation, with associated low overhead burden.
2. Integration of manufacturing operations -- fewer subcontractors.

FOLDOUT FRAME 1

FOLDOUT FRAME 2

TABLE 5-6a

TYPICAL PROTOTYPE PRODUCTION UNIT COST BREAKDOWN - 1500 KW WTG

	PRIME CONTRACTOR				SUBCONTRACTORS					PRIME CONTRACTOR BURDEN	TOTAL
	LABOR		PURCHASED PARTS	OTHER	LABOR		PURCHASED PARTS	OTHER	BURDEN		
	HOURS	($K)1	($K)2	($K)3	HOURS	($K)1	($K)2	($K)3	($K)4	($K)4	($K)5
SYSTEM TASK											
PROGRAM MANAGEMENT	1000	22		2						9	33
FINAL DESIGN/ ENGINEERING SUPPORT	1000	22		4						10	36
SYSTEMS INTEGRATION	3000	67		18						32	117
EQUIPMENT COSTS											
ROTOR/BLADE					10,600	237	223	23	181	249	913
TRANSMISSION			170							64	234
ELECTRICAL } CONTROLS }			217							81	298
TOWER (STEEL)			99							37	136
PINTLE			15							6	21
SUBTOTAL			501			237	223	23	181	437	1602
ASSEMBLY COSTS											
NACELLE			53				24		9	32	118
SITE DELIVERY			13							5	18
ERECTION							140	21	61	83	305
SYSTEM CHECK-OUT	2200	49						8	3	22	82
SUBTOTAL		49	66				164	29	73	142	523
SITE COSTS											
LAND PREPARATION	2000	45			4,500	101	33		50	86	315
CONTROL BUILDING			5							2	7
TOTAL		205	572	24		338	420	52	304	718	2633

NOTES: 1) Based on average Aircraft Industry labor rate of \$10.80/hr., OH rate of 107% used for all prime contractor labor costs.

2) Includes catalog/off-the-shelf type items/raw materials -- suppliers OH, G&A, margin burdens included, but not definable.

3) Includes travel and living, publications, computer, etc. -- direct cost.

4) Based on average Aircraft Industry G&A/Matl. Hdlg. rate of 25% and assumed margin of 10%.

5) Sum of all ($K) columns.

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TABLE 5-6b

TYPICAL PROTOTYPE PRODUCTION UNIT COST BREAKDOWN - 500 KW WTG

	PRIME CONTRACTOR				SUBCONTRACTORS					PRIME CONTRACTOR BURDEN	TOTAL
	LABOR		PURCHASED PARTS	OTHER	LABOR		PURCHASED PARTS	OTHER	BURDEN		
	HOURS	($K)[1]	($K)[2]	($K)[3]	HOURS	($K)[1]	($K)[2]	($K)[3]	($K)[4]	($K)[4]	($K)[5]
SYSTEM TASK											
PROGRAM MANAGEMENT	800	18		1						7	26
FINAL DESIGN/ ENGINEERING SUPPORT	800	18		3						8	29
SYSTEMS INTEGRATION	2,000	45		14						22	81
EQUIPMENT COSTS											
ROTOR/BLADE					7,400	165	157	16	127	174	639
TRANSMISSION			105							39	144
ELECTRICAL, CONTROLS			149							56	205
TOWER (STEEL)			69							26	95
PINTLE			8							3	11
SUBTOTAL											
ASSEMBLY COSTS											
NACELLE			39				18		7	24	88
SITE DELIVERY			8							3	11
ERECTION							91	14	39	54	198
SYSTEM CHECK-OUT	1,500	34						8	3	16	61
SUBTOTAL											
SITE COSTS											
LAND PREPARATION	1,200	27			2,700	60	20		30	51	188
CONTROL BUILDING			5							2	7
TOTAL		142	383	18		225	286	38	206	485	1,783

NOTES:

1) Based on average Aircraft Industry labor rate of $10.80/hr., OH rate of 107% used for all prime contractor labor costs.

2) Includes catalog/off-the-shelf type items/raw materials -- suppliers OH, G&A, margin burdens included, but not definable.

3) Includes travel and living, publications, computer, etc. -- direct cost.

4) Based on average Aircraft Industry G&A/Matl. Hdlg. rate of 25% and assumed margin of 10%.

5) Sum of all ($K) columns.

3. Improved processes.
4. More cost effective material utilization.
5. Specification improvement to eliminate over-design.

Different cost items will be affected to different degrees as an overall learning curve is realized. Items such as the transmission, which are well developed and essentially state-of-the-art will operate at a low learning rate in comparison to an item such as the rotor. Program management costs and associated overhead and administrative expenses will decrease rapidly as a production environment is reached -- another example of rapid learning.

The first units will use towers of proven steel truss construction designed to stringent load requirements. Cost improvements in production may be achieved either through design refinements in the steel tower or a change to a different concept, such as the concrete tower concept of Table 5-6. Switchgear will be used in the first units at considerably higher cost than the microprocessors assumed to be available in a WTG production time frame.

Due to the multiplicity of learning rates inherent in the overall WTG rate, there is much less error in predicting the future cost of the unit than in predicting the cost of any single element. Thus, although the costs of each item in Table 5-6 present the best estimate under the stated ground rules, the item costs should be given considerably less importance than the total WTG cost. The reader should not, for example, assume a lower cost for a particular item and adjust the total cost accordingly. It is conceivable that the Table 5-6 total costs might be achieved with a different component mix, for example a system without a transmission.

Finally, it should be appreciated that the costs of Table 5-6 are at a point in time within the production history of the device. The first unit built will cost \$2.633 million in 1975 dollars. Learning curve theory projects a steady decline in WTG system cost (in 1975 dollars) as cumulative production increases. Application analysis for early WTG systems should use the Table 5-6a and 5-6b cost data as a starting point rather than the mature product estimates of Table 5-6.

Fabrication of the system is assumed to proceed in the following manner:

- Assembly of the nacelle at the system contractors plant
- Erection of the tower from pre-assembled (or pre-cast sections by local engineering contractors
- Delivery of the assembled nacelle to the site
- Delivery of the rotor direct to the site
- Placement of the nacelle and rotor atop the tower
- System Check-out

The purpose of the system check-out is to ensure that all electrical connections made at the site have been done correctly and that all aspects of the design are functioning properly. The latter determination will be made by microcomputer self-tests of the WTG. The electrical check-out will consist of AC and DC insulation and megohm tests as well as continuity checks of all cabling connected at the site.

A major aspect of the cost estimate is the projected profit margin required by the systems contractor. A review of General Electric's operating margins over the last decade indicates a 10 to 15 percent figure before taxes and depreciation. Therefore, a 15 percent operating margin is shown in Table 5-6.

System costs are calculated to be $935/kW for the 500 kW unit and $430/kW for the 1500 kW unit in the 12 and 18 MPH median wind sites, respectively.

A detailed economic model, shown in Figure 5-5 was used to calculate the power generating costs for these 2 systems. The following assumptions were made in this model:

- Straight line depreciation over the system life (30 years)
- Capitalization method - 50 percent bonds, 50 percent equity
- Cost of capital - 9 percent interest, 11.5 percent return on equity
- 48 percent tax rate on profits
- Use of a sinking fund to repay bondholders at end of 30 years
- Additional capital costs for rotating parts and maintenance costs included over the system life

Based on these ground rules, the power generating costs are 4.04 and 1.57¢/kW-HR for the 12 and 18 MPH systems, respectively.

A breakdown of the annual energy cost is provided in Table 5-6a.

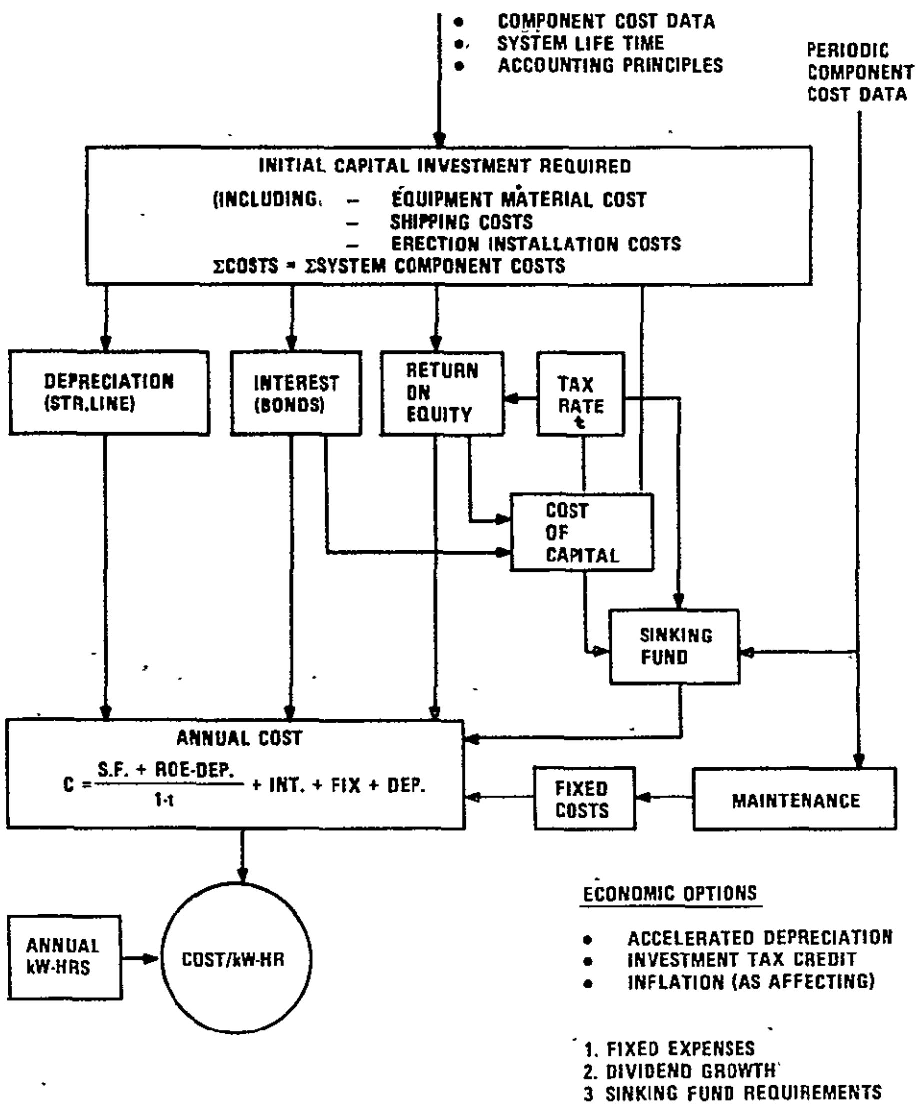


Figure 5-5. Economic Analytical Model

TABLE 5-6a. BREAKDOWN OF ENERGY GENERATION COSTS, ¢/KW-HR

System	Start Up Costs	Capital Fund	Return on Equity	Debt Interest	Federal Taxes	Maintenance	Total ¢/KW-HR
500 KW 12 MPH Median Wind Site	0.0921	0.1071	1.419	1.110	0.6492	0.6626	4.04
1500 KW 18 MPH Median Wind Site	0.0535	0.0414	0.5498	0.4303	0.2517	0.2432	1.57

5.1.6 SYSTEM FABRICATION AND ASSEMBLY

Various approaches were evaluated to determine the minimum cost fabrication and assembly procedure for the WTG. The preferred approach is to maximize the amount of assembly performed at the plant and to minimize the number of site operations.

The pacing items in the WTG fabrication cycle under production conditions are the rotor, transmission and generator. Normal lead times for these subsystems are expected to be 26 to 30 weeks.

The WTG fabrication process can be divided into 5 main areas as follows:

- Assembly of the nacelle at the plant
- Fabrication of the rotor at the plant
- Site preparation and erection of the tower
- Mounting of the nacelle and rotor with the tower
- Completion of system electrical connections and system check-out

Figure 5-6 illustrates the equipment contained within the nacelle. Based upon a preliminary assessment, it is estimated that 16 man-weeks would be required to perform the nacelle assembly for either design. Basically, this process would begin with attachment of the pintle and the main shaft subassemblies to the bed-plate, attachment of the transmission gearbox, insertion of the variable pitch input actuators, attachment of the generator, installation and connection of instrumentation and electrical leads and attachment of the nacelle housing. The nacelle could be transported to the site by means of a flatbed truck.

The rotor would be shipped directly from the factory to the site in a minimum of 3 subassemblies - two blades and the hub. An alternative, which may result in lower transportation as well as tooling costs, is to have a bolted joint in the blades. In either approach, complete assembly of the rotor subsystem would be performed on the ground at the site.

Prior to shipment to the site, the entire rotor subsystem would be checked out. However, the extent of the required inspection for a mass production effort has not yet been determined. It is anticipated that blade deflection, hub deflection and variable pitch change functional tests would be made on a sampling basis as a minimum.

Erection times averaging 6-8 weeks are reasonable expectations for truss towers; cost of the tower erection is included in the "tower" cost in Table 5-6. Since the weight of these towers is only about 40 - 60 tons, the foundation requirements are not as extensive as for a concrete tower. In addition, a full cure of 28 days would not be required for the foundation before the steel sections could be erected. The steel beams used in this type of construction can be easily transported by truck.

The larger weights involved in the concrete tower could complicate the program logistics depending upon the site location. Two alternative approaches are possible: the precast sections can be trucked directly to the site from a central location to the raw materials supplier/tower subcontractor, or, the raw materials can be trucked directly to the site where the precast sections could be made.

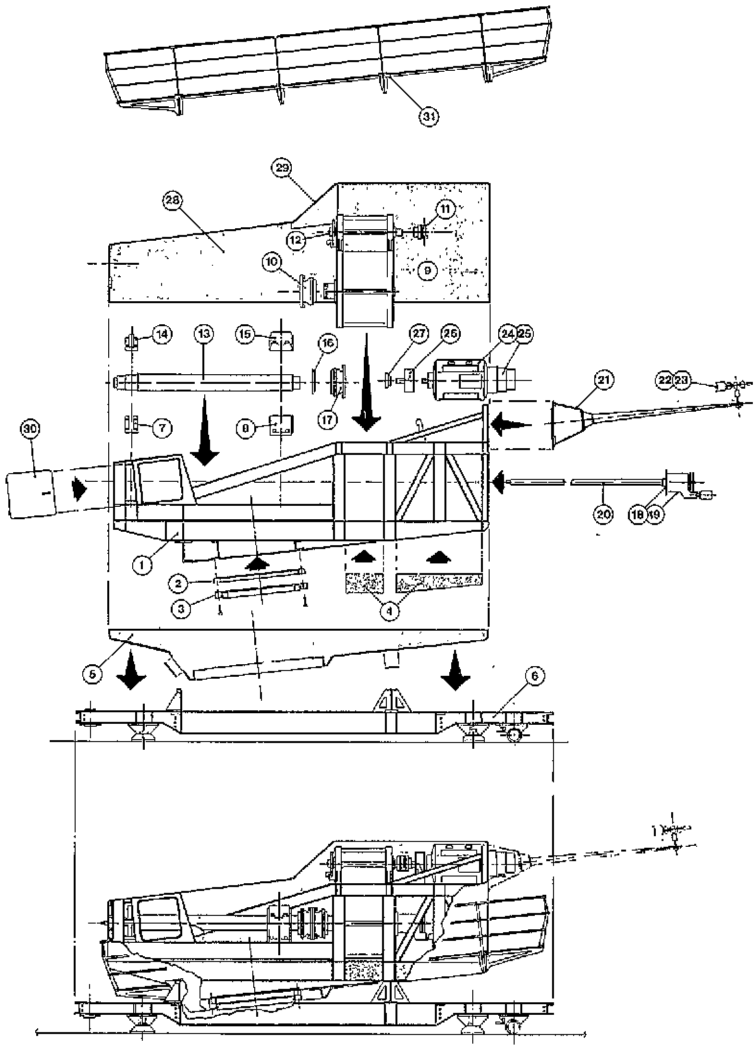
The weight of the concrete towers is estimated to be 225 to 325 tons for the 500kW and 1500kW systems, respectively. Therefore, larger foundations will be required for these structures. The heavier weight of the concrete tower may also require that a full cure period of 28 days be observed for the foundation before the tower is erected. Consequently, total erection time for the concrete tower is expected to be 8 to 12 weeks.

The equipment can be hoisted atop the tower in 1 or 2 steps. Preferably, the rotor system will be attached to the nacelle on the ground and the entire unit hoisted atop the tower. Alternatively, the nacelle would be put in place first and the rotor hoisted to be mated with the nacelle. The deciding factor as to which approach is used will be the weight of the nacelle and rotor as well as the available crane weight and height capacity. It is assumed that mobile cranes will be used where system weight and site terrain permits; alternatively, a portable erection tower can be assembled where necessary. Two hundred (200) ton mobile cranes having vertical lift capabilities of 190 feet are generally available. Total weights to be lifted are as follows:

	500 KW	1500 KW
NACELLE	22,730 kgs (25 tons)	42,730 kgs (47 tons)
ROTOR	12,720 kgs (14 tons)	16,360 kgs (18 tons)
TOTAL	35,450 kgs (39 tons)	59,090 kgs (65 tons)

Following installation of the WTG major subsystems, final electrical connections will be made within the nacelle. In addition, a control building will be erected at the tower base which will house the microcomputer, various electrical equipment and electrical connections to the utility.

The final step in the assembly process will be to check out the system. This procedure will consist of establishing the continuity of all control and electrical circuitry, allowing the system to break away under low wind/no load conditions and then operation under load. The time required for the check-out of production units will be dependent upon the experience gained at the time.



1. MAIN STRUCTURAL SUPPORT FRAME
2. MOUNTING PLATE FOR AZIMUTH BEARING
3. AZIMUTH BEARING ASSEMBLY
4. BALLAST BLOCK
5. LOWER HALF FIBERGLASS FAIRING
6. ASSEMBLY AND HANDLING JIG
7. ROTOR HUB BEARING (LOWER ASSEMBLY)
8. MAIN SHAFT THRUST BEARING (LOWER ASSEMBLY)
9. TRANSMISSION ASSEMBLY
10. FLEXIBLE COUPLING (TRANSMISSION SIDE) (LOW SPEED)
11. FLEXIBLE COUPLING (TRANSMISSION SIDE) (HIGH SPEED)
12. BRAKE ASSEMBLY
13. MAIN SHAFT (ROTOR/TRANSMISSION)
14. ROTOR HUB BEARING (UPPER ASSEMBLY)
15. MAIN SHAFT THRUST BEARING (UPPER ASSEMBLY)
16. THRUST COLLAR
17. FLEXIBLE COUPLING (MAIN SHAFT SIDE) (LOW SPEED)
18. PITCH CONTROL MECHANISM
19. DRIVE MOTOR FOR PITCH CONTROL MECHANISM ("0" RPM)
20. PITCH CONTROL DRIVE SHAFT
21. FORWARD FAIRING AND INSTRUMENTATION BOOM
22. WIND DIRECTION INDICATOR
23. WIND SPEED INDICATOR
24. GENERATOR
25. EXCITER
26. TORQUE - RPM SENSING DEVICE
27. FLEXIBLE COUPLING (GENERATOR SIDE) (HIGH SPEED)
28. UPPER FAIRING ASSEMBLY
29. FAIRING ACCESS PANELS
30. MAINTENANCE CREW ACCESS HATCH
31. SERVICING PLATFORM

Figure 5-6. Nacelle Assembly Sequence and Item Description

5.2 SYSTEM OPERATION - STABILITY AND CONTROL

To develop the performance specification for the control system, a detailed description was needed of the system's normal operational sequence, off-line stability characteristics, on-line stability characteristics, system dynamic behavior and potential system failure modes and effects.

The information presented in the following sections was developed not only to determine the control system logic and design specification, but to identify system hardware design constraints as well. The information presented, while preliminary in nature, does identify and outline work areas to be developed in detail during any subsequent hardware program.

5.2.1 SYSTEM OPERATIONAL DESCRIPTION

The information presented in this section details the systems operational sequences from startup to shutdown including system response to loss of network load, emergency conditions, and restarts. Covered in depth are startup, synchronization and cut-in, operation between cut-in and rated power, rated power to cut-out and high speed cut-out.

5.2.1.1 Start Up

The following is a description of the system startup procedure from a condition of no wind and zero rpm to synchronous rpm after obtaining cut-in wind speed, V_{CI}, for a specified period of time.

Wind speed and direction are obtained from the wind speed and direction sensors located upwind on the nacelle meteorological boom. The sensors provide control input signals to the master control system.

Initial Conditions (Points 1 to 2 on Figure 5-7)

- Blade pitch angle $\beta 3/4_r = 90°$ (blade held in horizontal position by breakaway friction)
- Rotor Speed $N_R \doteq 0$ rpm
- Wind Velocity $V_W \doteq 0$ mph
- System is on auxiliary power derived from the network and control system is monitoring all input signals V_W, N_r, P, ϕ, $\beta 3/4$, T_G, E_G, I_G and other system status data

As the wind speed increases from $V_w = 0$ mph, its magnitude and direction will be continuously monitored by the control system. No active control will be initiated on any component until the wind sensor indicates that the wind speed is equal to or greater than V_{CI} for approximately 5 minutes. At this time the control system will sense, via the azimuth position indicator, any misalignment between the rotor axis and the prevailing wind direction and command the azimuth drive motor to align the rotor axis with the wind. Simultaneously, the Bendix type pitch change drive motor will be engaged to cause a change in the blade

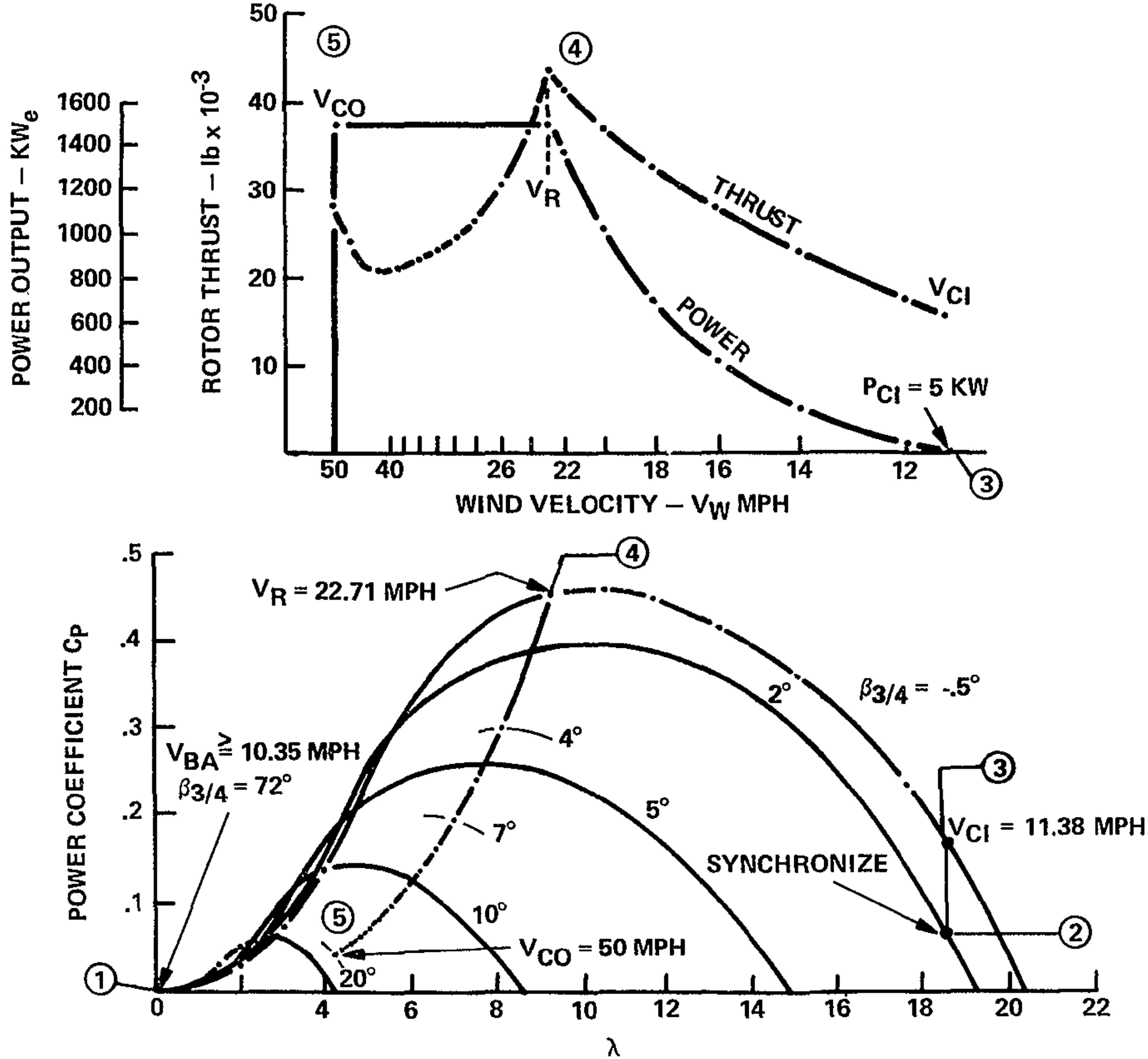


Figure 5-7. Effect of Wind Velocity and Pitch Angle β 3/4 on Cp, Power, and Thrust for 1500 kWe Preliminary Design

pitch from $\beta_{3/4R} = 90^o$ to the maximum rotor breakaway torque position of $\beta_{3/4R} = 72^o$. The blade pitch angle is determined from the output of a linear voltage differential transformer (LVDT) attached to the pitch change actuation shaft located in the pitch change power supply. Having the blade pitch to provide maximum torque and available cut-in velocity, the rotor will breakaway autonomously as the rotor axis is aligned with the wind. When the shaft speed sensor indicates a main rotor shaft speed of 0.5 rpm, the control system will command the pitch change mechanism to change $\beta_{3/4R}$ from 72^o to 2^o and the rotor will then accelerate to near synchronous rpm in approximately 7 minutes provided the wind is maintained. At 10% under synchronous speed the control system will take over and by modulating the blade pitch, monitoring the utility voltage, phase and frequency, and by utilizing the potential transformers, synchronize the generator to the utility network. Power in the rotor at this time will be sufficient to provide for generator friction and windage, core losses, per unit field losses and power train running torque.

Gusting conditions prior to cut-in will be entirely controlled by the control system software governor with feedback to the pitch change mechanism as explained in Section 5.3.4.

For wind direction variations of $\pm 15^o$ as determined by the wind direction sensor no action will be taken by the control system to affect realignment. For wind direction variation sensed at greater than $\pm 15^o$ for periods in excess of 6 min. the control system will command the azimuth drive to realign the rotor shaft into the wind by comparing the azimuth position data with prevailing wind direction data.

5.2.1.2 Synchronization and Cut-In

Initial Conditions (Point 3 on Figure 5-7)

- Blade Pitch; $\beta_{3/4R} = 2^o$
- Rotor speed $N_R = N_{rated}$ (synchronous)
- Wind speed $V_w = V_{CI}$
- Rotor power $P \leq 5$kwe at $\beta_{3/4R} = 2^o$

  $P \geqq 5$kwe at $\beta_{3/4R} = -0.5^o$
- Wind Azimuth $\phi = \pm 15^o$

Having satisfied the requirements for line placement, the control system will simultaneously command the main circuit breaker to close, thus connecting the WTG to the network. At this time the blade pitch will be changed from $\beta_{3/4R} = 2^o$ to $\beta_{3/4R} = -0.50^o$; the electrical power output will be approximately 5 kWe as determined by the current transformer and potential transformer data.

For the synchronous generator, the control of torque transients due to wind gusts subsequent to cut-in will be controlled by increase in the generator field strength in response to increases in torque as indicated by the torque sensor on the generator input shaft. For the induction generator, transients due to wind gusts subsequent to cut-in will be controlled by changing the rotor pitch angle. The control system will incorporate the feedback loop which maintains on-line stability, for either machine.

Variations in wind direction will be handled in the same manner as during line placement.

5.2.1.3 Cut-In Velocity to Rated Velocity

Initial Conditions (Points 3 to 4 on Figure 5-7)

- Blade Pitch; $\beta_{3/4R} = -0.5^{\circ}$
- Rotor Speed; $N_R = N_{rated}$
- Wind Velocity; $V_{CI} < V_w < V_R$

For wind velocities greater than cut-in and less than or equal to rated velocity, the regulation of power is controlled solely by changes in generator field strength in response to changes in the torque sensed by the generator shaft torque sensor. The control system will therefore issue no blade pitch change commands for $V_{CI} < V_W < V_R$ for the purpose of regulating power. Primary power control is based on generator input shaft torque instead of wind speed.

For wind direction changes, $\phi = \pm 15^{\circ}$, for wind velocities $< V_R$, there will be no change in rotor axis azimuth unless this azimuth variation is sensed for a period greater than 6 minutes. If a wind direction change occurs greater than $\pm 15^{\circ}$ for a period less than 6 minutes a slip clutch on the azimuth drive will allow the rotor axis to slip in the direction of the rapid change. The slip clutch is located on the output shaft of the azimuth drive speed reducer. The value of the slip torque would be preset and controlled during slip.

To control rapid gusts, blade pitch change and field driving* will be used in response to the torque sensed on the generator input shaft. The control system will initiate command signals to either or both pitch change and field strength depending upon the torque level, and the rate of torque increase.

5.2.1.4 Rated Velocity to Cut-Out Velocity

Initial Conditions (Points 4 to 5 on Figure 5-7)

- Blade Pitch; $\beta_{3/4R} = -0.5^{\circ}$
- Rotor speed; $N_r = N_{rated}$
- Rotor Power; $P = P_r$

* Field driving is used only with the synchronous machines, for the induction machine pitch control only is used to control gusting.

- Wind Velocity; $V_{co} > V_w > V_R$
- Wind Azimuth; $\phi = 0\text{-}15^o$

For wind velocities above rated velocity the control system will maintain constant rated power output based on current data from the current transformers located on the output of the synchronous generator and on sensed torque. As power increases above rated, the control system will sense the current and torque increases and increase pitch to maintain rated conditions. Should the power decrease, the control system will automatically try to reset blade pitch to -0.5^o, thereby increasing the power output. The equilibrium blade pitch angle will always be that angle which results in rated power output for V_w between rated and cut-out.

Gust control will be effected by proper application of either generator field driving or blade pitch change in response to variations in sensed torque. Again, the control systems primary response is to changes in generator input torque not wind speed or power output.

The tolerance on allowable wind azimuth variation for wind velocities greater than rated velocity will be proportional to the wind velocity up to cut-out. The actual angle ϕ will be determined for the relationship between ϕ and the moment due to inflow by determining the inflow angle variation with wind speed for a constant value of that moment. A slip clutch on the output shaft of the azimuth drive will provide for wind azimuth changes that exceed the preset angle for less than the dwell time.

5.2.1.5 High Speed Cut-Out/Shutdown

Initial Conditions (Point 5 on Figure 5-7)

- Blade Pitch; set at $\beta_{3/4}$ to give P_R
- Wind Velocity; V_w = 50 mph and increasing (1500KW)

  40 mph and increasing (500KW)
- Power = P_r
- Wind Azimuth; $\phi = \pm < 5^o$

The primary conditions for cut-out is achieved when the wind velocity as measured by the wind sensor reaches a value where small changes in the wind speed produce significant changes in wind turbine power. The precise control of the blade pitch angle to maintain absolutely constant power is difficult to achieve and therefore the system will begin to experience larger and larger current and voltage fluctuations for velocities of the wind above V_{co}. Therefore, when measured wind velocities are greater than V_{co} for periods greater than 6 minutes, the control system will increase blade pitch to bring the generator output current to 10% of the rated value in approximately 10 sec. When this value is achieved, the tie-in circuit breaker will be opened and blade pitch increased to prevent excessive overspeed.

If the measured wind speed falls below V_{co} for a period of 6 minutes, a high speed restart will be initiated.

Variation in power output due to gusts and wind inflow considerations will be controlled as previously described.

5.2.1.6 Low Speed Cut-Out and Shutdown

Initial Conditions (Point 2 on Figure 5-7)

- Wind Velocity; $V_w < V_{CI}$ (6 min.)
- Wind Azimuth; $\phi = < \pm 15^o$ (6 min.)
- Power; P = -10% for 6 min.
- Blade Pitch; $\beta\, 3/4 = -.5^o$

When the measured wind velocity falls below V_{CI} the generator will motor since the wind turbine will not have sufficient power output to supply the parasitic electrical and mechanical losses. This motoring mode will be sensed by the current and potential transformers and after a period of 6 minutes of motoring, the microcomputer will command the circuit breaker to open and thereby disconnect the WTG from the network. The control system will then set the blade pitch angle to 2^o in anticipation of starting the resynchronizing process. If, however, the wind does not return to V_{CI} after 6 minutes, the blade pitch will be set to 90^o or full feather, minimum drag configuration. The wind turbine will lose rpm slowly and the primary brake (located on the high speed through shaft on the speed increaser) will be proportionally applied at 0.5 rpm to stop the turbine blades in the horizontal position.

5.2.1.7 Restart at High Wind Speed

Initial Conditions (Point 5 on Figure 5-7)

- Wind Velocity; $V_w \leq V_{co}$
- Low magnitude gusts
- Small wind direction variations

When conditions of wind speed, gust value and $\Delta\,\phi$ are satisfied for 6 minutes, the restart sequence will be initiated. If the turbine speed is $N_R = 0.0$ the blade and azimuth will be changed until breakaway occurs. The pitch will be varied to control rpm for line placement as previously stated. Normal sequence will be used; supply power for generator friction and windage, running torque and core losses and +5kW on the line when the circuit breaker is closed. After closure, pitch change from 5kW to P_R will be accomplished in t = 10 sec. When cut-out occurs at V_{co}, the main breaker will have been opened, and the turbine will be allowed to overspeed* (not >10%) in a controlled manner.

* For the induction machine, the capacitor bank switch will be opened with the main breaker to prevent high stator voltages due to self excitation during overspeed.

For restart, β 3/4 is increased only to reduce rpm to the synchronous value and supply parasitic losses. When line placement occurs, β 3/4 will be reduced and power increased from a nominal 5kW to P_R in 10 secs.

5.2.1.8 Emergency Shutdown

There are operations or circumstances that will necessitate shutting down the Wind Turbine Generator in as short a time as possible. The most significant ones are listed below.

1. Imbalance or excessive vibration
2. Shaft overspeed
3. Loss of network (load)
4. Network line short circuit/Capacitor bank short circuit
5. Loss of critical control element, such as:
   a. speed and/or torque sensors
   b. hydraulic pressure
      1. pitch change mechanism
      2. azimuth slip clutch
   c. potential transformers
   d. data handling components
   e. actuators
      1. servo-valves, etc.

The following is a description of the emergency shutdown procedure resulting from the above conditions.

1. When an imbalance signal is received, the primary brake will be applied, blade pitch changed to full feather and the breaker opened.

2. An overspeed circuit signal will cause application of the primary brake, pitch change to full feather, the breaker is already open due to the loss of network or short circuit condition.

3. Under loss of network signal and then reclosure the main contactor will be opened* and the off-line to on-line procedure will be initiated. Under a lockout condition, shutdown will be effected using the primary brake and going to full feather; the breaker will already be open. In both instances, emergency power will be on until reclosure or shutdown is complete.

* For the induction machine, the capacitor bank contactor will be opened.

4. Same as item 3 but with no return to on-line.

5. On loss of signal from a critical control element, the circuit breaker will be opened, the primary brake applied, the blades feathered and the WTG status will be communicated to the dispatcher.

5.2.2 OFF-LINE STABILITY AND CONTROL

There are a number of considerations to bring the system to the "on-line" condition with minimum perturbation of the network. If the system is at zero speed, the torque provided by the rotor must be sufficient to overcome the breakaway friction of the shaft bearings, transmission and generator. The system must be brought to rated speed, maintained at that speed for a period of time while generator-to-line placing is established with minimal disturbance. This procedure is required for either the low or high wind cut-in velocities. Effects of gusts at these two conditions are examined below.

In the case of load loss to the generator, system protection against overspeed is required. Control requirements were established for this condition.

Sections 5.2.2.1 through 5.2.2.5 review the generator-to-line placing approach - it is applicable to both synchronous and induction generators.

5.2.2.1 Stability Models; Analysis and Approach

A general technique for synchronization is established utilizing blade pitch control to set WTG speed. A software governor, incorporated in the microcomputer, decreases shaft speed error and provides a blade angle position command. Speed trimming is utilized as required for generator-to-line phasing.

The approach for overspeed control with load loss is generally the same as described above for line placing. Pitch angle balance control by means of a software governor is utilized. After rated speed has been reestablished following the load loss, line placing may take place providing other system constraints and conditions are satisfactory. Line placing techniques are discussed in Section 5.2.3.

The analytical approach followed was to formulate a nonlinear model using the aerodynamic blade characteristics; and the breakaway and running frictions of the various components. A suitable governor was included into the non-linear model to study low and high wind velocity conditions, performance under gusting conditions and overspeed protection capability with load loss. Perturbation techniques were then utilized to develop a linear model of the control loop.

Figure 5-8 is the block diagram of the nonlinear system.

Load torque, Q_L, is made up of the torques representing the system losses and the torque equivalent of the power generated when appropriate. Q_1 is the breakaway torque and power generated when appropriate. Q_2 is the breakaway torque and exists only at zero speed ($\omega = 0$). Q is the running torque losses and is in general a function of the system speed. Since data was available only at rated speed, it was assumed constant over the speed range except at zero speed.

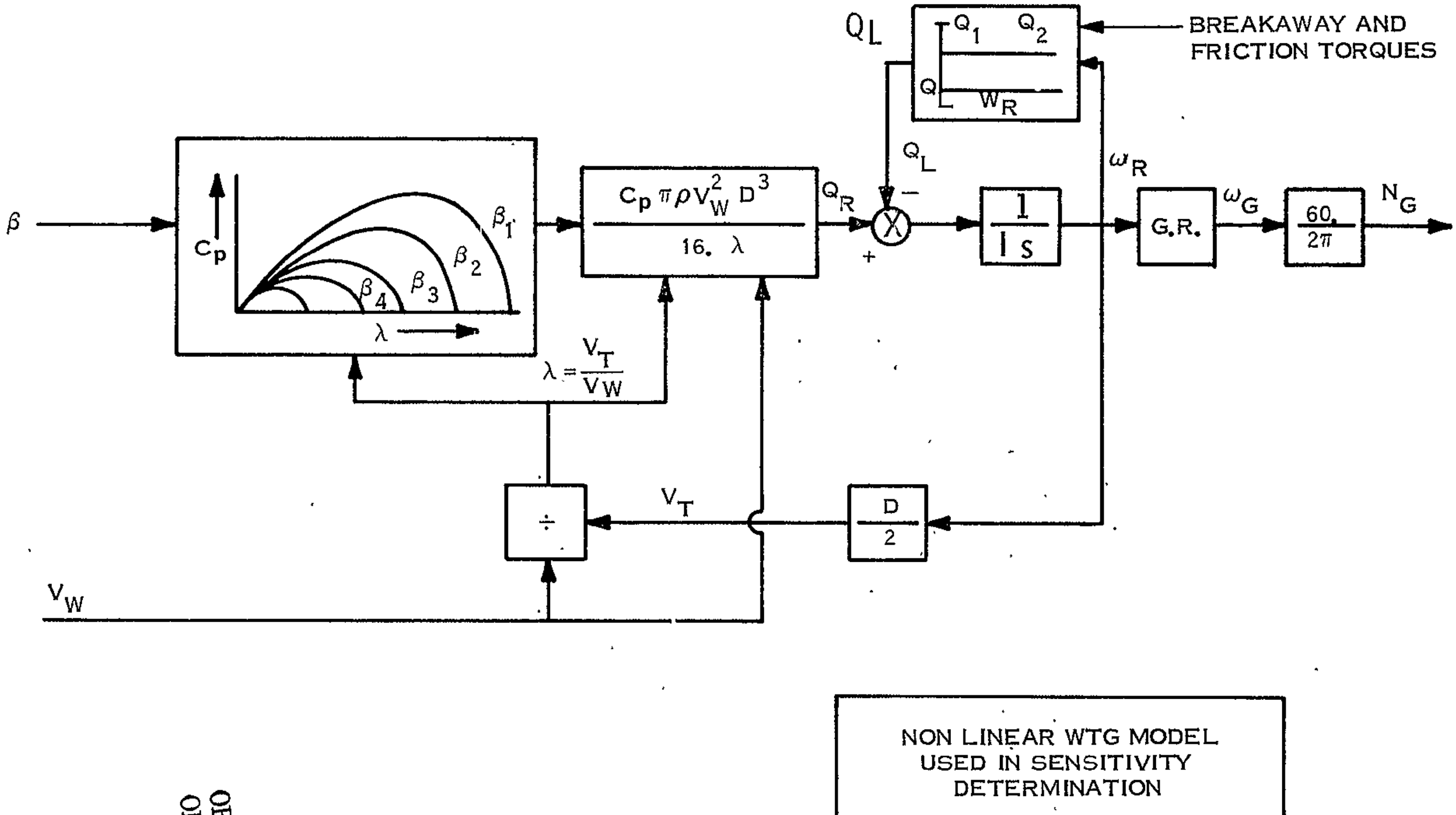


Figure 5-8. WTG Off Line Stability Model

Q_e is the torque required to produce the appropriate load on the machine for a specific wind velocity. For the off-line runs this term is zero. The unbalanced torque is integrated resulting in rotor speed, ω_r.

The rotor speed is used to calculate tip velocity, V_t, as well as the generator speeds ω_g (rad/sec) and N_g (RPM) through the transmission gear ratio, G.R.

The linear stability and response analysis requires the partial derivatives or perturbation coefficients to be derived from the model. Several operating points were selected which included the minimum and maximum wind, and several intermittent wind conditions. At each of these conditions, each independent variable was perturbed and from these data the partial derivatives or stability coefficients developed. The first set of data developed was the unbalanced torque derivatives which are plotted for various wind velocities and shown in Figure 5-9. A block diagram of the linearized system using these coefficients is shown in Figure 5-10.

Modification of Figure 5-10 results in the block diagram shown in Figure 5-11 which is given in terms of generator speed. Figure 5-12 is a plot of the speed partial derivatives with respect to wind velocity. Either of the two linear models may be used for linear stability and response analysis. Further, the torque partials are used to investigate system stability and performance for both on-line and off-line operation and for wind condition changes, such as gusts.

The data in Figures 5-9 and 5-12 show variations of about 50:1 in the linear parameters over the normal range of operations. These must be accommodated by the control system for suitable off-line operation.

5.2.2.2 Software Governor

A typical governor type was selected for off-line speed and synchronization control of the system. A block diagram of the linearized system with the governor is shown in Figure 5-13. It is an isosynchronous governor with temporary droop (proportional plus integral control).

Bode diagrams for the system using unity proportional and integral gain constants (K_g = 1, T_g = 1) at low wind cut-in and high wind cut-in velocities indicated the response characteristic will vary about ten to one over the operable wind velocity range. At high wind velocity the system natural frequency is about 5 rad/sec and is heavily damped while at low wind velocity the system natural frequency is 0.5 rad/sec and underdamped. This results from the wide range of stability derivative variations in the system (50:1) since the governor gain and compensation was fixed for this analysis.

The governor characteristics determined by the linear analysis were inserted into the nonlinear model (Figure 5-8) and used for off-line performance predictions and rated speed studies.

5.2.2.3 Control of Rated Speed

Figure 5-14 shows the acceleration of the machine to rated speed for low and high wind conditions. The actual time plots begin at the point where governor

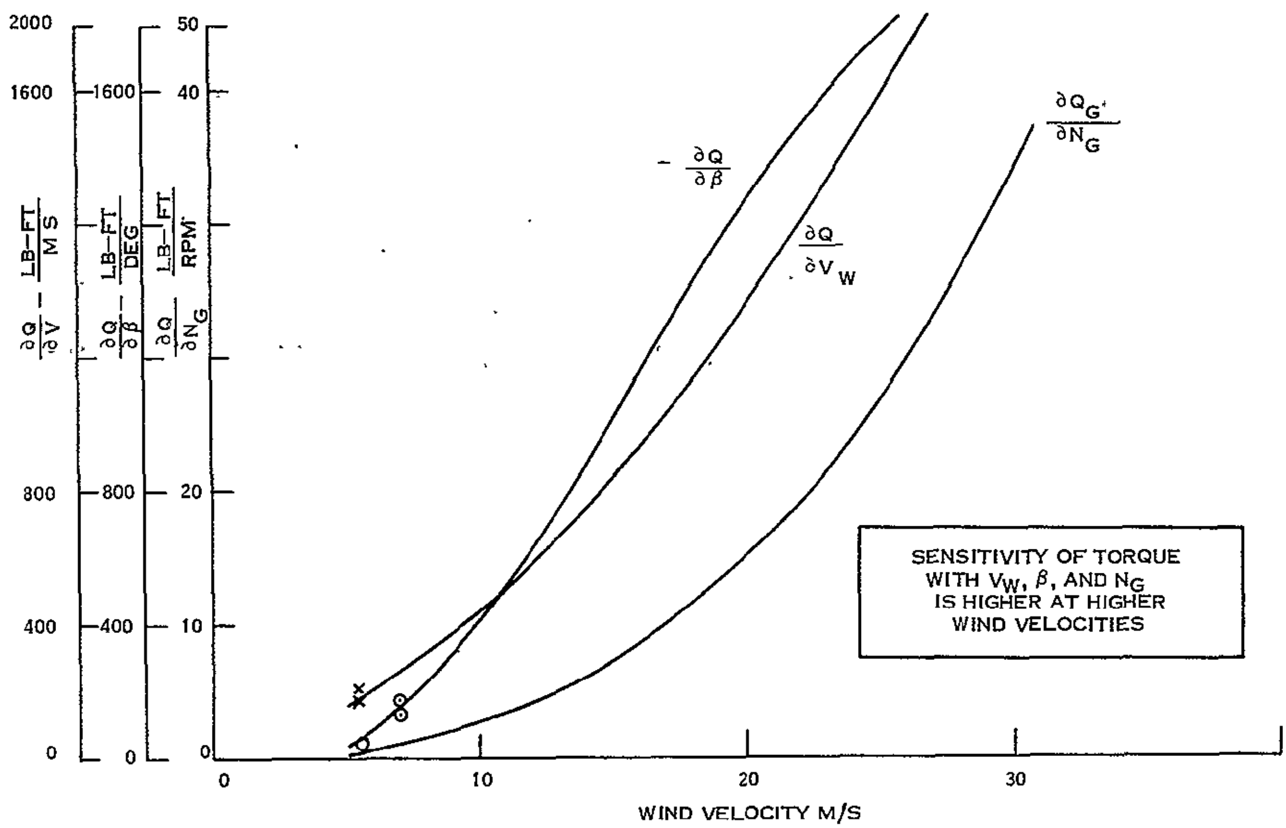


Figure 5-9. Torque Sensitivities

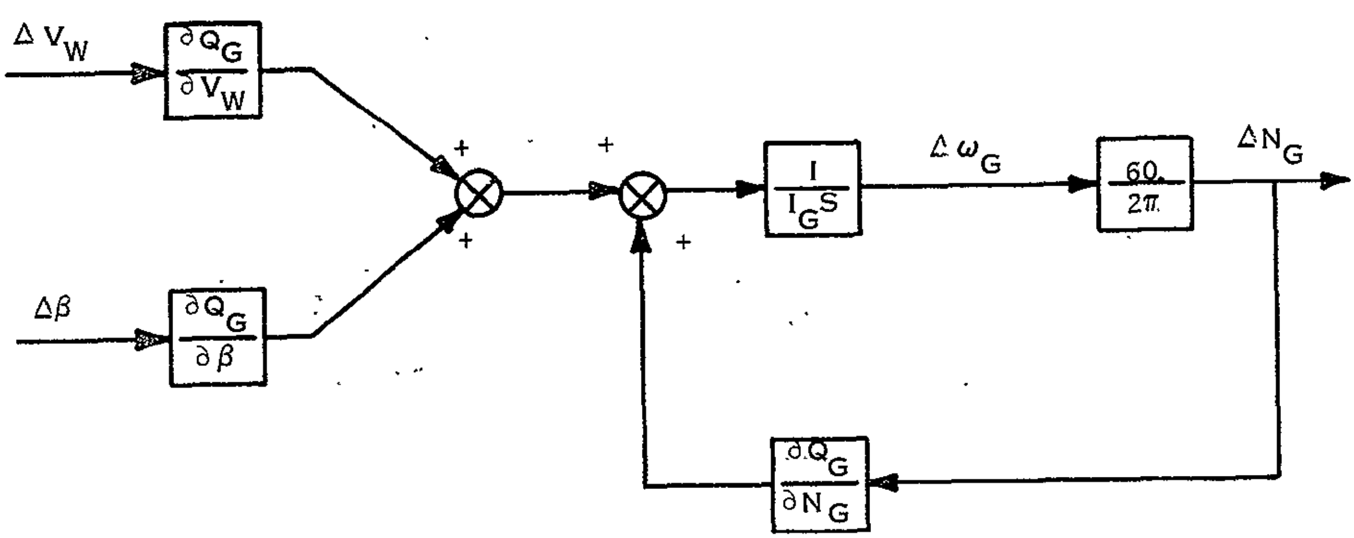


Figure 5-10. Linear Off Line Stability Model

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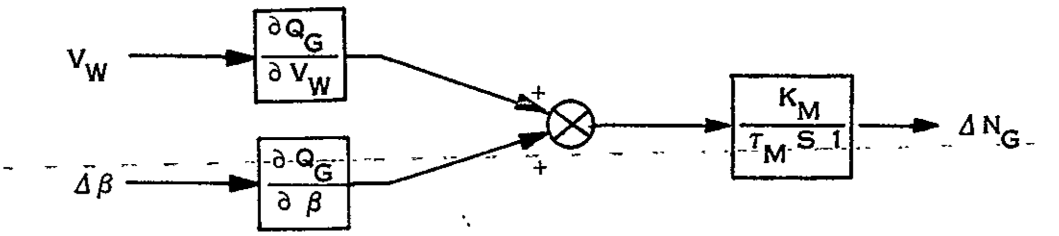

RESULTS

V_W M/S	β DEG	N_G RPM	K_M RAD SEC DEG	τ_M SEC
7.0	2.8	1800.0	0.131	61.0
30.0	20.0	1800.0	0.00242	1.13

60/1 RANGE IN
WTG DYNAMIC
CHARACTERISTICS AT TIME
OF SYNCHRONIZATION

Figure 5-11. Linear Off Line Stability Model Results

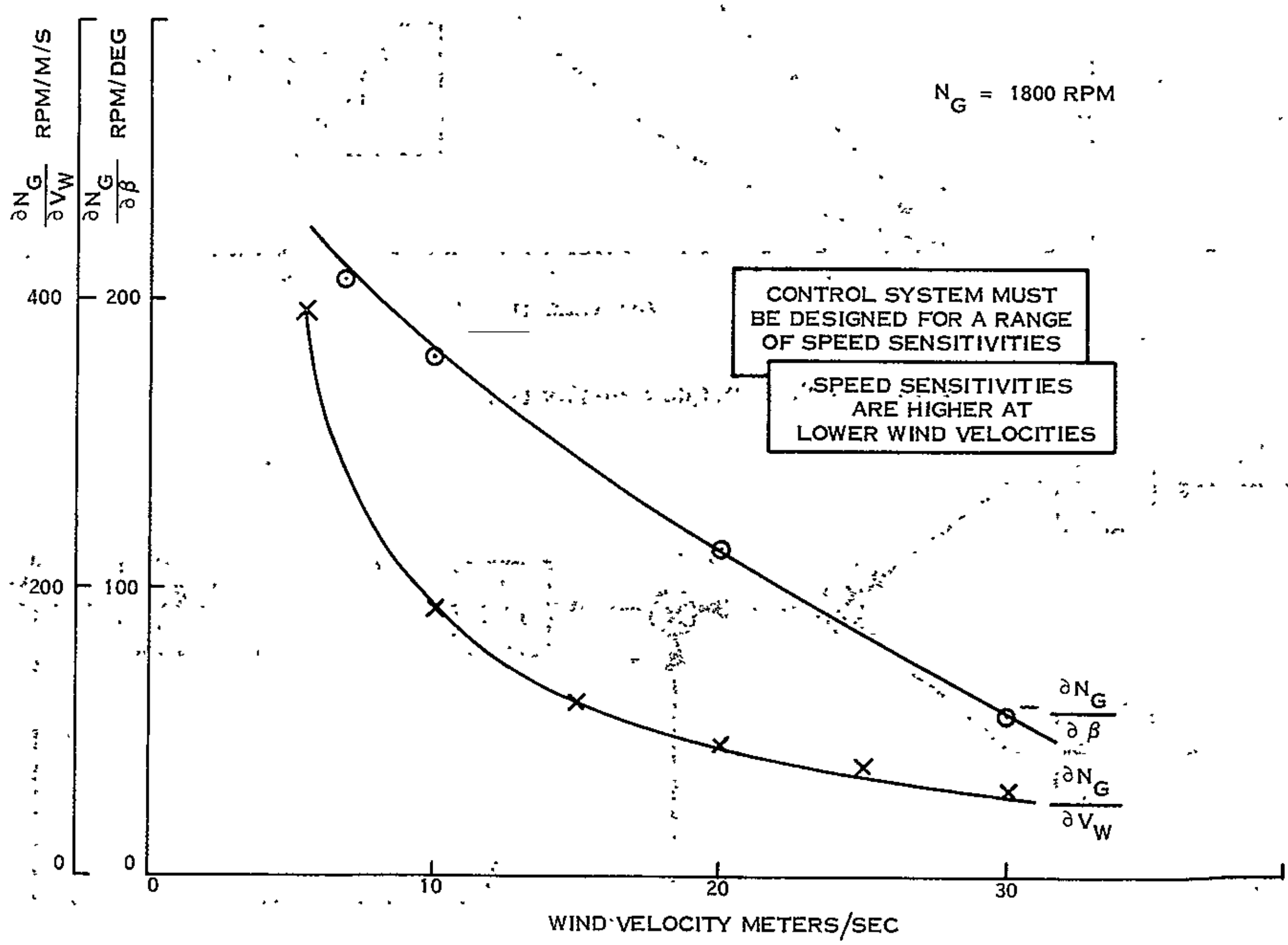

Figure 5-12. Speed Sensitivities

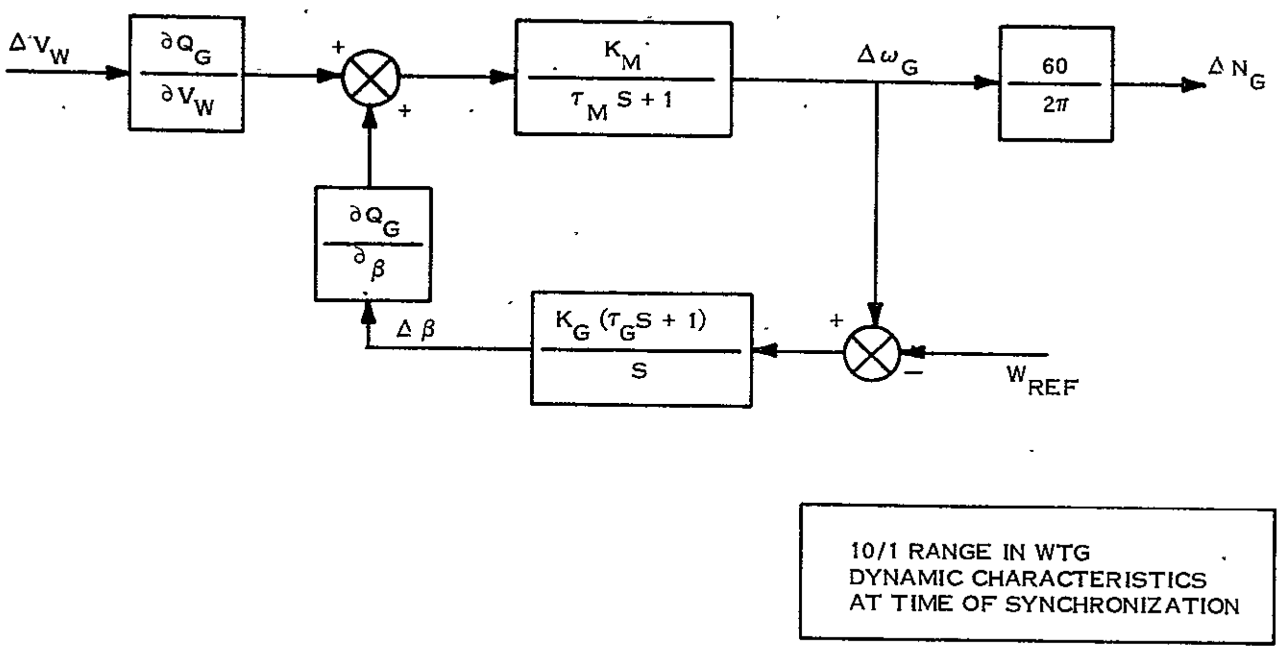

Figure 5-13. Linear Off Line Stability Model with Software Governor

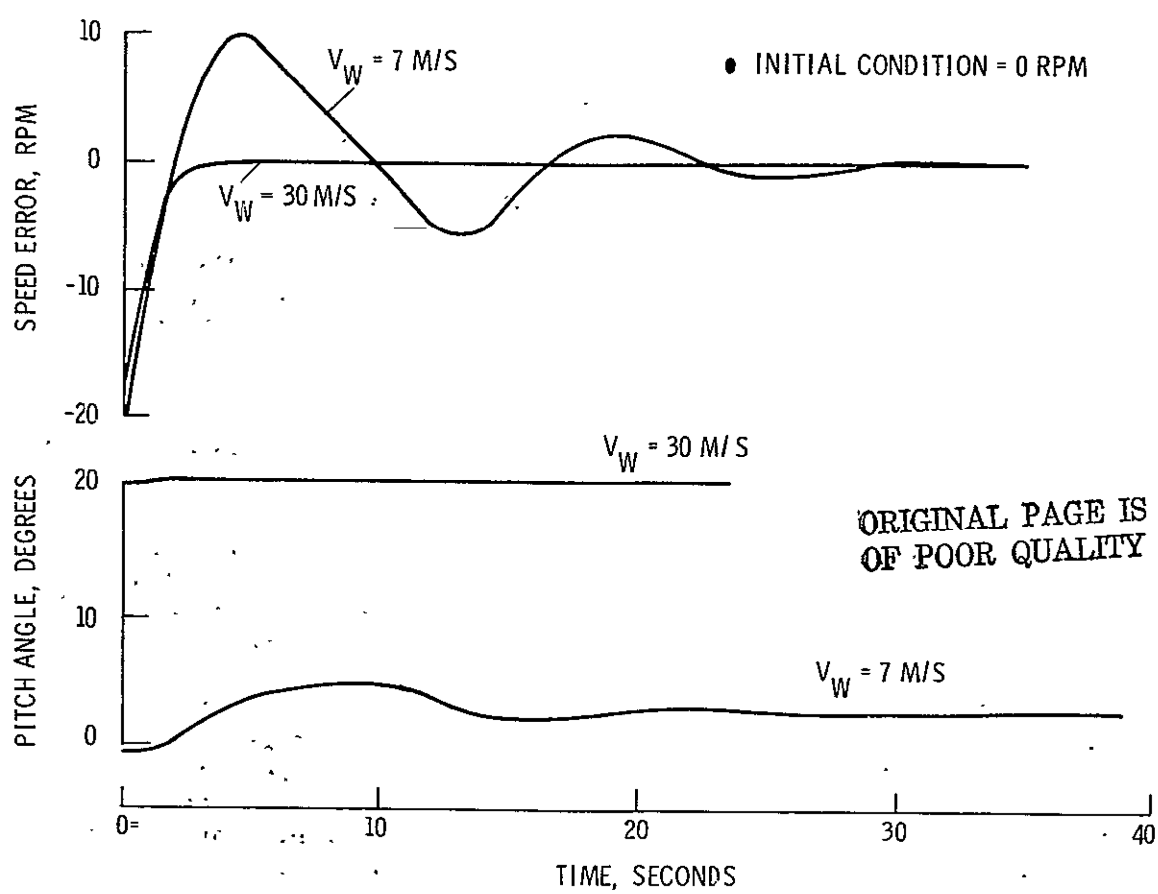

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Figure 5-14. Generator Speed Response Under Constant Wind Conditions control takes over which is 180 rpm (10% of rated speed) below rated speed, 1800 rpm. Below that speed the pitch blade angle is fixed and only released when the generator speed exceeds the threshold. The plot shows the generator speed and blade angle for each of the wind conditions. The high wind condition indicates an over-damped control system with no speed overshoot, while the low wind conditions shows the typical under-damped response with very small speed overshoot, 10 rpm at the generator shaft (about 1-2%). Line placement could be attempted for the over-damped system (high velocity cut-in) about five seconds after exceeding the control threshold. For the low wind velocity condition about 30 seconds is required. In each instance additional perturbations or disturbances to the system would increase the above times.

5.2.2.4 Gusting

The effect of gusting was examined to determine its effect on the off-line speed holding capability of the system and the ability to synchronize and cut-in to the line during gusting conditions.

The gust model used was that provided by NASA and is shown in Appendix 8.3. This model was included in the nonlinear digital model. The gust model was applied to the off-line controlled speed system using the previously described governor, for low and high wind velocities and short and long gust periods. These results are shown in Figures 5-15 and 5-16. Figure 5-15 shows the speed and blade responses for all second gust period at the low wind velocity state. The rotor speed was initially at rated speed when the gust was applied. It can be seen that the response is dictated by the system dynamics since the gust period is very short. The maximum speed error is about 3 generator rpm* (about 0.17%) and requires about 30 seconds to settle to a condition where line placement is possible.

Figure 5-16 shows the speed and blade angle response for a 100 second gust period for high and low wind velocities. In these instances, the system response is dictated by the gust period rather than system dynamics. Maximum speed error is 4 generator rpm (0.22%) but requires a time in excess of the gust period (100 seconds in this instance) before line placement can be attempted.

5.2.2.5 Overspeed Protection

One extremely important aspect of off-line control is overspeed control of the WTG with the loss of electrical load to the generator.

The governor configuration with its associated constants shown in Figure 5-8 was used in this analysis. Rated load was applied to the nonlinear model shown as a load torque. Without an overspeed governor control, the speed exceeded 200% of rated in less than 35 seconds with the loss of load. When the governor was included into the model, the maximum overspeed and time for recovery to rated speed were dependent on the slew rate of the pitch mechanism. With a 3 deg/sec slew rate, the peak overspeed was 1985 rpm or about 10% overspeed and occurred 3 seconds after the load loss. For the 7 deg/sec slew rate, the peak overspeed was 1884 rpm or 4.5% overspeed and occurred 2 seconds after load loss. Rated speed was reattained in 80 seconds. The results indicate that overspeed protection is required to protect the machinery from overspeed stresses and a pitch slew rate of

* This speed error is the same for both synchronous and induction machines.

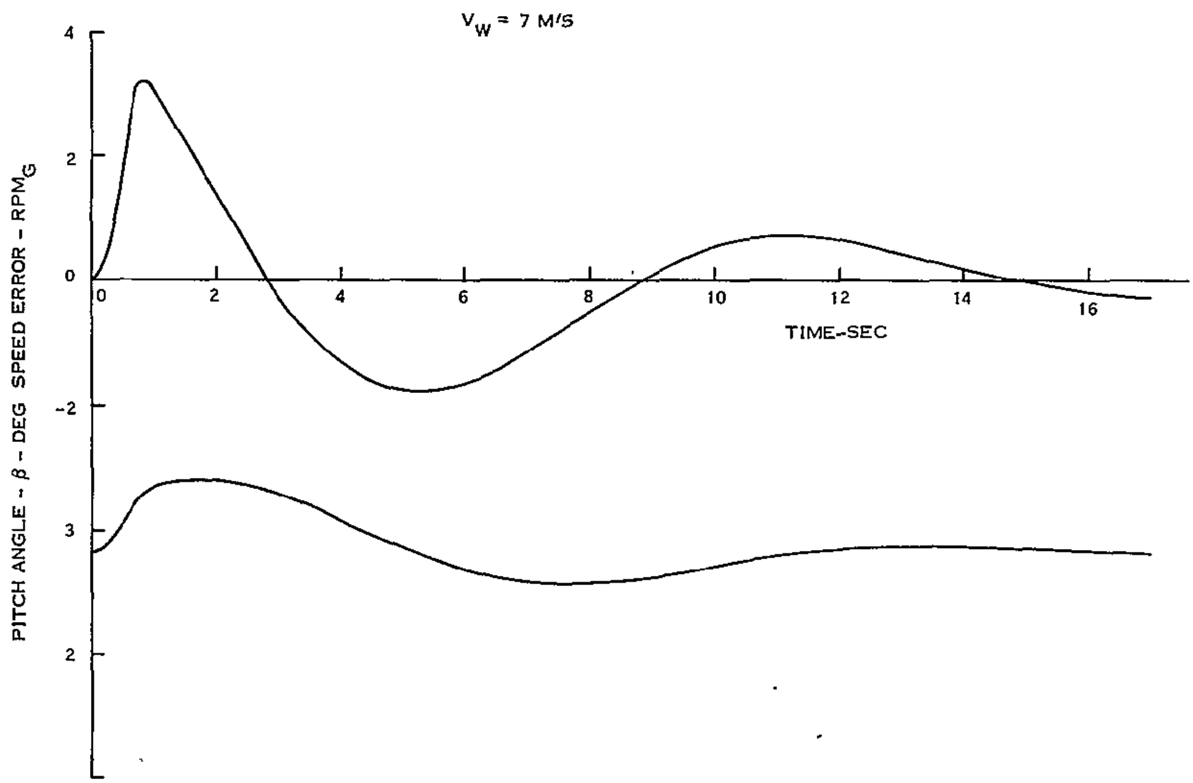

Figure 5-15. One Second Gust Response

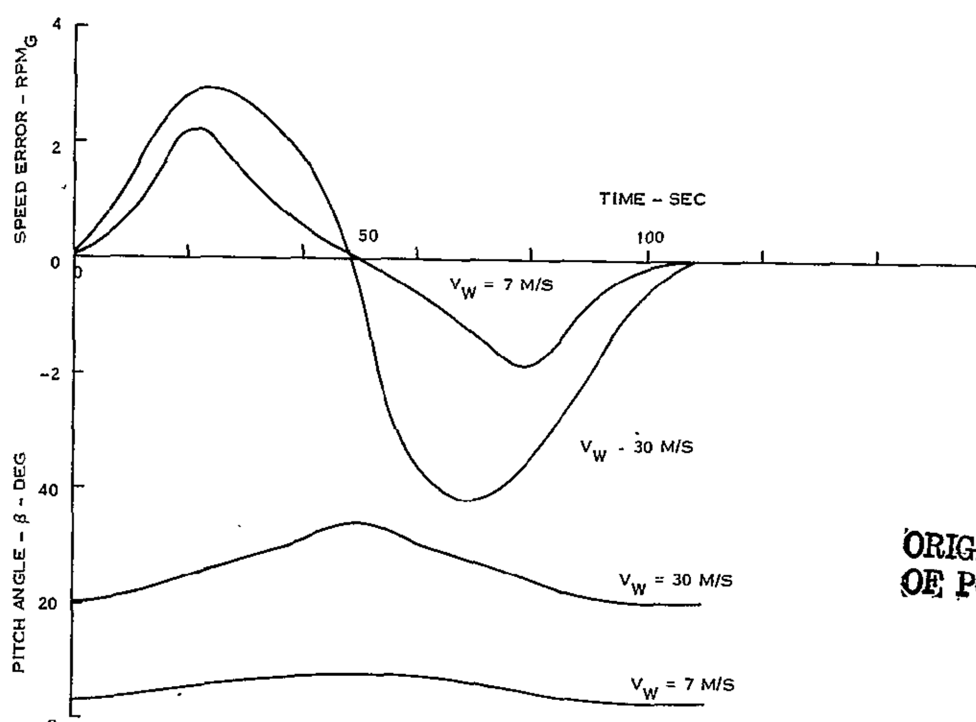

Figure 5-16. One Hundred Second Gust Response

3 deg/sec or greater is satisfactory. This approach is also implemented as software in the microcomputer and does not require additional hardware.

5.2.3 LINE PLACING

Line placing is the technique used to connect the alternating current machine, synchronous or induction, onto the network bus. The next 2 sections describe line placing for both synchronous and induction generators.

5.2.3.1 Synchronous Generator

The classic technique for synchronous machines is outlined below.

The generator regulator sets the excitation to produce the required line voltage. The alternator speed is brought to and held at synchronous speed. The speed reference is developed directly from the utility line so that the generator frequency and line frequency are equal. The phasing of the three lines is established using zero crossing detectors and by comparing the timing of the crossings of the generator and line. Minor trimming of the speed reference will produce adjustments in the phase relationship if required. For switching purposes (cut-in to the line), maintaining the phases within $\pm$ 18° for 1/4 second will permit closing the breakers. This is the equivalent of a 5% voltage fluctuation between each phase of the line and generator. This is also equivalent to a rate of change of frequency of less than 0.4 Hz/sec or 12 rpm/sec in the vicinity of synchronous speed (1800 rpm). Closing of the breaker normally will be attempted from the underspeed to overspeed condition (positive acceleration) to assure generator action rather than motoring at this time.

The synchronizer uses the network or line frequency as a reference with potential transformers measuring voltages across the lines and generator. The zero crossing detectors are associated with these transformers. The logic for synchronization is programmed in the microcomputer which in turn operates the breaker to connect the WTG to the line.

Synchronization and cut-in are under microcomputer control. Synchronization logic determines the generator frequency and its time rate of change. The phasing between the line and generator is also determined. A speed reference trim computation is determined based on these data and used for adjustment. When synchronization conditions are met, the breaker is closed connecting the line to the generator. The pitch angle is then set to be consistent with the measured wind velocity to produce power.

5.2.3.2 Induction Generator

Line placing of the 500 kW induction machine will proceed as follows:

1. A separate 4160V contactor connects and disconnects the 300 KVAR capacitor bank.

2. A high precision speed reference (in the microcomputer) is used to speed govern the wind rotor to 1803 RPM (approximately 15% of the generating output).

3. The capacitor bank is connected to the induction generator.

4. The output of the generator (capacitor-bank voltage) and network voltage are compared and synchronized to create the least surge on closing the main contactor. The -0.17% slip ensures generator action.

5. The rotor blades having been feathered to achieve speed regulation are now set at optimum pitch (-0.5° at the 3/4 chord position). The generator should then produce power with the minimum voltage disturbance on the distribution line.

Other approaches to achieve induction generator-to-line placing were examined; these are discussed below.

The simplest manner for placing the induction machine on-line is to close the main breaker when the induction generator is being governed at a speed between 1780 and 1820 RPM. If the speed is less than 1800 the machine will motor. At 1780 RPM, full load current will be drawn from the network. At speeds above 1800 RPM, the machine will generate power into the network. At 1820 RPM, while connected to the network, full load will be delivered to the utility. This technique requires a speed reference in the microcomputer which could be developed from the network frequency using software.

The method described above will probably result in a large current surge (motor or generator) with a resultant voltage dip on the distribution line of the network. Moreover, power factor correcting capacitors must also be connected and most probably a current onrush of 1500 amperes will occur with an additional voltage dip.

Alternatively, the capacitor bank can be permanently connected to the induction machine and placed on-line with the induction machine. In this case, the small amount of residual magnetism on the induction machine will be sufficient to begin charging the capacitors and regenerative action will occur. (a 188 KVAR bank is sufficient to obtain line voltage at no load). If the main contactor is now closed such that the network voltage adds to the instantaneous charge on the capacitor bank, and the speed of the induction machine is very nearly 1800 rpm, very little surge in current will occur. However, should the instantaneous voltage of capacitors and network be of opposite polarity, a large inrush current (3000 amps) of very high frequency (1K - 10KHz) will occur. Voltage dips or circuit breaker outages could result from this inrush. In addition, a motor or generator surge can occur depending on the slip speed (% deviation from synchronous) of the machine.

Furthermore, when disconnecting the machine and capacitor bank from the network, the self-excitation of the induction machine while being overdriven 10% by the wind rotor will generate a line-to-line voltage of approximately 6000 volts on the stator coil (a voltage rise of about 40 to 50%). Speed control of the wind rotor (4% overspeed in 1 to 2 secs.) could minimize this voltage rise, but over the years the repeated voltage surges could shorten the life of the stator insulation.

Permanently connecting the capacitors to the network side of the main contactor eliminates the capacitor from the line placing of the induction machine, but keeps a constant lending P.F. load of 300 KVAR on the distribution line which will cause a voltage rise on the line when the induction machine is disconnected from the network. Also, the capacitors are not protected by the main contactor of the WTG system. Should a capacitor short occur, fuses on the capacitor bank will open. Failure of the fuses could cause an upstream circuit breaker to open or bursting of the capacitor could result.

On disconnecting the induction machine both the main contactor and capacitor bank switch should be opened simultaneously. The capacitor bank will self-discharge in about 5 minutes. If reconnection is desired before the capacitors are fully discharged, reconnection to the induction generator will not cause momentary inrush greater than 400 amps, due to the impedance of the generator, and self excitation will not cause high voltage as long as the speed is governed to 1803 RPM on closing.

5.2.4 ON-LINE STABILITY

The purpose of the analysis was to determine the limits of generator on-line stability for torque angle transients due to wind gusting, short circuits, and network load redistribution, (as may be occasioned by the loss of a large generator supplying the utility network). Determination of the limits of stability were undertaken to establish requirements for preliminary design of the electrical system of the WTG. Particular elements which it was perceived might be affected are: the pitch control of the wind rotor, the generator voltage regulator, the generator exciter, the control electronics (micro computer), and other control elements associated with generator/network protection. It is emphasized that the on-line stability strongly depends on specific applications of the utility network (line impedance), the generator excitation design, and the pitch control capability of the rotor, and must be analyzed accordingly for any given utility network, WTG systems design, and wind site.

5.2.4.1 Stability Model: Analysis and Approach

The generator on-line stability model is shown in Figure 5-17. The inertia of the wind rotor (blades and hub), transmission gears, and shaft is that which would be viewed from the generator shaft to the wind rotor. The wind rotor is turning at a constant RPM and the generator shaft speed is fixed at 1800 RPM. The pitch control mechanism is capable of changing the pitch angle at 70/sec (under control of the microcomputer). The brushless type synchronous generator is a salient-pole machine whose stator output is connected to an infinite bus (E_e) by way of a power transformer, and does not include machine reactances. The voltage regulator is the transformer SCR variety which regulates and supplies the dc field on the stator of the exciter. The three-phase ac generator on the rotor of the exciter is rectified by the diodes imbedded in the generator fan and thus the exciter output drives the salient-pole fields of the synchronous generator without benefit of brushes or collector rings, and field control of the generator is affected via phase control of the SCR's in the voltage regulator.

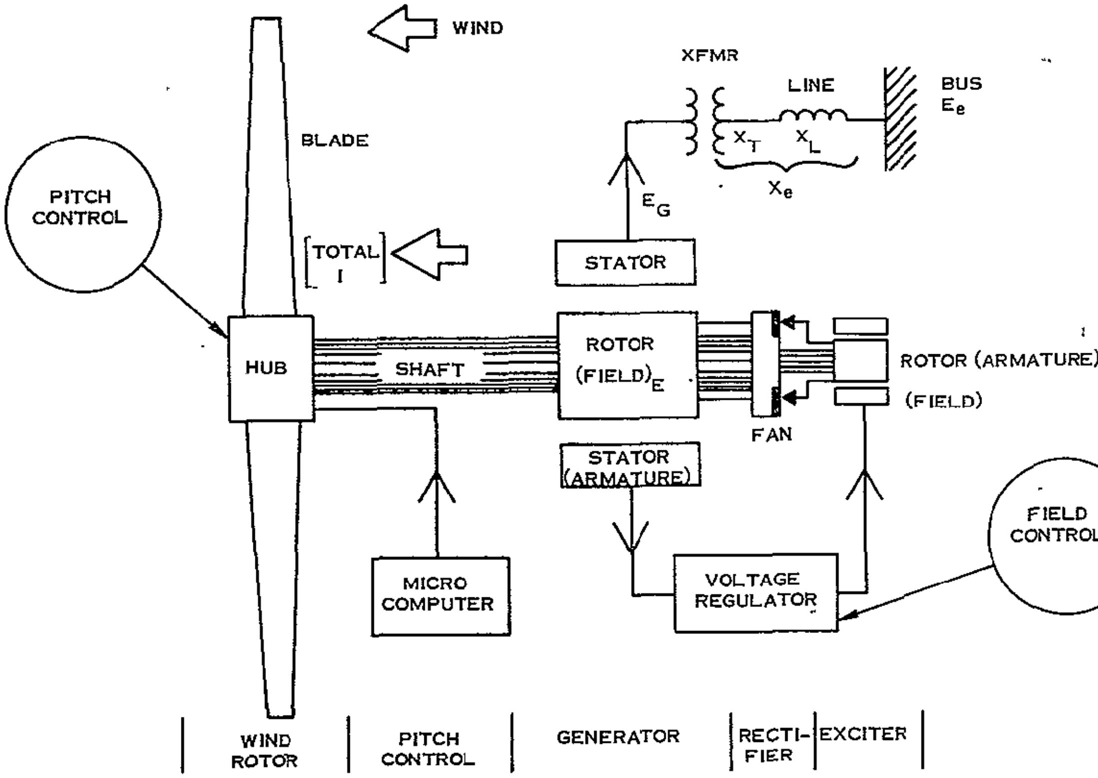

Figure 5-17. Generator on Line Stability Model

The general equation for such a system is given by:

$$I \frac{d^2\delta}{dt^2} + K_D \frac{d\delta}{dt} + k_s \delta = Q_{Applied} - Q_{Developed}$$

where:

I = Inertia of the total rotating mass.

K_D = Damping coefficient due to amortisseur windings (omitted from the analysis, since this analysis is not concerned with the decay of oscillations but more the amplitude of the first swing).

K_S = Resilience (spring) of the shaft, gears, hub, etc. (neglected for this analysis)

$Q_{Applied}$ = The torque, caused by gusting of the wind which impinges on the rotor blades. This will be discussed more in following paragraphs.

$Q_{Developed}$ = The reaction torque developed in the synchronous generator. This will also be discussed more in the following paragraphs.

The general approach to solving this model is to express the inertia, and torques in terms of per unit machine values. The wind gust and resulting torque pulse is modeled to obtain the applied torque as a function of time, and the generator and its excitation system are also modeled. It is well known that the developed torque of the generator is a function of the applied field*, and therefore as shown in the model, Figure 5-17, the voltage regulator and exciter are included.

Since the analysis is parametric rather than directed to a specific design, parametric ranges were selected for: wind rotor to generator inertias (total rotating mass); generator sizes (per unit constants); and line impedances** (generator to infinite bus and transformer). A step by step numerical analysis was then run on the indicated model.

5.2.4.2 Wind Gusts/Torque Pulses

A preliminary wind model for application in dynamic and control analyses of wind generators was supplied by NASA (See Appendix 8.3). The model provides a steady state wind profile and discrete gusts which are superimposed on this profile by vector addition. The gusts produced are large enough to cover the entire swept area of the wind rotor. The model is to be used for mean wind speeds of about 18 mph, but will give good results down to 12 mph. As the period of the gusts approaches 60 to 90 seconds, the magnitude of the gusts produced approach a limiting value, see Figure 5-18. It was noted that the limiting value was never a 100% increase over the median wind (as measured at 9.14m) speed, and therefore, rather than deal with many median wind conditions (9 mph to 21 mph) and tower heights (21.3-51.8m) it was decided to express the gust amplitude as a percent increase above the median wind speed (Figure 5-19). The results indicated that little error would be introduced by considering the 18 mph (median wind) gust to be typical of all gusts. Discrete gusts (See Figure 5-20) were then determined for the 18 mph median wind at a rotor axis height of 30.5 meters. Generally, if the median wind is greater for that tower height, the maximum amplitude would be about 5 percentage points higher, but the gust period would be the same. If the tower height were increased, and the median wind was the same, the maximum amplitude would be lower. The 21 mph curve of Figure 5-19 represents a worst case since it is the greatest median wind at the lowest tower height. The torque developed by the wind was then determined using the expression: $\tau = k\, C_p\, V^3$

where:

k = rationalizing constant plus air density

V = wind velocity

* For the induction machine, developed torque is a function of rotor slip, so that the above equation applies except that the angle is measured in mechanical degrees.

** For the induction machine the reactance of the capacitor bank would be included.

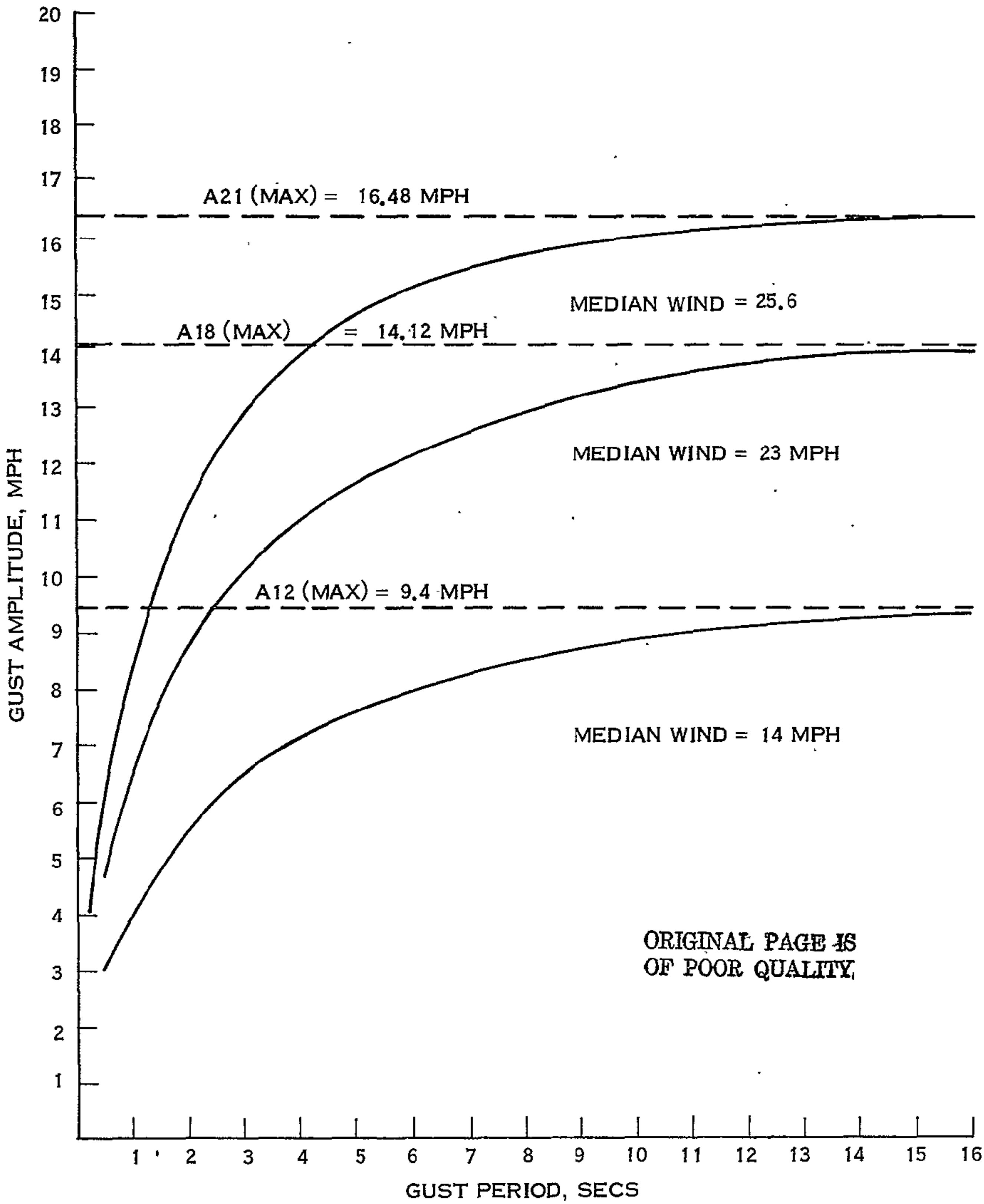

Figure 5-18. Gust Amplitude vs. Time

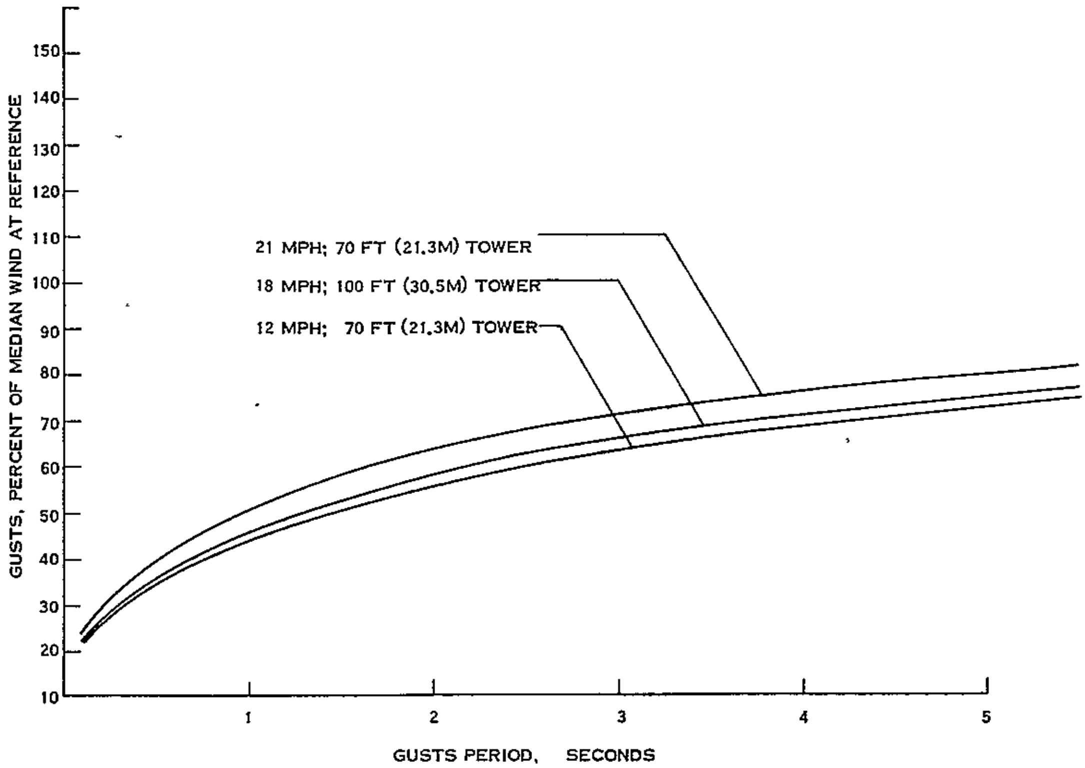

Figure 5-19. Gust Amplitude vs. Time

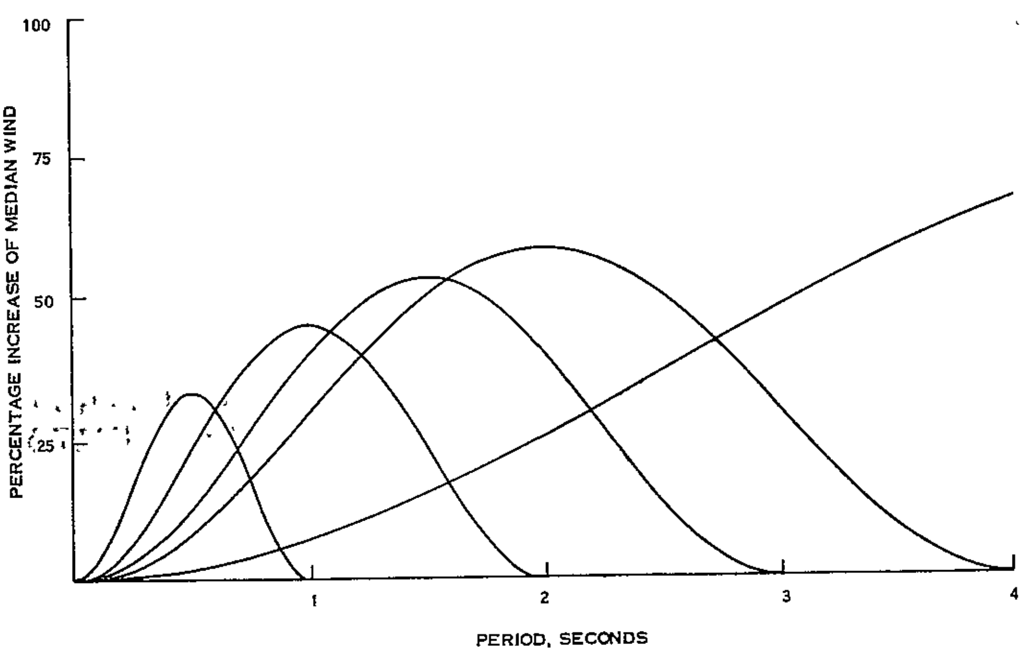

Figure 5-20. Normalized Discrete Wind Gusts

C_p = power coefficient

Typical results of the effect of gusts upon torque are shown in Figure 5-21.

Note from Figure 5-20 that the maximum gust amplitude and therefore maximum torque amplitude always occurs at the 1/2 period point of the gust period. The results of Figure 5-21 position the maximum torques with respect to time(rather than duration of gust or period of gust). Assuming that the generator is operating at rated wind speed it can be seen that gusts of 0.60 to 0.80 second duration will produce a 100% change of torque in 0.3 to 0.4 seconds; a very likely transient for causing out of step (unstable) generator operation. For this reason, it was decided to apply an assumed pitch control to reduce the wind torque whenever the generator developed torque exceeded 0.8 PU and the limiting rates of net torque reduction were thus determined.

5.2.4.3 Generators

Analyses showed that, when comparing machine constants (reactances, etc.) of synchronous generators at 0.8 PF, the characteristics were sufficiently close over the range from 100-3000 kW so that the stability performance of one would be generally applicable to any synchronous machine for the specified power requirements. On-line stability analyses of induction generators, conducted in separate studies, showed that the criteria for stability analysis is significantly different, being primarily pull-out torque with a consequent voltage dip.

5.2.4.4 Line Impedance (Network)

Generalized characteristics of the developed torque in a synchronous generator were analyzed as a function of torque or power angle with per unit field and per unit line impedances as parameters. Indications are that the developed torque in the generator increases directly as the per unit field, where per unit field is defined by the saturation curve. Conversely, if the system impedance is increased, the torque developed in the generator is lessened. The high impedance connecting line to the infinite bus (0.4 P.U.) is referred to as a "weak" system and the low impedance line (0.2 P.U.) is referred to as a "stiff" system. An example of each was studied, and the effects of network impedance was found to have a significant effect on stability.

5.2.4.5 System Inertia

In the per unit system, the inertia is expressed by H-factor which is:

$$H = \frac{\text{Kinetic Energy Stored in Rotating Mass}}{\text{Basic KVA}}$$

The H factor is calculated for specific rotor/hub design inertias and for specific power ratings and generally will not vary significantly ($\pm$ 20%) over the range in power levels for 100 to 3000 kW.

5.2.4.6 Stability Under Short Circuit

An examination of the generator reactances and the inertias of the WTG was made. The values indicate that the units should be generally acceptable to utilities from the standpoint of riding through system faults. Specific studies only appear necessary if the inertias are far less than those used in this analysis.

5.2.4.7 Stability - Loss of Large Network Generator

Measures to combat transient instability under a loss of network generator condition fall under one of the following categories:

a) Reduction of the disturbing influence by minimizing the fault severity and duration (Critical Clearing Time).

b) Increase in the natural restoring synchronizing forces through strengthening of the transmission system. (Short Lines).

c) Increase in restoring synchronizing forces through transient forcing of excitation and consequent boost of internal machine flux levels. (Field Driving).

d) Injection of braking energy through fast prime mover energy control (Pitch Control).

e) Injection of braking energy through temporary switching of resistors or other network parameters. (Dummy Load).

Measures b) and c) would be used for transients due to gusting, as previously stated. Measures d) and e) would be used for overspeeding of the wind rotor, as for example, upon the opening of the distribution line connecting the generator to the infinite bus. The WTG has no capability for increasing the torque speed characteristic in response to a demand transient so that, the only measure remaining is for the WTG to ride through the re-distribution of power angle (as though it were a short circuit) depending upon network system controls to minimize the fault duration.

5.2.4.8 Stability Under Wind Gusting

In general, for low frequency gusts of significant amplitude instability will result if no pitch control action is taken. To maintain stability, torque reduction by pitch control was used and the limiting rates of net torque reduction determined. The essential parametric results are given in Figure 5-22. This plots the minimum torque reduction rate (%/sec) required to maintain stability as a function of gust period. Two curves are shown, one for a relatively "stiff" system (X_e = 0.2 P.U.) and the other for a weaker system (X_e = 0.4 P.U.). X_e is the total transformer and equivalent system per unit reactance looking out from the transformer HV terminals. It does not include any machine reactances.

As indicated, for a weak system, the critical gusts are short ones (such as 1 sec. duration) and very high rates of pitch control are necessary to maintain stability. The effect of high performance excitation systems is also shown on Figure 5-22 and indicates that an SCR system of 10 p.u. ceiling produces a rather dramatic reduction in the pitch control requirements.

The second major result is that inertia is relatively an unimportant parameter (from the gust stability standpoint). A variation of $\pm$ 20% in total H constant

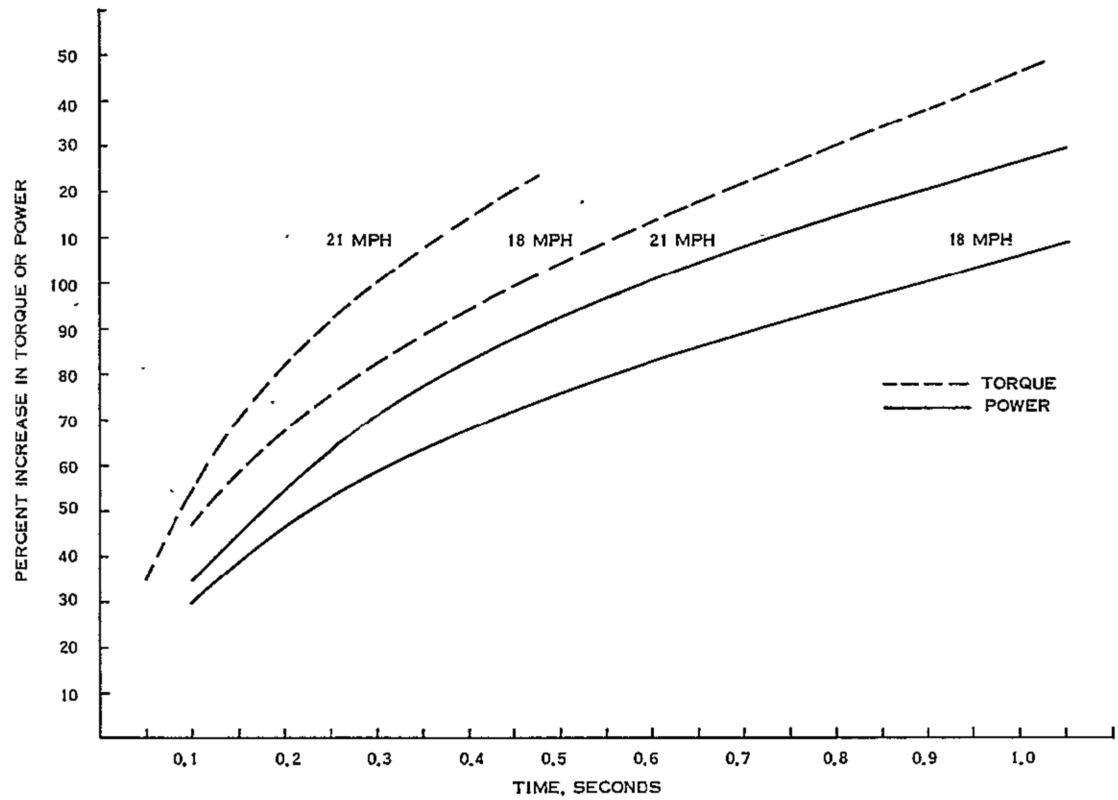

Figure 5-21. Envelopes of Maximum Torque and Power Pulses vs. Time

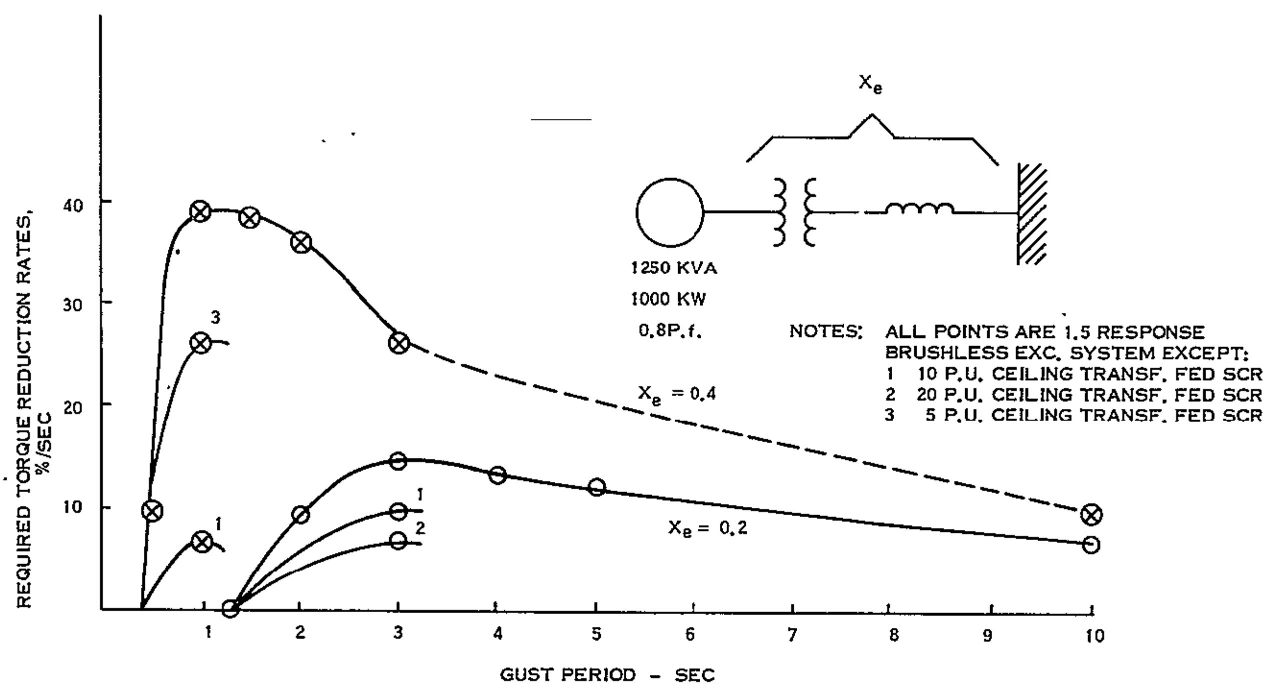

Figure 5-22. On Line Stability Torque Reduction Requirements

produced insignificant effects on the required torque reduction rates.

5.2.4.9 Conclusions

Transient swings of generator torque angles resulting from short circuits will not adversely affect the synchronism of the generator in the WTG as long as the fault is cleared within the nominal number of cycles for typical circuit breakers. Similarly, transient swings due to network load redistribution will not cause loss of synchronism.

Wind gusting of the order of one to three seconds can cause instability problems unless:

a) The WTG is designed to minimize the line impedance, X_e.*

b) High performance excitation systems capable of 5-20 pu field driving are utilized during transients due to gusting.

c) The pitch control is quick enough to produce the torque reduction rates indicated. This solution appears to have the most dramatic results.

It is further emphasized that on-line stability depends on utility network characteristics, wind site characteristics and system design; therefore, stability analyses should be performed for any specific application.

5.2.5 SYSTEM DYNAMIC ANALYSIS

The design of large WTG's is affected by dynamic interactions, resulting in load magnifications, which can have serious reliability and/or system cost implications. To perform a dynamic analysis on the WTG system requires that all of the load inputs to the system be defined in terms of their magnitude and frequency as discussed in Section 5.2.5.1. This, along with system configurations, component weights and operational design conditions, is required prior to any detailed sizing of structural members that satisfy the life and design requirements.

This section describes the design considerations of the rotor, tower and main shaft with respect to the dynamic load factor problem. Section 5.2.5.2 discusses the rotor blade structural dynamic analysis, Section 5.2.5.3 presents the rotor/tower design considerations arising from the system dynamics and Section 5.2.5.4 discusses the design of the main shaft from a dynamic response viewpoint.

5.2.5.1 System Loadings

Before proceeding with these sections, it is important to understand the source of the system loadings which play a critical role in the design. In addition, to the basic thrust imparted to the rotor blades, 4 additional system loadings are of interest; these arise from the wind shear, wind inflow, tower shadowing and gyroscopic effects. A discussion of these phenomena is provided below.

* For Reference: $X_e \approx 0.2$ p.u. for a 500 KVA generator would result from about 10 miles of 336 ACSR conductor in 32 inch delta spacing, feeding a 100 KVA substation transformer (0.035 p.u.) connected to a 34.5 KV 100 MVA bus (0.005 p.u.)

Vertical Wind Shear

The variations in wind velocity with height above the ground is known as vertical wind shear or simply wind shear. The exact variation with height is dependent upon the surface roughness characteristics of the local terrain over which the wind passes. The mathematical expression characterizing this variation is a power law relationship reported in Reference 5-2; as $\frac{V}{V_o} = \left(\frac{H}{H_o}\right)^n$ where the typical values for the exponent n are given as follows:

Flat open country,	.16
Rough wooded Country; city suburbs	.28
Heavily built up urban centers,	.40

For the purposes of this study flat open country was selected and therefore all analytical calculations were made with n = 1/6, (see Figure 5-23). This relationship is very important in the performance analysis of the wind turbine rotor system since all of the velocity duration curves are referenced to a height above ground of 9.14m and all of the power and energy calculations are performed at the rotor axis. The low reference height is deceiving in that more power and energy is available at higher altitudes. However, wind shear results in a rotor moment that varies from zero to a maximum, twice per revolution (see Figure 5-24). The moment is caused by differential lift across the turbine rotor diameter; the magnitude of this moment is directly proportional to the thrust or wind velocity. This wind shear moment also creates a torque reaction resulting from the gyroscopic effects of the turbine rotor.

Wind Inflow

Variations in wind direction also produce rotor moments which are a function of the wind inflow angle ϕ and wind velocity. The origin of these moments is best explained by referring to Figure 5-25 (a) and (b). Here a wind turbine rotor is shown in the 12 o'clock - 6 o'clock position (0^o - 180^o) while rotating in a counter-clockwise direction. Figure 5-25 (a) shows the velocity vector diagram resulting from a wind flow parallel to the rotor axis. The wind velocity shown is the free stream velocity with the required momentum theory velocity correction to account for stream tube expansion at the rotor plane. As the upper diagram shows, the effective velocity V_e on the top and bottom blade is the same and neglecting wind shear the resultant blade lift would be identical, producing zero net moment. The lower figure shows the vector summation for a wind inflow angle $\phi = 15^o$. As can be seen there is a differential velocity and angle of attack (α) across the rotor diameter which produces a net positive or negative moment value depending on the wind inflow direction.

This moment can, therefore, add or subtract from the moment induced by wind shear depending upon the direction of the wind change and the rotation of the blades. Figure 5-26 shows the magnitude of this inflow moment as a function of wind speed and inflow angle ϕ for the 1500 kWe unit.

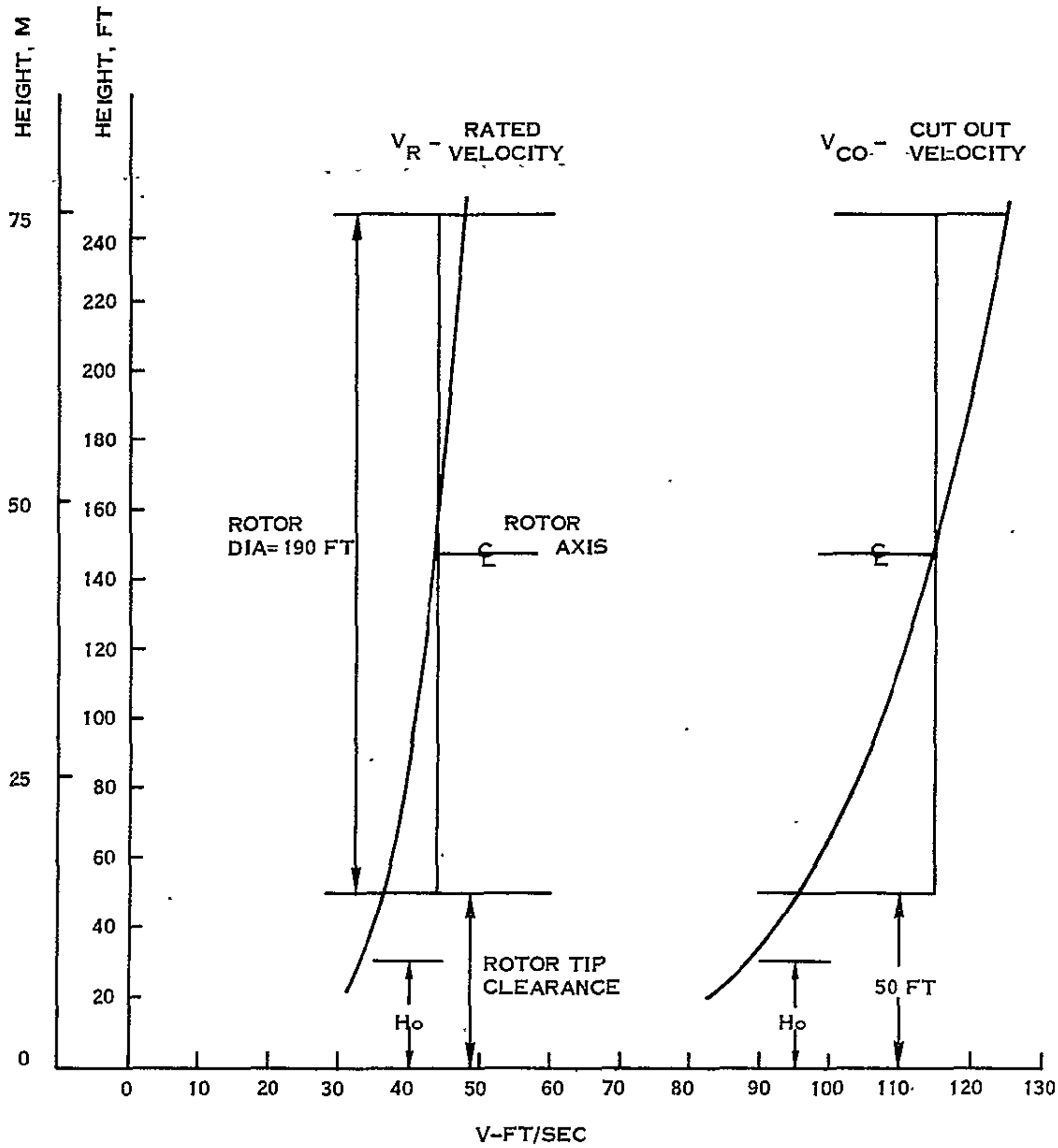

Figure 5-23. 1500 kW WTG Wind Shear Data

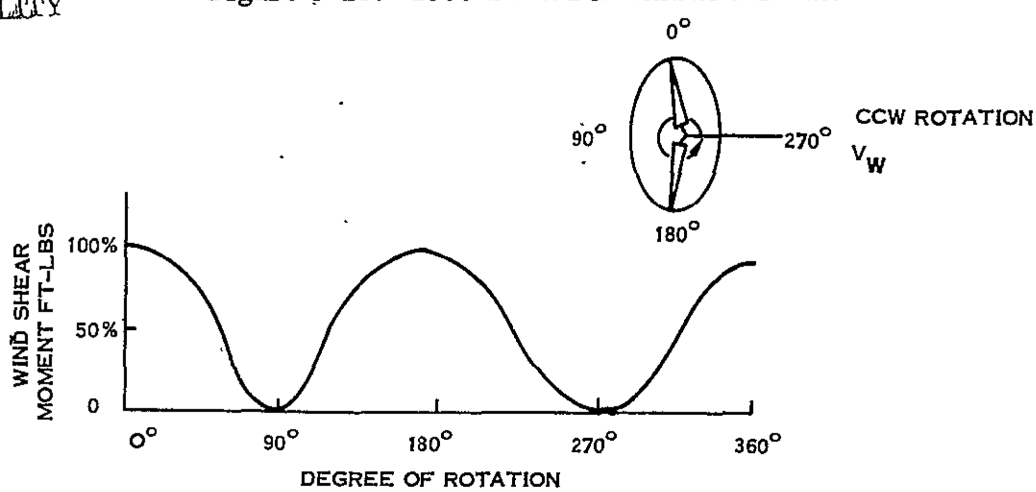

Figure 5-24. Wind Shear Moment vs. Rotor Position

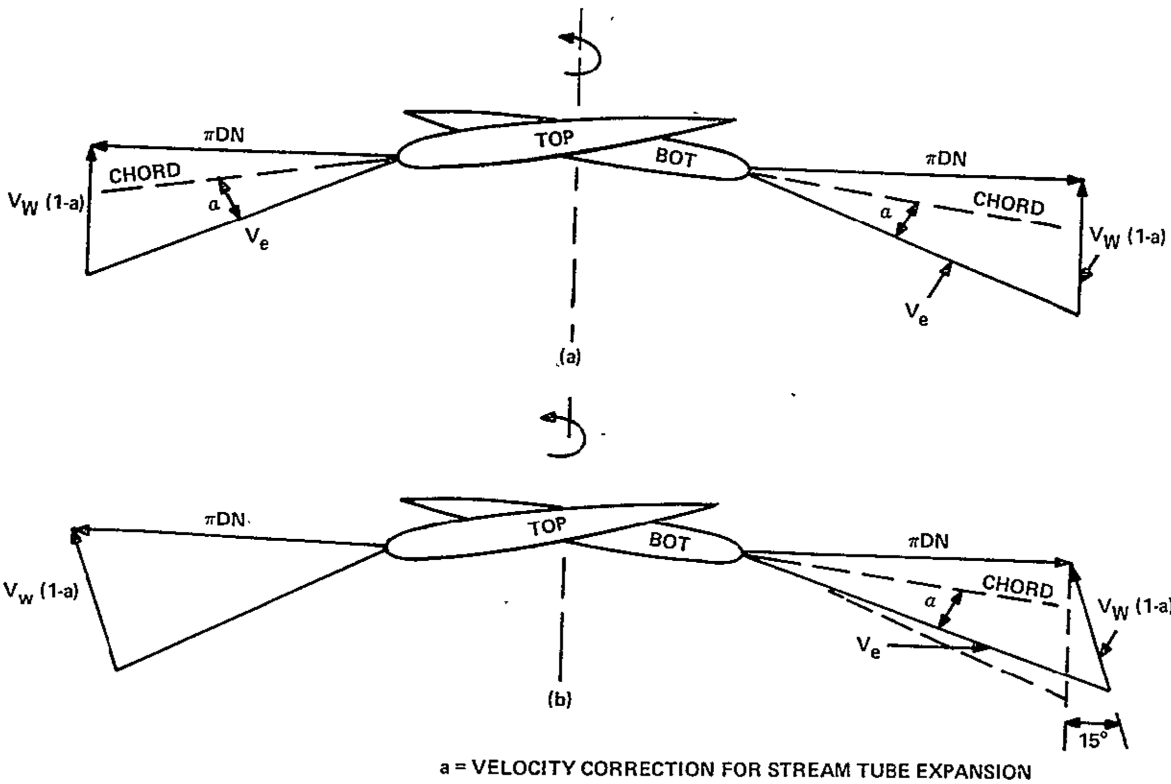

Figure 5-25. Effect of Wind Inflow Angle

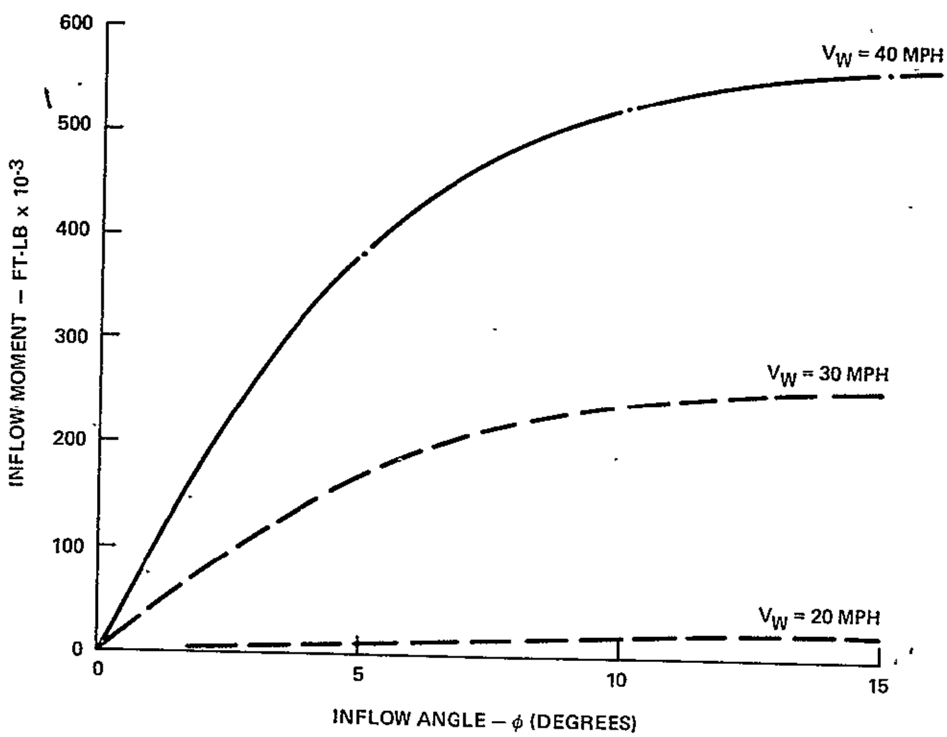

Figure 5-26. Effect of Wind Velocity and Inflow Angle on Rotor Bending Moment

Tower Shadowing

Placing the rotor downwind of the tower results in an obstruction of the wind flow into the wind turbine rotor blades as they pass behind the tower structure. This phenomena is known as tower shadowing. The velocity decrease caused by tower shadowing results in at least a 2/rev load pulse on the rotor. To determine the magnitude of this load variation various aerodynamic references were examined to determine if there were analytical solutions available that mathematically describe the velocity distribution behind a given structure. For symmetrical bodies, such as cylinders, approximate solutions to this problem were given by Reference 5-3 as:

$$V = 0.946V_x (C_D d/x)^{0.5} e^{(-Y^2/0.0888C_D dx)}$$

where,

V_w = wind speed - ft/sec .

C_D = cylinder drag

d = diameter of cylinder-ft.

x = distance behind cylinder-ft.

y = distance in direction of wake width-ft.

Thus by assuming that the truss tower is constructed of members of cylindrical cross section an approximate solution to the velocity distribution and magnitude may be obtained using the above solution. Two tower shadowing conditions were studied as shown in Figure 5-27. Velocities were obtained at blade tip clearance heights of 15.2m and at the top of the tower, 41.1m. Tables 5-7 and 5-8 present the results.

It is also important to note that the wake width shown is half the actual width. In addition, it is noted that cross member wake effects have not been included. For the truss tower designs presented, inclusion of cross member effects is expected to change the velocity profile at the rotor plane, but, not the magnitude of the maximum velocity change. It is recommended that the ΔV shown be used as the maximum uniform value of the decrease in velocity across the tower width and the wake dimension be used to bring the velocity up to V_∞ , or free stream value, for the parallel flow case.

Gyroscopic Loading

The rotating blades of the wind turbine rotor form an inertial disc. Since this rotor disc is able to rotate about an axis other than its spin axis gyroscopic reactions will be induced in both the rotor and the tower. Typical loadings that produce the gyroscopic reactions are rotor overhang, wind shear, tower shadowing and wind inflow. Wind shear loading (Section 5.2.5.1) produces a moment, shown in Figure 5-29. The value of this moment varies from 0 to a maximum and varies as the square of the velocity. This moment induces a gyroscopic torque about the aximuth axis that is in phase with the wind shear moment in both

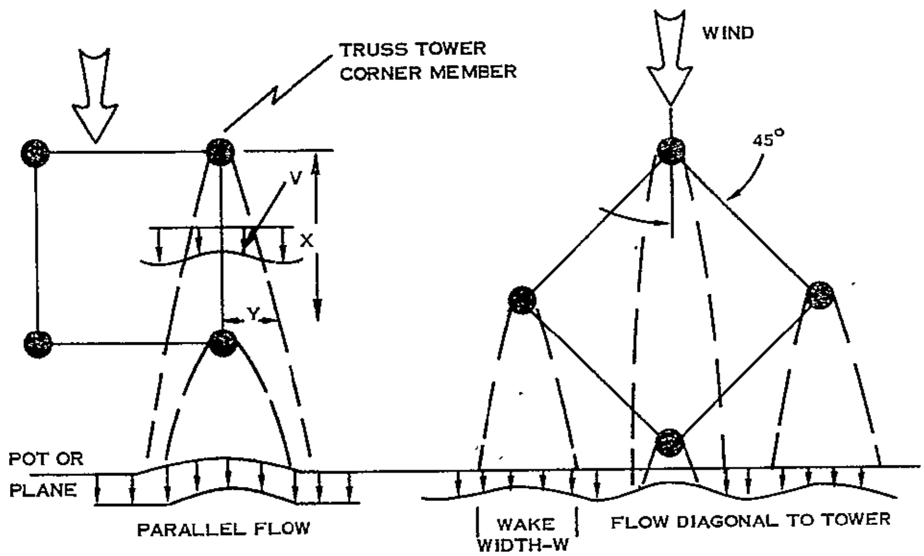

Figure 5-27. Truss Tower Shadowing Models

TABLE 5-7. TRUSS TOWER WAKE VELOCITY FOR PARALLEL FLOW

WIND SPEED, MPH	HEIGHT m(ft)	VELOCITY m/s (fps)	DISTANCE TO ROTOR, m(ft)	ΔV, m/s (fps)	WAKE WIDTH, m(ft) W/2=y
18	15.2 (50)	8.84 (29)	3.05 (10)	1.2 (4)	2.74 (9)
	41.1 (135)	10.4 (34)	3.05 (10)	1.2 (4)	2.74 (9)
40	15.2 (50)	19.5 (64)	3.05 (10)	2.1 (7)	3.05 (10)
	41.1 (135)	22.9 (75)	3.05 (10)	2.4 (8)	3.05 (10)
80	15.2 (50)	38.7 (127)	3.05 (10)	4.0 (13)	3.05 (10)
	41.1 (135)	46.0 (151)	3.05 (10)	4.6 (15)	3.05 (10)

TABLE 5-8. TRUSS TOWER WAKE VELOCITY FOR 45° FLOW

WIND SPEED	HEIGHT m (ft)	V, m/s (fps)	DISTANCE TO ROTOR, m (ft) L	CTR	RT	ΔV, m/s (fps) L	CTR	RT	WAKE WIDTH,m(ft) W/2=y L	CTR	RT
18	15.2 (50)	8.84 (29)	3.7 (12)	.91 (3)	3.7 (12)	.91 (3)	1.8 (6)	.91 (3)	3.4	1.8	3.4
	41.1 (135)	10.4 (34)	3.7 (12)	.91 (3)	3.7 (12)	.91 (3)	2.1 (7)	.91 (3)	3.4	1.8	3.4
40	15.2 (50)	19.5 (64)	3.7 (12)	.91 (3)	3.7 (12)	1.8 (6)	4.0 (13)	1.8 (6)	3.4	1.8	3.4
	41.1 (135)	22.9 (75)	3.7 (12)	.91 (3)	3.7 (12)	1.8 (6)	4.9 (16)	1.8 (6)	3.4	1.8	3.4
80	15.2 (50)	38.7 (127)	3.7 (12)	.91 (3)	3.7 (12)	3.4 (11)	8.2 (27)	3.4 (11)	3.7	1.8	3.7
	41.1 (135)	46.0 (151)	3.7 (12)	.91 (3)	3.7 (12)	4.0 (13)	9.4 (31)	4.0 (13)	3.7	1.8	3.7

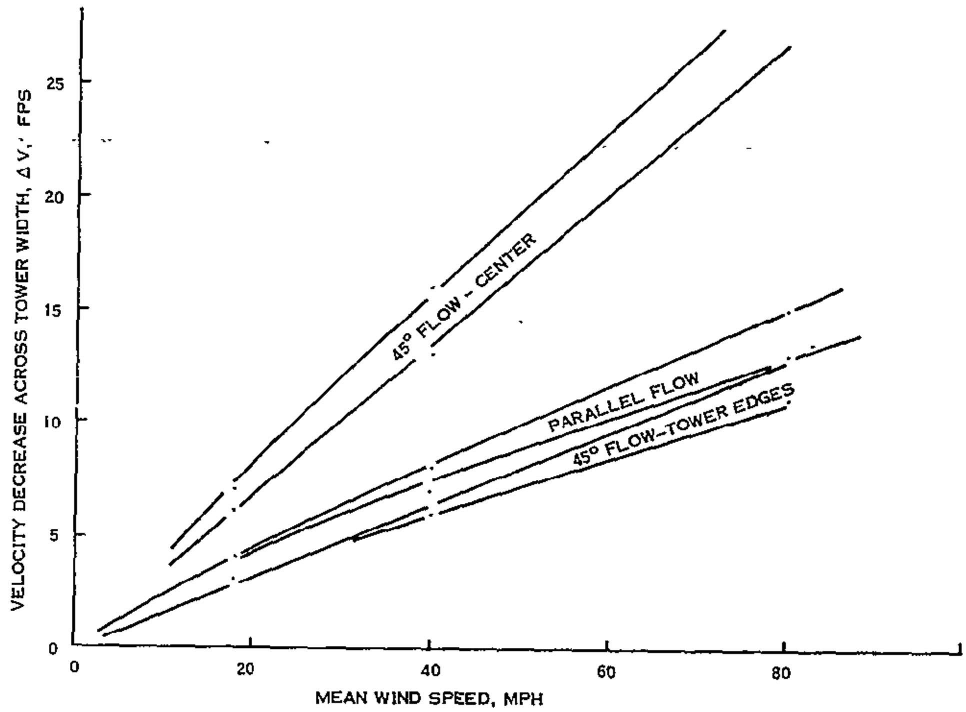

Figure 5-28. Estimated Velocity Decreases Across a Truss Tower

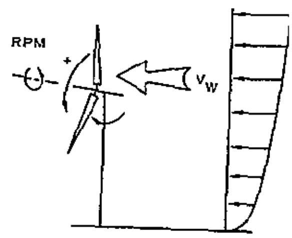

Figure 5-29. Over-Turning Moment Due to Wind Shear

frequency and magnitude. Adding to this gyroscopic loading is the moment induced by tower shadowing. These loadings are depicted on the moment - *torque plane* of Figure 5-30.

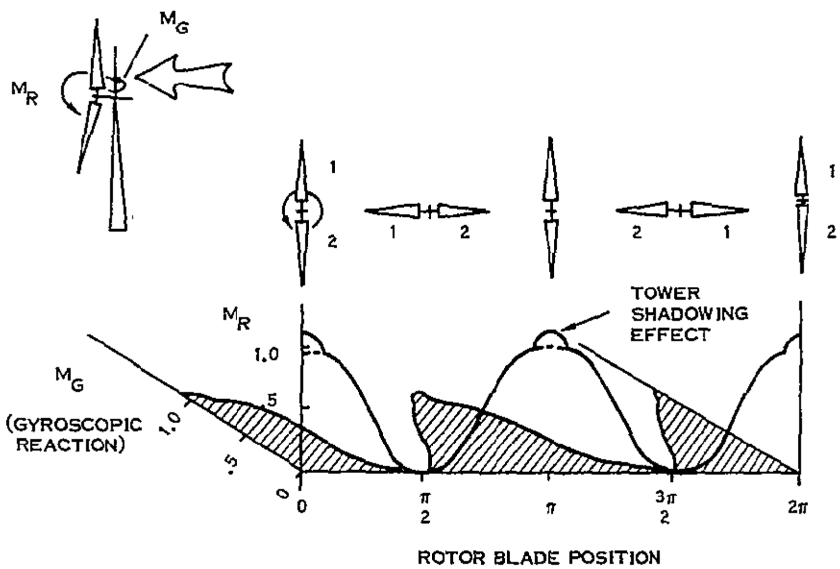

Figure 5-30. Effect of Tower Shadowing and Wind Shear on WTG Overturning and Gyroscope Moments

A further complication arises due to wind inflow. Reactions due to wind inflow produce both a moment and an in-plane horizontal force that changes direction with positive or negative changes of inflow angle. Typical magnitudes of the inflow moment as a function of windspeed and inflow angle were shown in Figure 5-26. Gyroscopic reactions due to inflow complicate the loading of the rotor, nacelle and tower in that they can add to or subtract from the reactions due to wind shear, tower shadowing or rotor overhang. One of the considerations in the preliminary design was a trade-off of rotor overhang moments so as to minimize the cumulative effect of the applied moments and gyroscopic reactions.

5.2.5.2 Rotor Blade Structural/Dynamic Analyses

Based on the rotor blade preliminary design specifications (See section 5.3.1), the resultant blade frequencies with respect to excitation frequencies are shown in Figure 5-31. This illustrates that the desired frequency placements have been achieved. The separation of frequencies and the level of the torsional frequency will insure a stable blade. The achievement of these frequency values, particularly the first two bending modes, required increases in the thickness ratio in the mid-blade ragion (comparing conceptual with preliminary design) in order to reduce weight (see Figure 3-15 and 5-39). Calculations were made of the blade deflection profile for the highest load conditions. Figures 5-32 and 5-33 show the blade displacement as a function of blade radius. The vertical axes of these plots represent a plane perpendicular to the axis of rotation through the center of the hub. The horizontal axes represent the axis of rotation. The static position represents the unloaded, non-rotating position of the blade due to the cone angle built into the blade retention at the hub. The blade deflection curves under steady state, uniform inflow velocity, are shown as the center line. Superimposed on the steady state conditions is the deflection range which results from the cyclic loadings which occur once per revolution due to the effect of the vertical wind velocity gradient and the flow aberration caused by the tower which is upwind of the turbine. The deflections shown are downwind, away from the tower. The deflections shown require that the 1500 kW turbine have a mechanical diameter of approximately 58.7 m (192.5 ft.) to achieve the required 57.9 m (190 ft.) aerodynamic diameter.

Figure 5-33 shows the blade deflection shape under cutout wind velocity. At wind velocities above rated the blade angle is changed to maintain a constant output power. This results in reducing the loads on the blade but eventually puts the outer portion of the blade at a reverse angle of attack where it is absorbing power rather than generating it. This reverses the load on that portion of the blade and the net blade tip deflection is toward the tower thus reducing the tower clearance to less than that which exists under static conditions.

It can be seen that a stiff turbine structure is required to maintain tip deflections to reasonable levels as well as to achieve the required frequencies and stability.

The bending stress distributions for the rated and cut-out wind velocity conditions are shown in Figure 5-34 in normalized form. As can be expected from the deflection curves, the actual stresses are in opposite directions for the two conditions. These stress distribution curves illustrate that stress is the

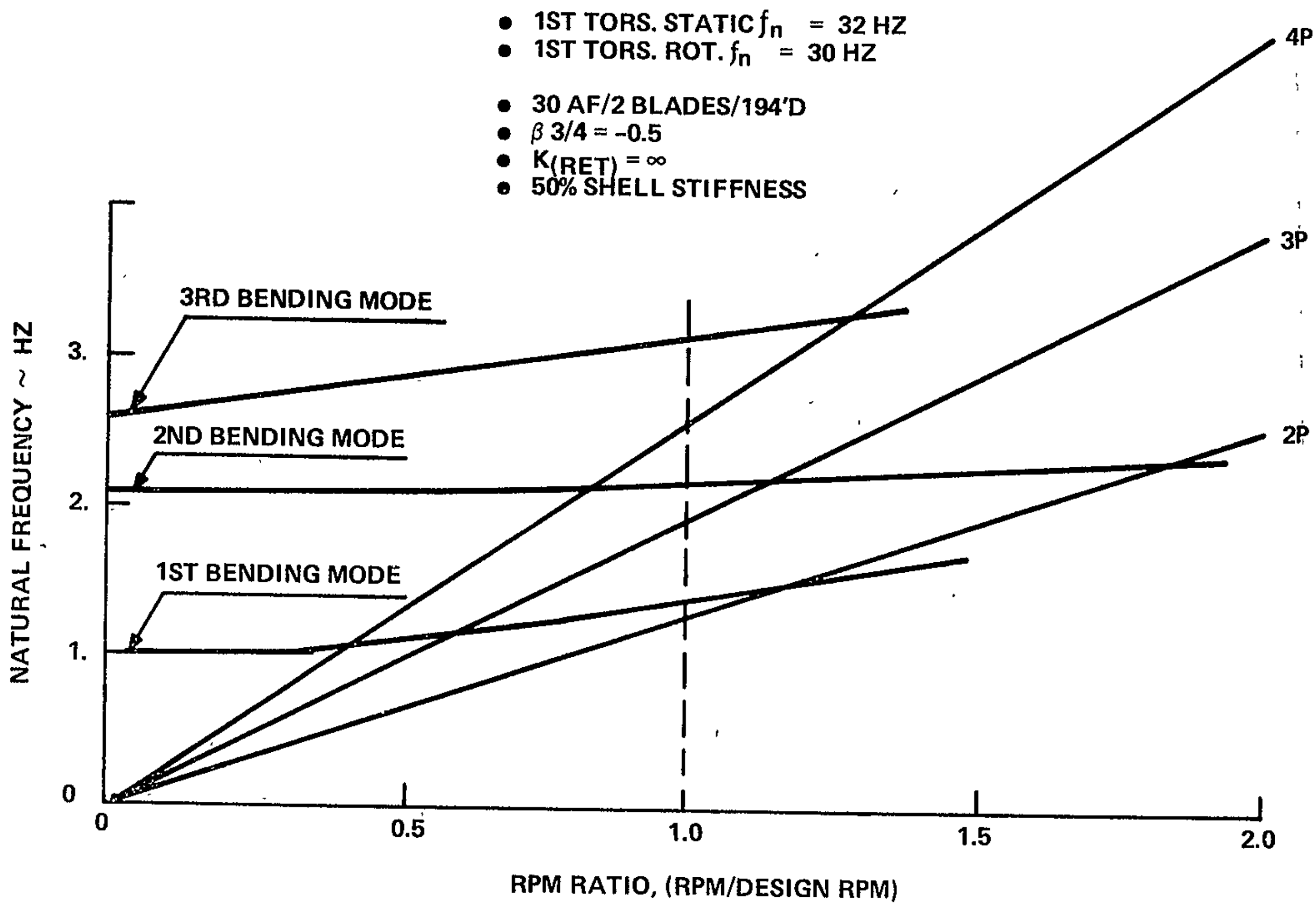

Figure 5-31. Blade Frequency Spectrum

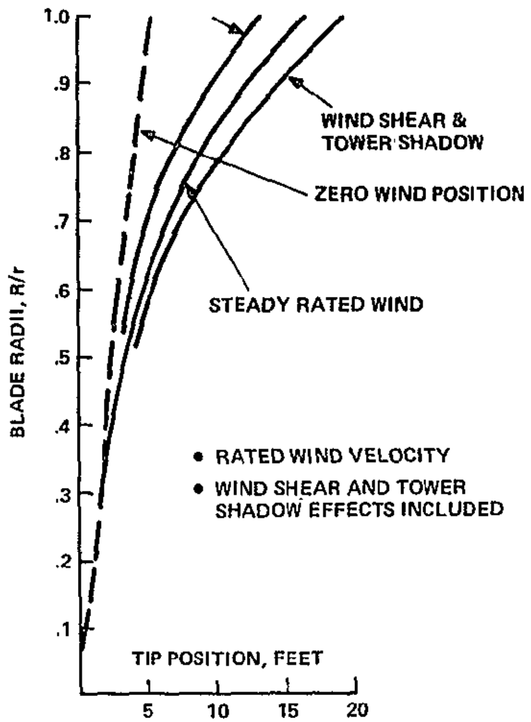

Figure 5-32. Blade
Deflection-Rated Velocity

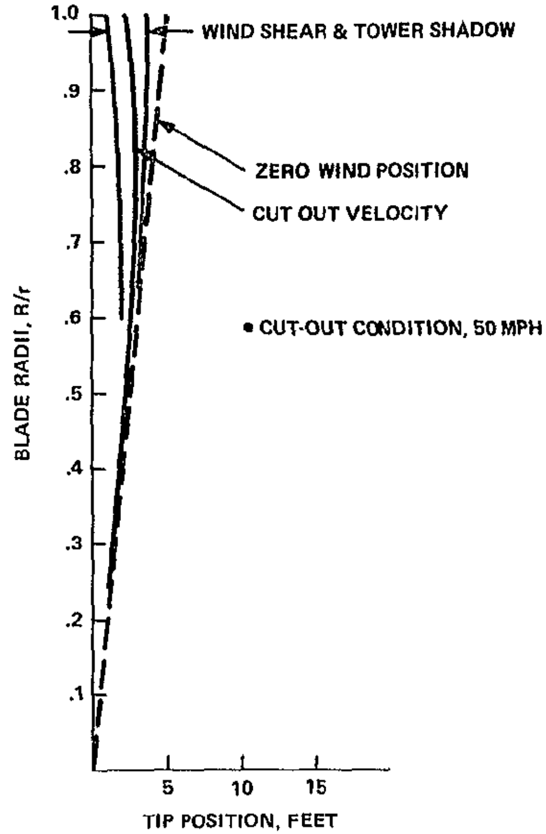

Figure 5-33. Blade
Deflection-Cutout Velocity

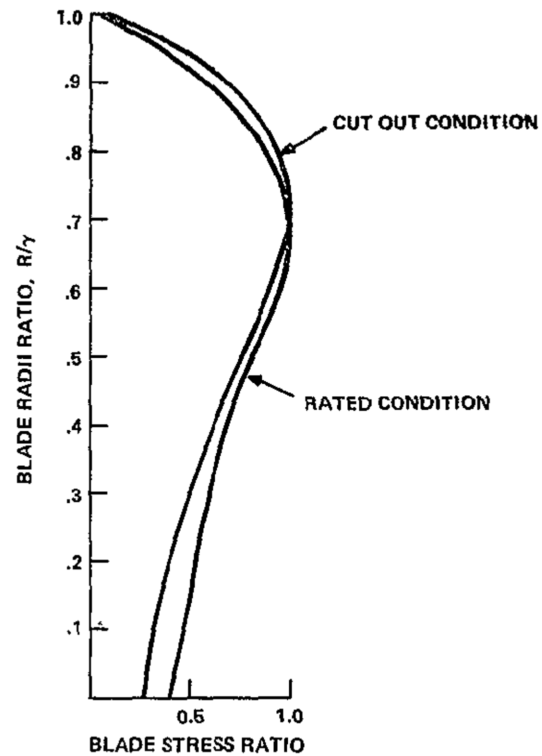

Figure 5-34. Blade
Stress Distributions

principal design factor over only a small portion of the blade. The factors which control the blade design over the various sections of the blade are shown in Figure 5-35.

Blade coning is also incorporated into the design to reduce steady state operation root bending stress by causing the deflection to be reduced by the centrifugal force on the blade in the coned position.

The preliminary blade design has been based on fiberglass/epoxy material with a ± 45 degree fiber orientation to the radial axis and using stress allowables based on past experience with aircraft propeller blade shells of similar material and fiber orientation. The assumed orientation is not necessarily optimum for the wind turbine application and the selection of the most efficient fiber orientation is a part of the detail design effort. In addition, stress allowables will be determined based on tests of the actual blade material.

The calculated stress levels with respect to the boundaries for the primary operating conditions are shown in Figure 5-36. This illustrates ample margin for the cutout condition and no margin for the rated condition. This stress/allowable relationship will be improved by increasing the preset blade cone angle slightly which will decrease the steady stress at rated and increase it at cut-out. There is ample margin for the transient conditions which may occur.

5.2.5.3 Main Shaft Dynamic Analysis

The mainshaft is the dynamic link between the wind turbine and the transmission. As such, the dynamic response of the mainshaft in bending and torsion is critical to avoid mainshaft resonance coupling with the rotor. Calculations were made to determine the mainshaft natural frequencies. These calculations were based upon the equations presented in References 5-4. The principal expression used in the design of the hollow main shaft is shown below.

$$\sigma_{max} = (16/\pi d_o^3 B) \sqrt{K_M^2 (M_{av} + k_b M_r)^2 + (T_{av} + k_t T_r)^2}$$

where $B = \dfrac{1}{1 - \alpha^4} = \dfrac{1}{1 - \left(\dfrac{d_i}{d_o}\right)^4}$

$$K_M = M_{av} + (\sigma_{YP}/\sigma_e)\ k_b\ M_r/(M_{av} + k_t M_r)$$

$= 1.5\text{-}2$ for suddenly applied, minor shocks (used 2)

K_T = same exc. T not M and k_t not k_b = 1-1.5 (used 2)

M_{av} (M_r) = bending moment, average (range)

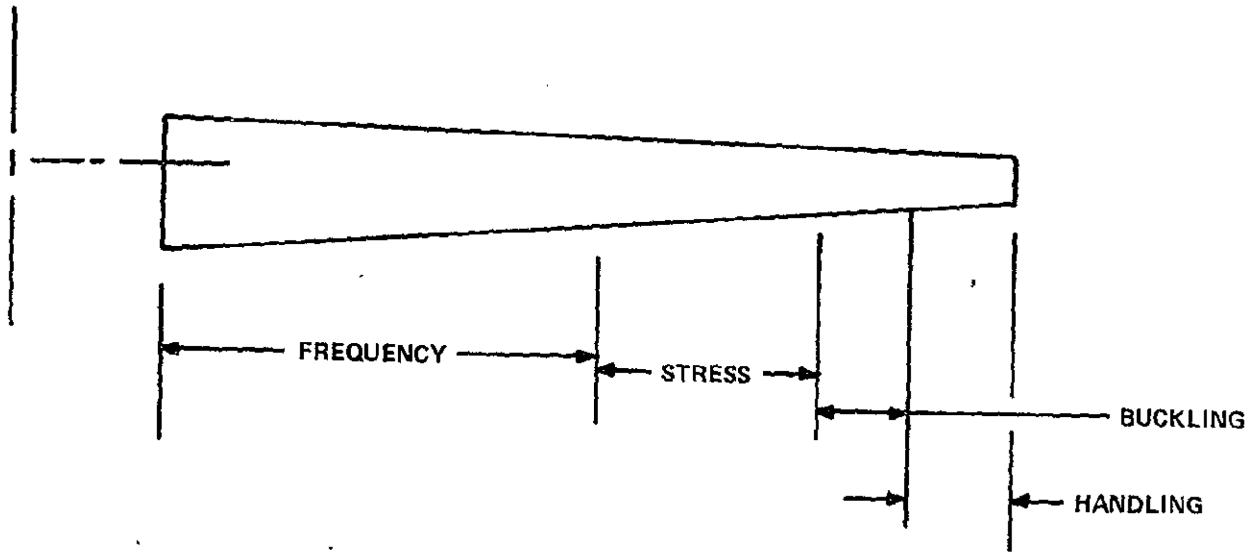

Figure 5-35. Blade Sizing Criterion

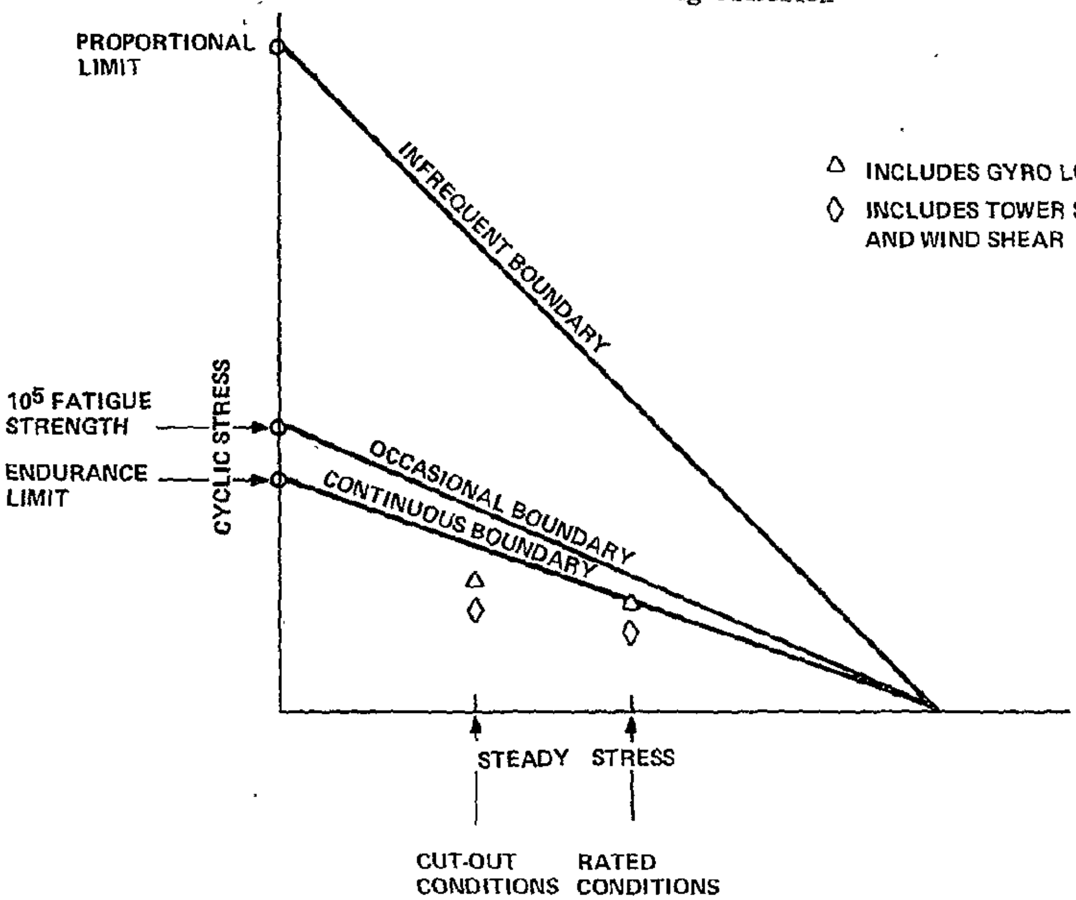

Figure 5-36. Composite Wind Turbine Blade Strength Boundaries

T_{av} (T_r) = torque, average (range)

d_o = outside shaft diameter

d_i = inside shaft diameter

k_b = stress concentration factor in bending

k_t = stress concentration factor in torsion

σ_{yp} = yield stress

σ_e = fatigue limit stress

The calculated f_n and their ratio to the main shaft operating RPM are listed in Table 5-9. While a value which is marginally acceptable exists for the 500 kW shaft in torsion, a possibility of resonant torsional operation exists for the 1500 kW shaft. This indicates a necessity for conducting a detailed shaft critical frequency calculation during detailed design. Better definition of all related inertias, especially the rotor inertia, and related torsional spring constants are required.

For the possible shafts of Table 5-9, the translational or bending critical frequencies for 1500 kW were 584-606 RPM, and for 500 kW were 588-631 RPM, all well above the operating speeds.

5.2.6 TOWER/ROTOR DYNAMIC ANALYSIS

It was recognized early in the program that the design of the rotor and tower must include an analysis of the interactions between these 2 subsystems. Failure to consider potential dynamic coupling effects could result in system failure or, as a minimum, overdesign of either the tower, rotor, or both. Therefore, an analog computer simulation of the tower/rotor was developed to understand the most cost effective design approach for the rotor blades and tower.

The basic questions which were addressed in this analysis were:

- The effect of 2 as compared to 3 bladed systems upon the tower design
- The use of a teetered as opposed to a rigid hub
- The required tower stiffness

These areas of investigation are discussed below; specific results of this work are contained in Appendix 8.2.

TABLE 5-9. POSSIBLE MAIN SHAFT DESIGNS USING AVAILABLE BEARINGS

1500 kW - $d_o{}^3/B$ = 2520.07

(1500 kW - $d_o{}^3/B$ = 2154.27)

Brg. No.	Brg. ID mm	Brg. ID in.	Shaft OD Adptr. ID in.	Shaft ID in.	X-Area in^2	Wall t in.	Weight lb/ft	$\frac{\pi}{32}(D^4-d^4)$ J-in^4	$\frac{\pi}{64}(D^4-d^4)$ I-in^4	Torsional f_n - RPM	f_n/f -(5)	
X1300	-	13.000	←	-	-	-	-	-	-	-	-	
				(4.855)	(114.221)	(4.073)	(387.9)	(2749)	(1375)	(35.8)	(1.23)	(1)
XXX72	360	14.173	13.438	-	-	-	-	-	-	-	-	
				(7.778)	(94.312)	(2.830)	(320.3)	(2842)	(1421)	(36.4)	(1.25)	(2)
XXX76	380	14.961	13.938	7.151	112.413	3.394	381.8	3448	1724	38.2	0.94	(3)
				(9.372)	(83.598)	(2.283)	(283.9)	(2948)	(1474)	(37.1)	(1.27)	
X1400	-	14.000	←	7.483	109.962	3.259	373.4	3464	1732	38.3	0.94	(4)
				(9.532)	(82.574)	(2.234)	(280.4)	(2961)	(1481)	(37.2)	(1.28)	
XXX80	400	15.748	15.000	10.642	87.774	2.179	298.1	3711	1855	39.6	0.98	
or X1500	-	or 15.000	←	(11.633)	(70.436)	(1.684)	(239.2)	(3172)	(1586)	(38.5)	(1.32)	

NOTES:
(1) Preferred 500 kW split-race bearing design <u>if</u> 4.855 in. ID is sufficient.
(2) " " " non-split-race bearing design.
(3) " 1500 kW " " " " "
(4) " " " split-race bearing design.
(5) f = 29.1 RPM for 500 kW, 40.6 RPM for 1500 kW at 1/rev. The possibility of 2/rev and higher excitations exists.

Two Versus Three Bladed Rotors

The basic options studied were two and three bladed propellor type rotors mounted on vertically cantilevered towers. The two bladed rotors were either of the hingeless or teetering hub type. The models were three or four degree of freedom analogs which focused on the lateral bending characteristics of both the blades and the WTG towers; the so called "flapping" mode was considered most critical.

All the models were limited in that torsional couplings and higher order bending modes were neglected.

In comparing the three bladed system with either case of the two bladed rotor, it is clear that shaft moments, dynamic magnification and range oscillations all generally decrease for the three bladed rotor. The tower benefits in particular from the three bladed system since, the loads are considerably alleviated. Considering that each blade in a three bladed rotor is likely to be lighter than each in a two blade system, the three blade advantage to both the rotor and tower would seem to be even greater than indicated by the models.

An overall examination of the data also reveals, as suspected, that there is a sensitivity to tower stiffness on blade loadings. Significant overall advantage can be gained from tuning the tower and blades in relation to any particular rotor speed (excitation frequency). A particularly interesting result is that for blades with a natural frequency of approximately 2.5P, tower dynamic magnifications tend to increase with increasing tower stiffness for 2 blade rotors. The implication is that as the tower is stiffened, at additional cost, it must absorb greater loads so that a stiff tower design may be no better from a strength standpoint than a tower designed on the basis of strength to begin with. Since the tower "sees" only 2ω as the fundamental excitation, it would seem as practical and more economical to design a tower with a natural frequency less than the 2ω harmonic exciting frequency if the start up and synchronizing sequence allows powering through the resonance speed. Such an approach would make the use of high strength steels more attractive in lattice towers as much less weight would be required for the same strength. Reinforced concrete towers would also become more attractive as the lower elastic modulus of concrete would become a less significant design factor. Also, with a softer tower the time for resonant amplitudes to reach the same degree of severity is greater, so that more time could be allotted for powering through the critical region. It is also well to keep in mind that for large systems, there is a greater percentage increase in tower cost to attain the same degree of additional stiffening. For the two blade case, it is clear that the tower shadow effect has a large bearing on tower loads for the rigid hub design. Use of a teetered hub does not significantly alleviate the situation. An increase of wind shear magnitude (i.e. less WTG ground clearance) tends to adversely affect every motion. The addition of wind gusting to the wind shear and tower shadow did not affect the basic results. The overall response was stable and followed the overall excursions.

For the three bladed rotor, the effects of wind shear and tower shadow are also of importance. The analysis generally shows the tower shadow effect to be significant with respect to producing dynamic load magnifications between the tower and rotor. Both the magnitude and duration of the blade unloading are important parameters in determining the harmonic content of the tower shadow forcing function. Since that should be known in relation to both blade and tower stiffness, it becomes an especially difficult problem to design a tower whose own size and shape not only determines its own dynamic characteristics, but whose size and shape determine the nature of the forcing function acting on itself. It would seem desirable to undertake a further Fourier analysis of several tower shadow situations in order to determine if there is a consistent harmonic nature to the excitations as a function of blade unloadings and shadow duration.

Several runs were made to assess the effect of reducing the nacelle weight atop the tower. For a 50% reduction in weight the tower displacements were about 25% less for the two blader while the three blader entered a resonance condition. More runs would have to be made of course, but mounting the generation equipment on the ground appears to be dynamically beneficial.

Overall, the simulations have shown that not only are there significant differences among the three rotor types studied, but significant gain is possible within any particular approach by properly tuning the rotor/tower combination to not only avoid the usual resonances, but also to reduce the magnitude and range of displacements and moments.

Teetering Vs. Rigid Hub

It was learned very early in the simulation that the totally undamped teetering hub is unstable. As soon as damping is introduced, however, the teetered natural frequency changes enough to stabilize the system. It is at this point that the teetering vs. hingeless controversy becomes more interesting. For a purely teetering hub there can be no moments transmitted to the mainshaft due to wind shear, tower shadow, inflow angle or other loadings. When damping is introduced, however, reaction forces are reinstroduced into the system such that the moments are no longer eliminated. The approach is to allow enough teetering to stabilize the motion but to minimize moments. When the teetering motion was allowed to vary during the simulation it was found that there was no change in the moments even for teetering angles of up to 15°. The net result is that a damped teetering motion of about 1° or less is all that is required. Less damping would increase the teeter amplitude but not affect the moments. Consequently teetering situations of only 0.5° were studied. An examination of those runs reveals that, over a wide range of tower and blade stiffness, one could expect about a 20% to 45% reduction in moments with a teetered hub of less than 1° motion, depending on the relative tower and blade stiffnesses. The reduction would also depend on the wind shear and tower shadow magnitudes. Greater teetering action through less dampening would be academic. It is also probable, but not confirmed in the present analysis, that aerodynamic damping would be insufficient to limit the teetering oscillations to less than 1°, and mechanical implementation would probably be necessary. Teetering of up to 1° also has the benefit of reducing displacement magnification factors for both the blades and tower. That is undoubtedly due to reduced inertial loadings, again, over a wide range of combined stiffnesses. For heavier blades that reduction would probably be even greater. It is well to note that not only are the peak dynamic deflections reduced but the displacement range is similarly reduced. That is a very important factor in designing long life blades, the relative value depending on the reversing blade dead weight loads.

Tower Design Approach

During the conceptual design phase, an assessment was made as to the degree of stiffness required in the tower design. Towers could be designed on the basis of strength which is adequate from a fifty-year load point of view. The fundamental bending natural frequency of strength towers could be above or below the fundamental exciting frequency (twice the rotor rpm). A more costly, stiffer design would meet the overriding frequency criteria that the natural tower frequency be above the fundamental exciting frequency and thus would theoretically be stronger than required from a strength viewpoint. The analytical results of this effort strongly suggests that a much more accurate concept would be to design a tuned system, where the stiffness of tower and blades were such that dynamic responses due to the varied excitations resulted in reduced dynamic loadings. A minimum is to be sought on the cost versus dynamic response curves for both the tower and blades, within the framework, of course, of avoiding operation at natural frequencies.

In summary, the major results of the dynamic analysis are listed:

1. The three blade rotor design is dynamically superior to either the hingeless or teetering 2 blade design.

2. Towers designed on the basis of strength rather than stiffness are a potentially more cost effective approach and can be technically more attractive for certain wind generator systems.

3. The totally undamped teetering hub is unstable.

4. The teetering hub approach with damping can reduce maximum mainshaft moments by 20% to 45% over the 2 blade rigid hub design.

5. The teetering hub approach with damping will generally reduce both blade and tower peak displacements and oscillating range displacements over the 2 blade rigid hub design, but not to a significant degree.

6. Teetering motions of approximately 1° or less are sufficient; further increases are of little benefit.

5.2.7 FAILURE MODES AND EFFECTS ANALYSIS

A failure modes and effects analysis (FMEA) was conducted for the major system operational modes to determine potential system failure mechanisms, the primary effect of the failure, potential secondary effects or cascading, and identify corrective action or procedures that would be incorporated into the system or component design philosophy. For each of the failure modes listed in Table 5-10, it is assumed that there is sufficient wind to achieve the function described but a system component or operation fails to perform as designed. The end results of this preliminary design has been to incorporate some of the corrective procedures into the system preliminary design.

Future failure mode and effects analyses will try to expand to the component level and assign probability of occurrence numbers and then evaluate the seriousness of the failure such that efforts may be directed to those components with the most impact on the system performance and reliability.

TABLE 5-10. FAILURE MODES & EFFECTS ANALYSIS

Failure Mode	Cause	Primary Effect	Secondary Effect	Corrective Procedure
1. Failure to align to the Wind	• Lack of external power • Lack of power to Azimuth drive • Frozen azimuth pinion bearing or azimuth gear • Ice on pinion & Azimuth gear • Fail to sense	• Loss of Electrical Power	• Potential tower strike by blade with reverse wind (180° out) • Increased operating costs • High load on nacelle and tower (inflow). • Overload on azimuth drive motor • Damage to azimuth drive shaft due to overload $V_W > V_{CO}$	• Emergency power • Provide protective cover on gearing • Provide redundant capability or weather vane mode with slip clutch • Redundant sensors
2. Failure to Achieve Breakaway	• $\beta\, 3/4 \neq$ to max breakaway torque setting • Increased breakaway torque – Dry shaft bearing – Rusted bearings – Foreign objects in bearings – Dry gear box – Misalignment of main shaft or gear box/generator – Ice build up between Rotor & nacelle – Generator bearings – Brake applied • Maintenance rotor lock bolts not removed • Rotor misaligned to wind	• Loss of electrical power generation	• Potential damage to rotor during high winds	• Reset actuator for correct β vs position indicator • Protective coating on bearing seals • Insure proper alignment or flex couplings • Brake on/off sensor • Weather boots • Locked rotor sensor or maintenance control procedure
3. Failure to Synchronize	• Loss of speed control (Pitch) • Reverse rotation (Wind 180° away) • Loss of circuit breaker • Loss or false sensing on frequency and phase • Loss of Voltage Regulation • Loss of Exciter • Loss of Rotating rectifier	• Loss of Electrical Power Generation	• Potential overspeed • Potential Tower strike • Potential Loss of Control power with • Potential for overspeed structural loads • Potential damage to voltage regulator. • Damage voltage regulator	• Exercise speed control prior to startup • Brake to shutdown on loss of speed control sensor • Provide mechanism or procedure to prevent reverse rotation • Use proportional control brake as back up speed control • Use squirrel cage for back-up start sequence • Redundant synchronizing sensors
4. Failure to Cut-In	• Loss of circuit breaker • Failure of Pitch change mechanism • Failure or loss of control logic • False Output of sensor	• Power Generator Loss	• Potential damage at high winds	• High reliability components • Control System redundancy • Internal signal test
5. Failure to Maintain Power Regulation	• Loss of pitch change mechanism • Loss of alignment control • Loss of voltage regulator or exciter	• Loss of Electrical Output	• High current surges • High thrust loads on tower • Overload outage • Voltage Dip • Insulation degradation • Overtemperature in generator	• Redundant Power control system • Improve component reliability
6. Failure to Cut-Out	• Pitch Failure • Circuit breaker fails to open • Control failure • Loss of sensors	• Generator Overloads • Rotor overloads generator motoring • Generator and rotor overloads	• Network line current surges • Damage to rotor	• Failsafe Pitch change mechanism • Redundancy in controls and sensors
7. Failure to Feather $\beta_{3/4} = 90^{\circ}$	• Failure of pitch change actuators • Loss of sensor • Failure of hydraulics • Control failure	• High blade and tower loads	• No restart ability	• Failsafe pitch change mechanism • Sensor and control redundancy
8. Failure to Generate Power	• Short circuit in Generator • Network short • Failure of mechanical power transmission	• Potential overspeed condition	• Loss of power output	• Feather rotor blades
9. Failure of Main Shaft Couplings	• Fatigue • Overload • Corrosion	• Rotor overspeed Rotor failure	• Loss of power output	• Improve reliability
10. Failure of High Speed Shaft Coupling	• Fatigue • Overload due to misalignment • Corrosion	• Loss of generator output • Overspeed	• Loss of power output	• Improve reliability

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5.3 SUBSYSTEM DESIGN

The Wind Turbine Generator (WTG) system is composed of five basic subsystems: rotor, mechanical power transmission, electrical, control, and tower, and approximately 170 components, as listed in Appendix 8.4. Preliminary designs of the five basic subsystems are described in the following sections with design descriptions of major components and specifications for many minor components. Both 500 kW and 1500 kW WTG designs are covered, and although the system designs are basically similar, scaling and operational parameters affect component specifications, cost and weights.

5.3.1 ROTOR

Definition of the two rotor systems for both the low and high power systems is summarized in Table 5-11 in terms of system operational and geometric specifications. It is noted that system operational and geometric specifications do not reflect the preliminary blade tip speed constraint of 106.7m/sec imposed during the parametric analysis and discussed in Section 4.3.1.2. Although the 500 kW system is not affected, the tip speed and RPM increased significantly for the 1500 kW system, which requires increased blade stiffness to control deflections and frequencies.

The loading conditions for rotor design are listed in Table 5-12. The characteristic rotor subsystem assembly drawing is shown in Figure 5-37, which shows the major rotor components: blade, hub, and pitch actuation mechanism. Aerodynamic performance of the blade was presented in Figure 5-7.

5.3.1.1 Key Design Features

Key design features of the rotor blades are: (1) a propeller type structure with spar/shell construction for optimum aerodynamic performance and structural reliability, (2) a fabrication process of filament winding with fiberglass material for light weight, durable blades and low cost, and (3) a two-piece assembly design for ease of transportability and handling. However, the design approach is equally applicable to a single piece assembly; a final determination on one versus two pieces would be made during a Final Design upon the basis of cost. The blade design also includes features of lightning protection and aircraft beacons at the blade tips.

Key design features of the rotor hub are: (1) welded tubular steel elements for high stiffness and low weight/cost; (2) a subassembly breakdown capability for ease of manufacturing, shipping, and installation, and (3) self-lubricating blade moment and thrust bearings for minimal maintenance requirements.

The key design features of the pitch actuation system are: (1) a self-locking system at any given blade setting, (2) a mechanical system that is powered by the rotation of the main rotor shaft, (3) automatic feathering when control power is lost, (4) a blade angle feedback signal, (5) a blade-to-blade angle adjustment, and (6) excess capacity for jam breaking. These features demonstrate the functional mode redundancy that must be built into a variable pitch control system.

TABLE 5-11. ROTOR DEFINITION SUMMARY

PARAMETER	SYSTEM 1	SYSTEM 2
Rated Generator Power, kW	500	1500
Rated Wind Speed, m/s(mph)	7.3(16.3)	10.1(22.5)
Cutout Wind Speed, m/s(mph)	17.9(40)	22.4(50)
Diameter at Rated Wind Speed, m(ft)	55.8(183)	57.9(190)
RPM	29.1	40.6
Rated Velocity Ratio	9.0	9.5
Tip Speed, m/s (ft/sec.)	84.7(278)	121(398)
Turbine Tip Ground Clearance, m(ft.)	15.24(50)	15.24(50)
Number of Blades	2	2
Blade Activity Factor	30	30
Blade Airfoils	NACA 230XX	NACA 230XX
Blade Angle Change Rate, rad/s	7	7

TABLE 5-12. ROTOR LOADING CONDITIONS

(A) <u>Limit Loads</u> (Yield Limit)

(1) No rotation, blades providing maximum lift, positive or negative at maximum wind velocity. (26.8 m/s /60mph/@9.1 m/30 ft./)
(2) Rotating at rated rpm, full load and cutout wind speed, blades go to feather instantaneously.

(B) <u>Intermittent Loads</u> (10^5 Fatigue Strength)

(1) Rotating at rated RPM and full load with wind velocity 2.5 times rated wind velocity, wind drops to 0 mph.
(2) Rotating at full load and rated rpm under rated wind velocity, wind velocity doubles instantaneously.

(C) <u>Continuous Loads</u>

(1) Rotating at full load and rated rpm the in-plane gravity moment is applied as a completely reversing load.
(2) Rotating at full load and rated rpm under rated wind velocity at a 15° in-flow angle, including case 1 above.
(3) Rotating at full load and rated rpm under rated wind velocity, wind velocity drops to 0 mph instantaneously, including case 1 above (Tower Shadow).

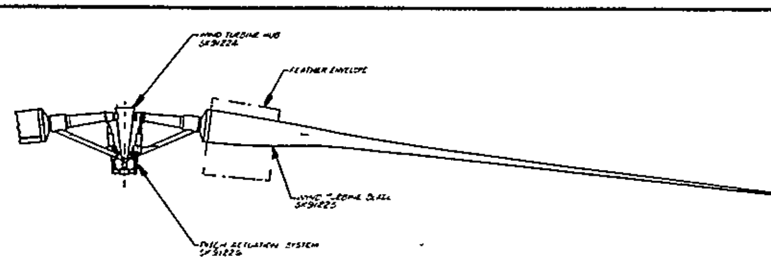


Figure 5-37. Rotor Assembly Design

5.3.1.2 Blade Design Description

The preliminary design of the rotor blade was concentrated on the 1500 kW system, and information was derived for the 500 kW system by sealing. Using the rotor specifications in Table 5-11 and the initial work accomplished during the conceptual design, alterations were made with configuration, thickness, and material distributions as prime variables to meet the requirements with respect to stress, stiffness, buckling, handling impact damage, and weight. These studies resulted in the selection of a spar/shell type of blade in the configuration shown in Figure 5-38. A full length spar is used and both the spar and shell are made by filament winding.

The inboard retention area is circular with a wound-in metal fitting. To facilitate manufacture, handling, and shipping, the blade is made up of two pieces with wound-in fittings in the spar similar to the inboard retention and a metal piece to effect a bolted joint.

Provisions are made for winding, under the final wrap, a lightning grounding strap and wiring for a tip warning light.

The tentative selection of the fiber for the blade winding is an E-type glass with K filament and 20 end roving. This material is sufficiently strong and a cost effective structural composite will result. The resin is composed of Epon 828 modified with Epon 815 to achieve low viscosity and relatively low temperature curing.

The metal retention pieces are cut from steel plate stock and the blade joining piece from aluminum plate.

The blade shape characteristics are shown in Figure 5-39. These are the same as the 30 AF blade configured previously with respect to planform taper and twist but the blade thickness ratio has been increased, particularly in the mid-blade region. This thickening was required to achieve the required structural characteristics at reasonable blade weights. The increase in thickness did not have a significant effect on performance.

The structural design description for the blades was reviewed in Section 5.2.5, System Dynamic Analysis.

The blades will be fabricated using proven filament winding practices and resin cure methods. A fiber delivery system will be used which provides controlled transfer of fiberglass rovings to the winding machine from multiple spools containing individual tension control adjustments. An in-line applicator cup applies resin to the fiber just prior to mandrel contact forming an impregnated tape. A programmed winding machine will apply the fiber tape on a mandrel at the selected angle. Layers of fiber will be applied from the root to varying radial positions to develop a structure with varying wall thickness as a function of radius.

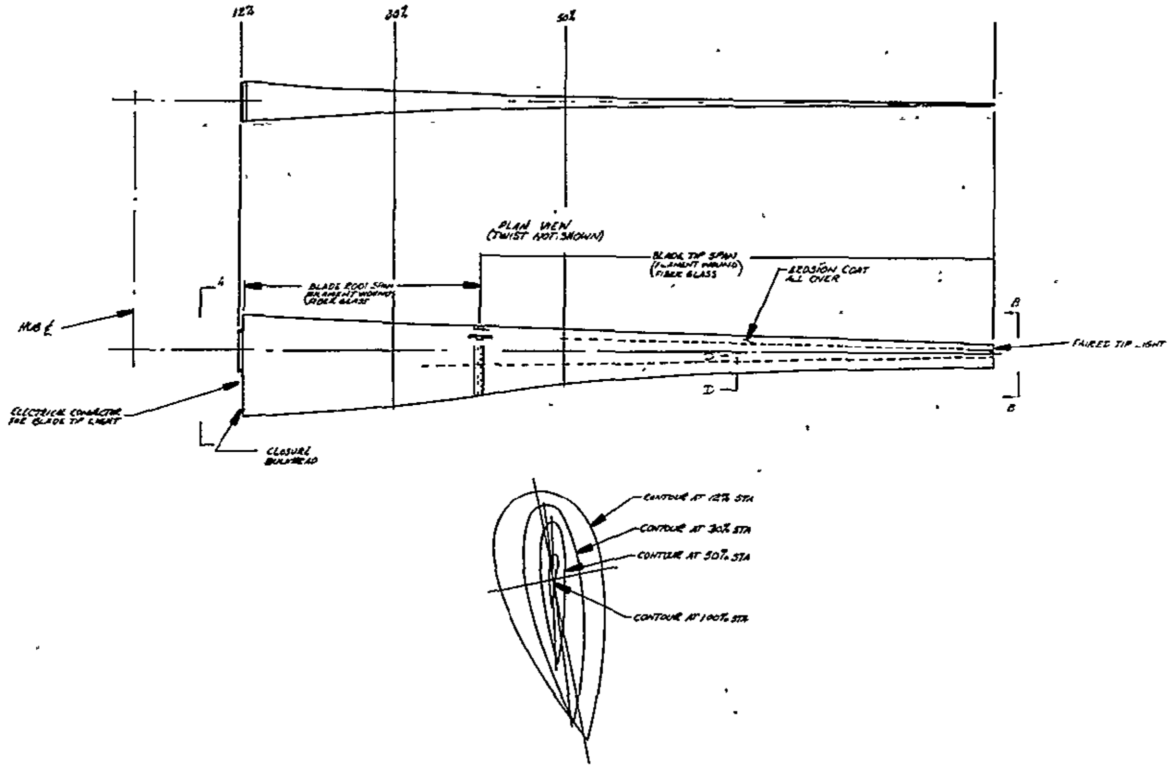


Figure 5-38. Blade Assembly Drawing

– PRELIMINARY –

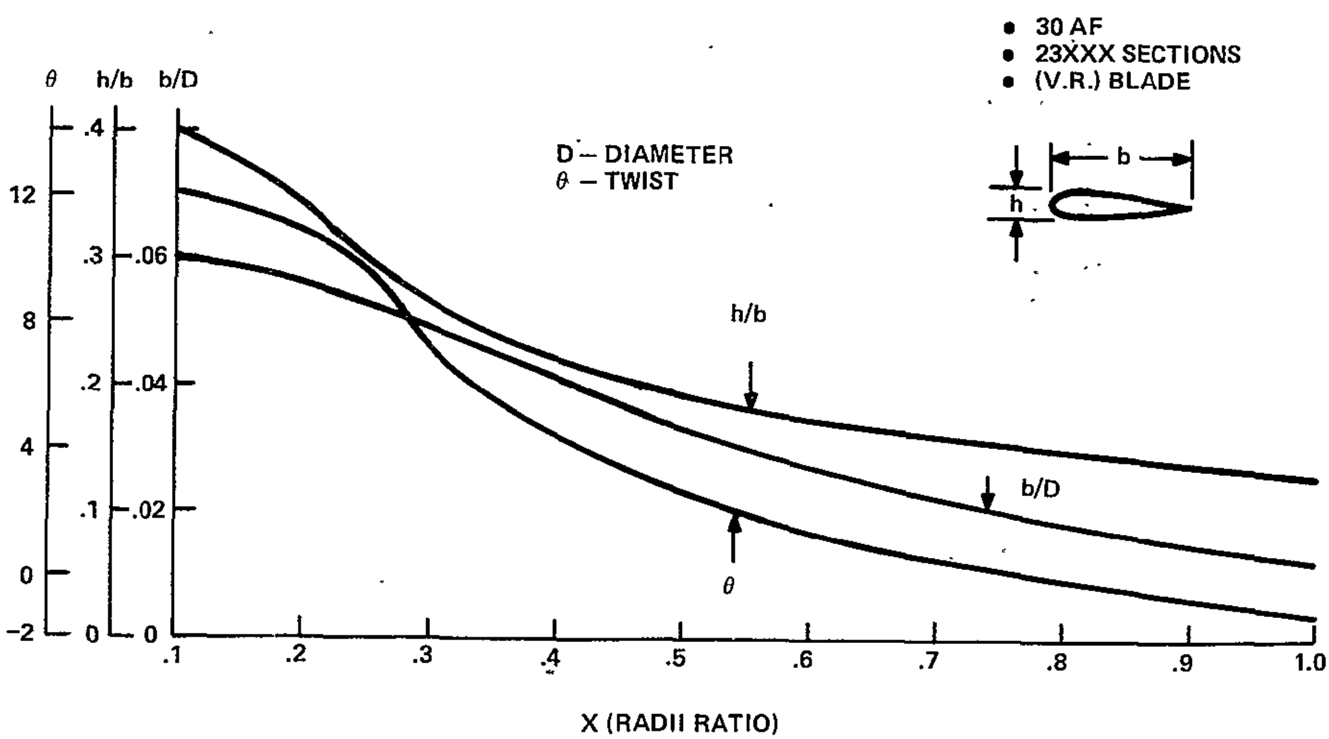


Figure 5-39. Blade Shape, Preliminary Design

The spar mandrel is shaped to the spar inside dimensions and is supported at both ends in the machine. The mandrel is in several pieces to permit extraction from the finished spar through the butt end opening. The spar is wound first, then additional mandrels are added to form the leading and trailing edge blade shapes. The shell is then wound over the spar and mandrels. On completion of the shell angular windings, a layer of chordwise windings are applied over the entire blade shell length for compaction of the helical windings and to provide a smooth surface. These chordwise windings also secure the lightning strap and wiring in the preformed trailing edge cap. Peel-ply cloth, vent cloth, plastic sheet, and binder tape are applied for the curing and finishing phase. Curing is accomplished in a hood with heat lamps placed over the blade.

Retention flanges at the root end and mid-blade joint are formed in the course of the spar winding process and the metal fittings become bonded-in components of the blade although the bonded joint is not considered a part of the eventual retention joint. Following the cure cycle, the outer cure wrappings are removed. The retention ends are machined to provide the final inside dimensions and provide a flat outside interface surface and the holes for retention bolts are drilled and tapped. The mandrels are then removed.

The blades are painted on the exterior surfaces with an electrically conductive, erosion resistant coating. The inside surface is protected from moisture attack by a gel coat applied to the mandrels prior to winding.

The blade is balanced by plates attached at the retention end which adjust the blade weight and the c.g. location in three axes to the specified tolerances.

The process described above avoids the need for expensive tooling for forming and curing the external form. However, this is possible because the selected airfoil does not have concave contours. It was this factor that made the final selection of the NACA 230XX airfoils over the 44XX type. Both of these airfoils are acceptable from both aerodynamic and structural considerations.

5.3.1.3 Hub Design Description

The selected hub design is shown in Figure 5-40. This concept is the same in principle as that developed in the Conceptual Design Task. However, several changes were made as follows:

- Because of the large size of this hub, approximately 6.1 m (20 ft.) in diameter, it was found that it was more efficient to accomplish the in-plane and out-of-plane bracing with tube structure rather than plates, since the latter require excessive size and weight to resist buckling.

- Also because of the size, it was concluded that to facilitate fabrication and shipping the hub should be made up of subassemblies.

- System studies resulted in a preference for a bearing mounted shaft support rather than a single bearing in the turbine plane.

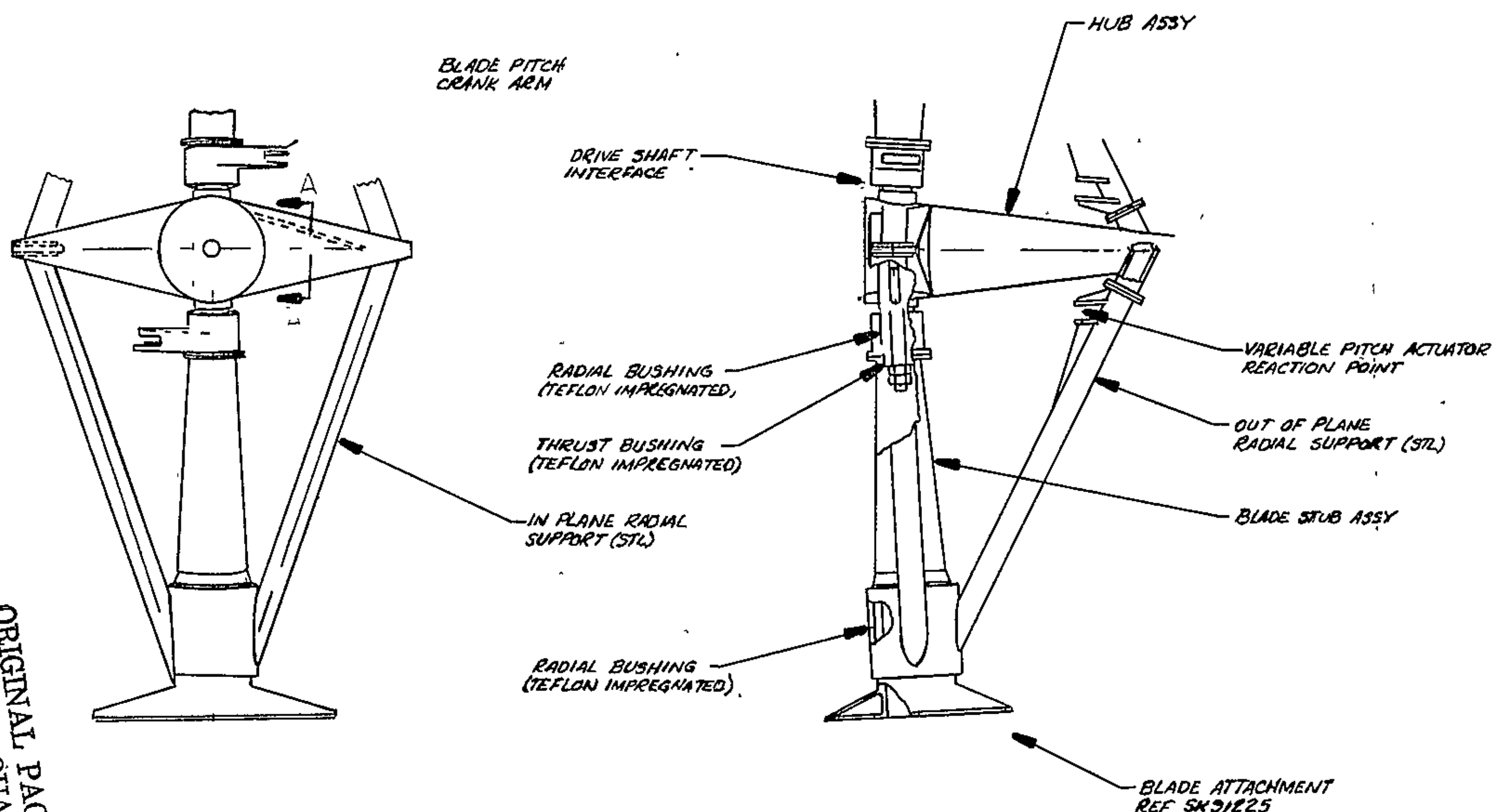


Figure 5-40. Hub Assembly Design

- The blade centrifugal restraint was changed to a Teflon fabric coated ball joint rather than a wire rope. This simplified the structure and reduced weight and cost.

The hub is made up of five basic components. The center tubular hub has an integral cross member which ties the two blade attachments together centrifugally and an outrigger structure to support the in-plane braces and carry torque loads. Two tripod assemblies support and brace the outer blade moment bearings. Two blade stub assemblies provide the interface with the blade attachment and carry the centrifugal and moment loads through the bearings to the hub structure.

The hub structure is fabricated from standard commercial steel plate, bar stock, and tubing. The bearings utilize standard Teflon fiber bearing material bonded in place.

The hub center section is made from several sections of tube sizes which, with minor machining, can be telescoped and welded to form a generally conical shape. Bar stock is inserted through holes in the center body and welded to form the blade stub inboard interface. The outer ends of this bar are machined to provide the design blade cone angle. Steel plates are welded to the center body, and to each other, to form a tapered box structure from the center body to the in-plane tripod mount.

The tripods are made from three steel tubes which are welded to a section of larger tube which forms the outer blade support. The inner ends of the struts have welded plates which are machined and drilled to provide a bolted flange joint at the hub. Plate sections are welded to the inner portion of the out-of-plane braces to provide support for the pitch change actuators.

The blade stub assemblies are welded from tubes and plates. The mid-section is made from plate which is cut, rolled, and welded to form a conical section. The bearing surfaces are machined and Teflon bearing material is bonded in place.

The subassemblies are fabricated in jigs which will assure proper dimensional control of the interfaces when the complete hub is assembled. For shipping, the center hub with inboard radial bushing attached, and the tripods with blade stubs in place can be separate assemblies. At the installation site, the hub assembly is made by bolting the three subassemblies together at the attachment flanges.

5.3.1.4 Pitch Actuation System Description

Based on the experience gained in the conceptual design studies and a more detailed study of system operation and loads of the candidate preliminary designs, some revisions were made to the previous pitch change system concept. The primary considerations in evaluating the pitch change system were:

- The failure to change pitch under operating conditions above rated wind speed or in gusts above rated could result in overloading of the blade or high over-speeds of the rotor when the load is cut off or both. Accordingly, a high system reliability is required.

- High pitch change reliability is also required for shutdown feathering to prohibit excessive blade and tower loads during non-operating high wind conditions. Feathering reliability is also required to insure turbine shutdown under other malfunction conditions such as loss of load or loss of control capability. The shutdown capability should not be dependent on the availability of power external to the turbine system.

- The pitch change system should not require power to maintain a fixed blade angle. Since the majority of operating time is at fixed blade angle, a requirement for power to maintain blade angle, which is the case in many hydraulic actuation systems, could represent a significant loss in net system energy delivery.

- An excess of pitch change load capability should be available to clear jams such as icing in the blade bearing areas.

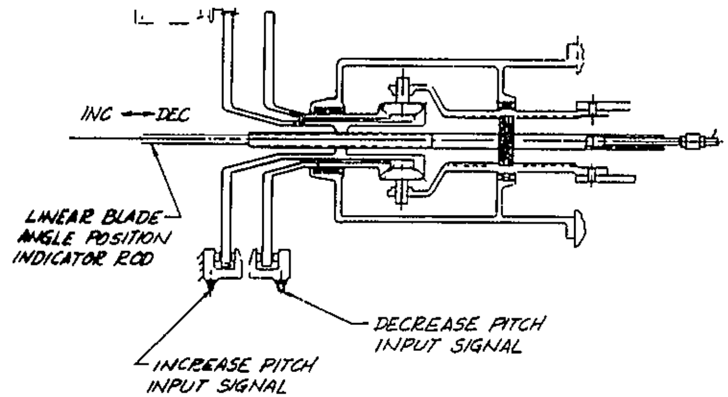
- A feedback signal of blade angle position should be available for the control system.

- Component costs should be low.

The system concept to be described is shown on Figure 5-41. The pitch change actuator, one per blade, is an industrial worm gear actuator. This actuator has a rotary shaft input to a worm gear mesh which in turn is coupled to an Acme threaded shaft. The rotary shaft input is thereby converted to linear motion of the output shaft. The high ratio worm gear input makes the actuator self-locking and loads applied to the output shaft will not result in shaft movement. Thus, no input torque is required to maintain position. The actuator is mounted in a pivot bracket on the hub out-of-plane brace and the output shaft is connected to the blade trunnion arm on the blade. The threaded shaft is protected from the elements by an accordion pleated boot on the output end and a fixed metal pipe cover on the retraction side of the gearbox. At installation, adjustment can be made either at the rod end fitting or by manually turning the actuator input shaft so that both blades can be accurately rigged to the same angle. This is an important adjustment which will minimize aerodynamic unbalance of the turbine.

Each of the actuators are connected by two angle gearboxes and a shaft with universal joints to the T gearbox mounted to the forward end of the hub. Torque limiting slip clutches are provided to protect the pitch change power system from any severe jam overload, as yet unidentified. All of the components in this part of the system are low cost industrial catalog items. The T box has an input shaft connecting to a shaft from the pitch change power system. Rotation of this input shaft with respect to the hub will drive the actuators to increase pitch or decrease pitch depending on the direction of input shaft rotation with respect to the hub.

The pitch change input shaft extends from the T box through the hub, the main drive shaft, and the main gearbox to the pitch change power unit which is mounted



SCHEMATIC
PITCH CHANGE POWER SUPPLY

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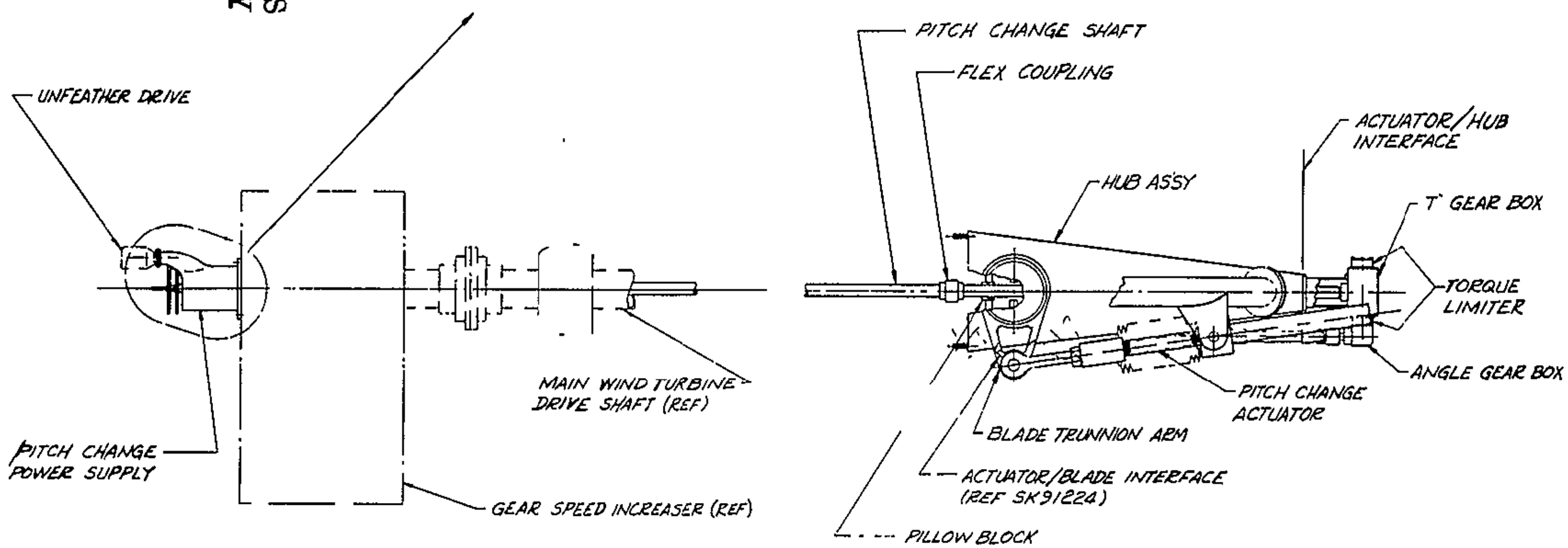

Figure 5-41. Pitch Change System, Design

on the gearbox housing. The pitch change shaft connects to the power unit center shaft by means of internal splines which permits the center shaft to translate with respect to the pitch change shaft while transmitting torque. In the power unit, the center shaft is connected through a threaded member to the outer shaft which is driven by the main turbine drive shaft. Thus, relative rotation of the center shaft and the drive shaft will cause translation of the center shaft which provides a blade angle position feedback. The center shaft is also connected through splines to an automotive type disc brake, the caliper of which is spring loaded to the "on" position and hydraulically actuated to the "off" position. When this brake is actuated it stops rotation of the center shaft and the pitch change shaft; the rotation of the turbine causes the pitch change actuators to be driven toward increase pitch. No external power or pressure signal is required for increase pitch all the way to feather. It should be noted that for feathering no fixed blade stops are required, since, if the blades over-feather, the rotation of the turbine will reverse, thereby reversing the direction of the pitch change actuators toward decrease pitch. The feathering action, therefore, automatically seeks zero RPM rather than a fixed feather angle.

For decrease pitch a second disc brake is provided. In this case the caliper is spring loaded off and hydraulically actuated on. This brake is connected to the center shaft by way of gears whose shafts are driven by the outer shaft. These gears are of the same type and size as those used in automotive rear ends to provide for differential rotation of the rear wheels. When the decrease pitch brake is applied, it stops the input mesh to the differential gears. The gears, driven by the outer shaft, also rotate on their shafts and drive the center shaft in the same direction as the turbine rotation, but at twice turbine shaft speed. Thus, the relative rotation of the pitch change shaft and the turbine causes the pitch change actuators to be driven toward decrease pitch.

The maximum pitch change rate is 0.122 rad (7 deg)/sec. Lower pitch change rates can be achieved by modulating the hydraulic signal inputs by intermittent energizing of the brakes.

For unfeathering under non-rotating conditions, an electric motor and an automotive engine starter gear drive engages gear teeth on the increase pitch brake and drives the center shaft in the decrease pitch direction. This would position the blades at the maximum static torque blade angle as indicated by the blade angle angle feedback position and would then be disengaged. Further pitch decrease after turbine rotation starts would be handled by the normal decrease pitch brake action. If it should be found desirable to have a non-rotating increase pitch capability, a second motor and gear drive could be added to turn the increase pitch disc in the opposite direction.

The system described meets the primary requirements previously discussed. The power train is entirely mechanical using industrial components which can be very conservatively rated with a small effect on cost. Pitch change power is derived from rotation of the turbine itself which requires no external power sources. Maximum reliability is assigned to the increase pitch function which is a direct connection not dependent on the function of the differential gears. The spring loaded increase pitch brake will automatically feather the blades in the event of loss of control system input signal. The selected actuator and drive train components will provide an excess of force capability to overcome binding or jam loads. A mechanical blade position feedback signal is provided for the control system. The use of standard industrial and automotive components results in a low cost system.

As discussed previously, the hub mounted components of the pitch change system are industrial catalog items except for the relatively simple brackets which support the gearboxes at their respective locations. The components are assembled at the installation site after the hub and blades are assembled. The actuators are installed in their brackets and adjusted as necessary to permit connection to the blade trunnion. They are then adjusted to align each blade with its respective rigging point. The connecting shafts and gearboxes are assembled and attached to the aft end of the hub. The pitch change shaft is installed in the hub assembly where it is guided into a spline connection at the T gearbox.

After installation of the turbine, the pitch change shaft section is installed through the gearbox and shaft with the use of internal guides to connect with the shaft in the hub assembly.

The pitch change power assembly is supported in a housing which, in small quantities of manufacture, would probably be fabricated from a weldment of commercial standard steel tubing and plates. The detailed design would be directed toward the adaptation of automotive gears, brakes, starter gear, and other commercially available components.

The power unit is assembled at the factory and functionally tested. The unit is adjusted to set the blade angle feedback position to correspond to the blade angle on the turbine assembly used for rigging the actuator system. During assembly, the unit is installed on the gearbox making connections to the pitch change shaft and main shaft of the gearbox as it is moved into position.

The estimated costs and weights for the two turbine assemblies are shown in Table 5-13.

TABLE 5-13

ROTOR SUBSYSTEM COST/WEIGHT ESTIMATES

Power Level, kW	Weight, kgs (tons)	Cost, $
1500	16,360 (18)	185,000
500	12,730 (14)	154,000

5.3.2 MECHANICAL POWER TRANSMISSION SUBSYSTEM

The mechanical power transmission (MPT) consists of those components necessary to transmit and speed condition the turbine rotor power such that the output is compatible with the generator requirements. In addition, the system design and component selection must also meet the requirements listed in Section 2.0 for load life, commercial availability, maintenance and cost. Figure 5-42 shows the MPT components and specific system configuration.

The key design features of this system are:

- Hollow main shaft
- Two bearing support
  - No. 1 bearing is a radial bearing
  - No. 2 bearing is radial and thrust bearing
- Modular flex coupling on low speed side
- Vertically mounted parallel shaft speed increaser with three-piece housing and hollow bull gear shaft
- High speed shaft failsafe hydraulic disc brake

The loading condition and operating characteristics used to develop the system design are shown in Table 5-14, load factors are in parenthesis.

5.3.2.1 Gearbox Speed Increaser

Subsequent to detailed design calculation utilizing vendor catalogues, applications data, and system loading specifications the speed increasers shown in Table 5-15 were selected.

Design specifications for the speed increaser are given in Reference 5-7. Besides the obvious functions of providing proper torques and rotating speeds, the following specialfunctions will have to be requested of the gearbox manufacturer:

- Both the input and output shafts are to extend through the gearbox housing (LR-LR arrangement).

- The low-speed shaft must be hollow to allow placement of the blade pitch control shafting, and should be tempered to high strength to minimize the weight and cost of the gearbox/main shaft coupling.

- Vertical orientation of the shaft centerlines is desired so that the nacelle weight moments can be balanced. This will require reorientation of the mounting pads and the housing split lines to minimize leakage and maximize maintainability.

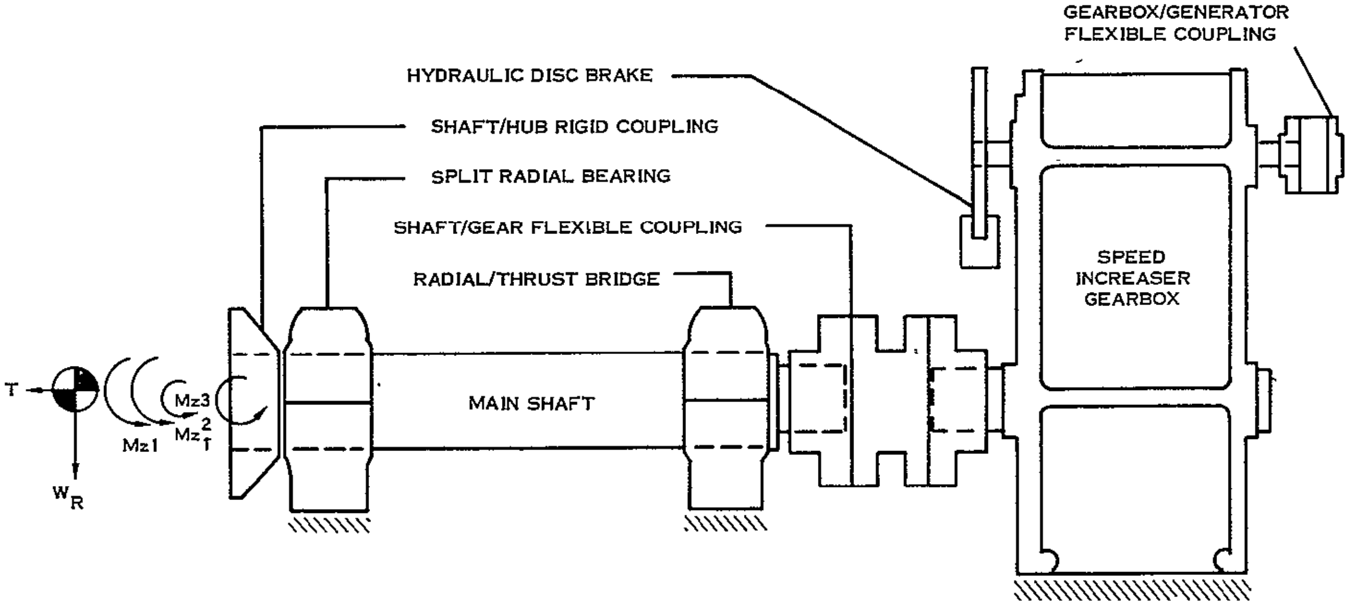

Figure 5-42. Mechanical Power Transmission Subsystem

TABLE 5-14. MECHANICAL POWER TRANSMISSION SPECIFICATIONS

FUNCTION

- TRANSMIT TORQUE FROM ROTOR SUBSYSTEM TO GENERATOR
- PROVIDE PROPER RPM TO GENERATOR SHAFT

DESIGN	500 KW	1500 KW	
INPUT RPM	29.1	40.6	KLB-FT
INPUT TORQUE	134 (1 ± 1)	3.7 (1 ± 1)	
GEARBOX SPEED INCREASE RATIO	61.9	44.3	
FORCES - G LOAD (ROTOR WT.)	27.6	35.1	KLB
- WIND INFLOW	1.9 (1 ± 0.7)	4.5 (1 ± 0.7)	KLB
THRUST	19.0 (1.4 ± 0.7)	44.6 (1.4 ± 0.7)	KLB
MOMENTS - AZIMUTH CHANGE	100	134	KLB-FT
- WIND SHEAR	400 (1.4 ± 0.5)	400 (1.4 ± 0.5)	KLB-FT

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TABLE 5-15. WIND TURBINE GENERATOR SPEED INCREASES

PARAMETER	500 kW	1500 kW
Input (rotor) speed, RPM	29	40
Output (generator) speed, RPM	1800	1800
Speed increase ratio	29	45
Service factor (min.)	1.50	1.50
Design HP	745	2235
Design HP x service factor (min.)	1118	3353
Potential Vendor	Horsburgh & Scott	Phila. Gear
Catalog No.	360-T	22HP3
HP at 1800 RPM	1202	3726
Allowable service factor	1.61	1.67
30-yr. failure rate	8.6%	13%
Weight, lb.	24,200	46,000
Vertical - Height, in.	124	125
- shaft separation, in.	73	67
Axial length, in.	50	58
Depth, in.	71	81
Efficiency	0.96	0.96

- Special mounting pads and lifting hooks are desired to mate with bed-plate and for ease of maintainability.

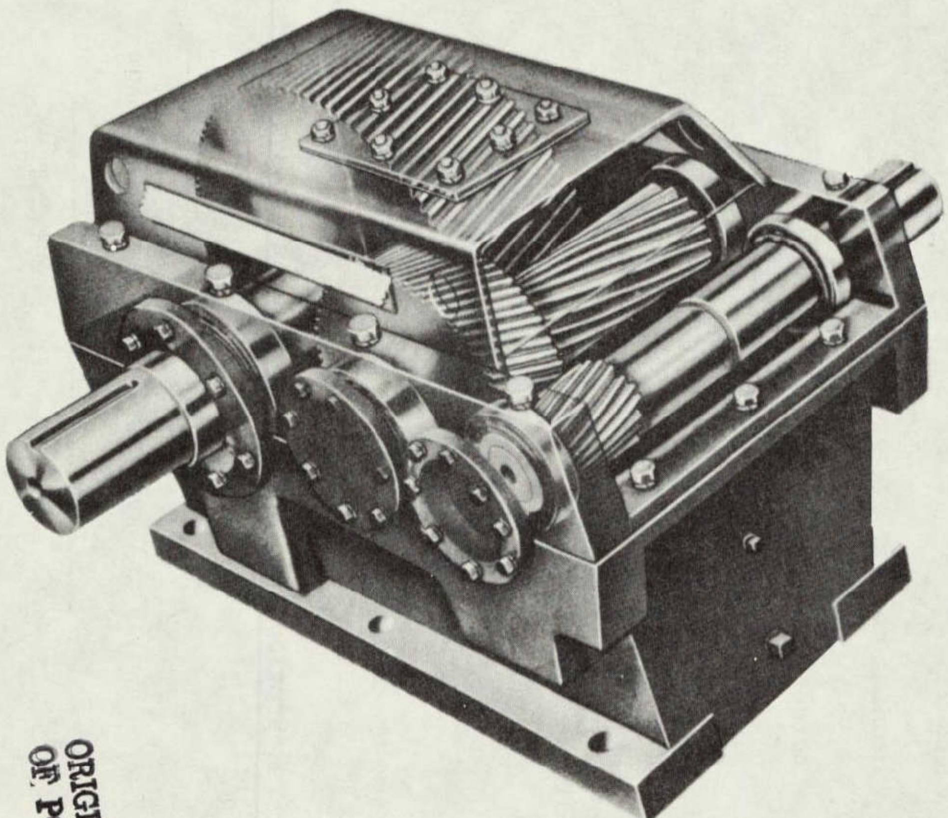
- It is not desirable to minimize the housing dimensions to the point where the thermal energy is less than the mechanical HP rating because cooling pumps, lines and radiators would be required.

- Lubricating oil heaters or fans may be required by the special environmental requirements.

- A redesigned case of the gearbox shown in Figure 5-43 would be used in the WTG.

5.3.2.2 Main Shaft Selection

Main shafts were designed for 500 and 1500 kW wind turbine generators (WTG) using a modification of the Westinghouse Code for hollow shafting:



500 KW SYSTEM
– 24,200 LBS
– 252 FT^3

1500 KW SYSTEM
– 46,000 LBS
– 338 FT^3

Figure 5-43. Gear Box

$$\sigma_{max} = (16/\pi d_o^3 B) \sqrt{K_M^2 (M_{av} + k_b M_r)^2 + T_{av} + k_t T_r)^2}$$

where $$B = \frac{1}{1 - \alpha^4} = \frac{1}{1 - \left(\frac{d_i}{d_o}\right)^4}$$

$$K_M = M_{av} + (\sigma_{YP}/\sigma_e)\ k_b M_r/(M_{av} + k_t M_r)$$

$$= 1.5\text{-}2 \text{ for suddenly applied, minor shocks (used 2)}$$

$$K_T = \underline{M_{av} + (\sigma_{YP}/\sigma_e)\ k_t M_r/(M_{av} + k_t M_r)}$$

$$= 1\text{-}2 \text{ (used 2)}$$

M_{av} (M_r) = bending moment, average (range)

T_{av} (T_r) = torque, average (range)

Two additional factors were derived to account for high thrust and shear loads, based upon the Hincky-von Mises effective stress equation. However, for the prescribed loads the required increase in diameter was negligible, and the factors are omitted from further consideration.

The following stress concentration factors were used:

	D/d	r/d	
k_b = 2 for	1.02	0.015	
$k_t \approx 1.5$ for	1.02	0.015	
k_{ta} = 2.85 for	1	0.015	(keyway in a solid shaft)

With all factors as previously mentioned, the basic shaft design equation is (changing T to T_q):

$$\sigma_{max} = (16/\pi d_o^3)\ 12B \sqrt{2^2 (M_{av} + 2M_r)^2 + 2^2 (T_{qav} + 1.5\ T_{qr})^2}$$

$$= (384/\pi d_o^3)\ B \sqrt{(M_{av} + 2M_r)^2 + (T_{qav} + 1.5\ T_{qr})^2}$$

$$\frac{d_o^3}{B} = \frac{384\ A}{75\pi} = \frac{5.12\ A}{\pi}$$

$$B = \frac{\pi d_o^3}{5.12\ A} = \frac{1}{1 - \left(\frac{d_i}{d_o}\right)^4}$$

$$d_i = d_o \sqrt[4]{1 - \frac{5.12\ A}{\pi d_o^3}}$$

where $A = \sqrt{(M_{av} + 2M_r)^2 + (T_{qav} + 1.5\ T_{qr})^2}$

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σ_{max} = 75 ksi allowable fatigue strength.

The assumed shaft material was 4340 steel, tempered and finished to a condition allowing a fatigue strength of 75 ksi. This fatigue strength is allowable for 4340 hardened to Rockwell C31 with a smooth polished surface in air, a smooth polished and oiled surface, a nitrided surface, or possibly a ground surface.

The selected material and strength are significantly better than for normal commercial shafting. A tradeoff study showed that it would be cost and weight effective to use a high strength shaft material because the bearing cost and the shaft weight increase rapidly with shaft diameter. Low strength shaft material could also make it very difficult to even obtain the required large diameter bearings and shaft material.

The shaft stock was assumed to be 4340 steel of 15.25 in. OD (solid) at approximately \$0.75/lb or \$465.22/ft. The finished shaft was assumed to cost \$2/lb (\$1240.60/ft.)

The raw stock OD and cost were obtained from Joseph T. Ryerson & Son, Inc. It is listed as a stock item as "4340 hot rolled alloy round; medium carbon-annealed; 14-16, 18-20, 20 or 20-22 ft. lengths; special straightened". Its guaranteed properties are:

	UTS	YS	2" Elong.	RA	Rockwell Hard.	Mach. Rating
Hot rolled-annealed	101 ksi	69 ksi	21%	45%	C15	45
Oil Q 1550°F-tempered 1000°F	182 ksi	162 ksi	15%	40%	C39	--

The shaft would obviously be cheaper if it is as short as possible. However, a long distance between the bearings decreases the radial loads on the #2 bearing.

The long shaft also helps appreciably in locating the gearbox forward to achieve nacelle moment balance about the yaw bearing centerline without (or with minimum) non-working counterweights. The extra length should cost relatively little more than the raw material cost because it requires minimal machining.

The identification and location of all of the MPT Subsystem components and the major loads are shown by Figure 5-42. The source and magnitude of the loads which the main shaft and bearings must react are shown by Table 5-14. The remaining components transmit only torques loads.

Thrust(T)

Continuous magnitudes T_{av} = 19 klb (500 kW) and 44.6 klb (1500 kW) are multiplied by a dynamic load magnification factor of 1.35.

Intermittent loads, due to operation at V rated and the wind velocity instantaneously doubling, are 32 klb (500 kW WTG) and 74.6 klb (1500 kW WTG). Limit loads for a wind velocity of 120 mph with the blades feathered and parked parallel to the ground are 6.7 klb for both power levels. This thrust must be reacted by the bolts and the hub/shaft rigid coupling, the shaft and the #2 bearing. The hub/shaft coupling is planned to be welded integrally onto the main shaft to avoid another set of bolts which would probably have to be blind-tapped into the shaft walls, either perpendicular or parallel to the shaft centerline. Since the hollow main shaft walls will be relatively thin because of the central hole, which is required both to save aloft weight and allow communication between the rotor pitch-change mechanism and its controls forward of the gearbox, the second set of bolts is undesirable. Based on this approach the #1 bearing will be assembled from the transmission end of the shaft.

An alternative approach is to design the #1 bearing for radial and thrust loads, however, this places a load path out over the tower - a cantilevered situation. More efficient use of structure is achieved in the proposed approach. In this instance, the shaft must absorb the thrust from the rotor to the #2 radial/thrust bearing, but the size and weight penalty is small. A relatively few mils of stock, to increase the shaft OD or decrease its ID, are sufficient to take the moments and torque when the shaft OD is approximately 13-14 in.

The present #2 bearing selection (SKF two-row spherical roller bearing with shaft adapter and pillow-block) does not require a thrust shoulder on the gearbox side of the #2 bearing, according to the manufacturer's listed ratings. If a thrust shoulder should become necessary, it is still theoretically possible to provide one in the form of a thrust nut or snap ring. This is indicated in Figure 5-44 by the threaded nut which penetrates the space blocked out for the #2 bearing. The nut would have to be locked to prevent disengagement during operation and sealed to prevent grease leakage. If the nut cannot be completely inside the pillow-block, the OD would have to be a pure cylinder to accept the rotating labyrinth seal member of the pillow-block. Therefore, it is highly desirable to select a #2 bearing which does not require a thrust shoulder. It would be impractical to provide a thrust shoulder for the #1 bearing with an integral hub/shaft coupling unless the bearing and the thrust nut were large enough to be installed from the gearbox end of the shaft, over the #2 bearing mating surface and the shaft OD.

Torque (T_q) loads are high because of the high power requirements at the low rotor RPM's but, their contributions to the shaft diameter requirements are only a fraction of the moment requirements, as later calculations will show. Their range is assumed to be the magnitude of their nominal rated levels, and they will be zero in the 120 mph parked condition.

The control system is designed to minimize transient torques, however, conservative design factors have also been included in the MPT. Rotor moments have the following sources:

M_Z1 = moment due to thrust, tower shadow and vertical wind shear

M_Z2 = moment due to vertical load

M_Z3 = gyroscopic moment due to yawing at 1/3 RPM

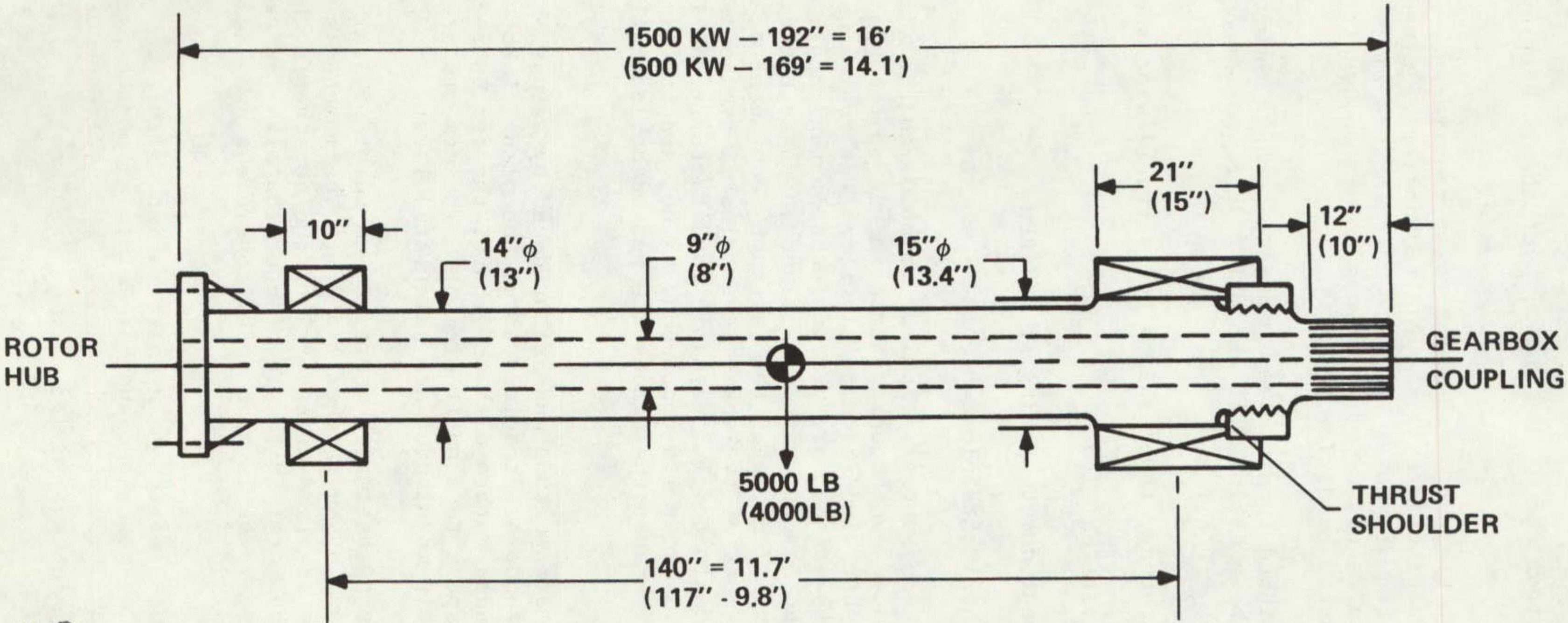
M_Y1 = moment due to horizontal load

M_Z1 is the greatest moment and the principal contributor to the shaft diameter requirements. It is produced by the aerodynamic forces on the blade which are greater because of the higher wind velocity aloft than at the ground. This moment is a maximum when the blades are vertical and approximately zero when the blades are horizontal. The moment pattern is fixed in space but varies over the full range of M_Z1 at 2/rev. Thus, even the "average" component of the moment is time variant and an element of the shaft sees M_Z1 as a cyclic load; therefore, M_Z1 is multiplied by the estimated 1.35 dynamic load factor. The intermittent magnitude of M_Z1 can be 1.5 times its continuous value, and the limit value in the 120 mph wind is zero, because the blades are parked horizontally to eliminate the wind shear moment.

M_Z2 and M_Y1 are of the same magnitude and are produced by aerodynamic forces on the rotor blades. These loads are resolved about the plane of rotation and main shaft axis. Since the blades rotate with the shaft, the force produces a moment which is constant with respect to a shaft element. The moments are caused by forces F_Y1 and F_Z1 and their magnitudes are calculated later.

M_Z3 has been calculated using estimated inertias lumped at several discrete locations in the nacelle and a yaw rate of 1/3 RPM. The moment is estimated without correcting for the inertias of several of the components which will not affect the shaft, since the rotor is the major contributor. Yawing occurs intermittently, but, its average value is zero under continuous and limit load conditions.

Figure 5-44 shows the results of the main shaft design calculation reflecting the load and requirements as described above. Natural frequencies were also calculated to determine the torsional and lateral bending frequencies and were previously described in Section 5.2.5.3.



MATERIAL: 4340 REF. AT $\sigma_f \sim$ 75ksi
SHOCK FACTOR OF 2 USED FOR ALL LOADS.
STRESS CONCENTRATION FACTORS USED FOR ALTERNATING COMPONENTS.
CRITICAL FREQUENCY PROBLEMS MAY REQUIRE LARGER O.D.'S.

Figure 5-44. Main Shaft Design

The main shaft is one principal item of the mechanical power transmission subsystem which must be custom designed and fabricated. There is little likelihood that a commercial item can be found which is directly useable or easily modified. Therefore, the main shaft is potentially a long-lead item, and its design must be coordinated with the rotor hub, bearings, and shaft/gearbox coupling designs.

The bearings are also long-lead items, and the shaft OD should be firmly determined as soon as possible so that the bearings can be ordered.

The shaft ID, which is required for rotor hub control and electrical wiring, must be firmed before the shaft design can be completed. The hub-to-shaft coupling interface requirements must also be firmly resolved as soon as possible.

The gearbox low speed (input) stubshaft OD must be firmed before a coupling design can be chosen, and the main shaft must be compatible with that stubshaft coupling's.

5.3.2.3 Couplings

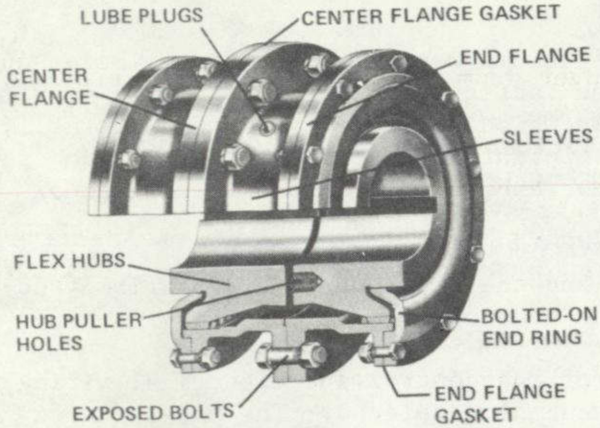
The requirements of the couplings are to transmit the required torques at the specified speeds while providing shaft misalignment capability, at minimum weight and maximum cost-effectiveness and maintainability. The main shaft coupling must also be hollow to allow passage of the rotor blade pitch controls from the forward end of the gearbox. It is desirable for the couplings to require no lubrication if weight and cost impacts are reasonable.

Couplings were selected for the main shaft-to-gearbox and gearbox-to-generator shaft. While a number of different products were surveyed, the entire spectrum of possibilities was not. Consideration should be given to having a single vendor provide the main shaft and flexible couplings. If this vendor should also be the gearbox vendor, greater compatibility would result. The selected main shaft-to-gearbox coupling is shown in Figure 5-45 and the gearbox-to-generator flexible metal disc couplings are shown in Figure 5-46. Additional data on the main shaft and generator-to-gearbox flexible couplings is given in Reference 5-8.

5.3.2.4 Hydraulic Brake

The cost of the entire brake system is insignificant with respect to the rest of the mechanical power transmission subsystem, so the preliminary design effort was not extensive. However, a system was chosen for the two wind turbine generators which differed only in disc thickness and was more than sufficient for both.

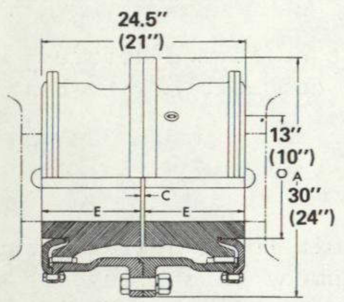
Selection of the brake system was based upon calculations of brake torque requirements, disc temperature rise, puck life and stopping rates. A schematic of the entire system is shown in Figure 5-47. The brake is a spring loaded caliper design which fails safe if the hydraulic pressure falls below 1310 psi. To prevent this from interfering with normal operation, a 460 in^3 accumulator is



CAST STEEL
FLEXIBLE-FLEXIBLE GEAR TYPE

FEATURES

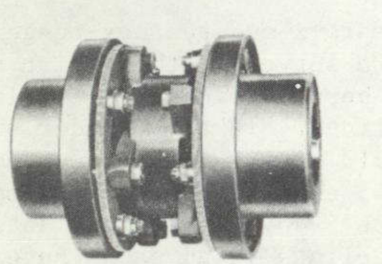
DESIGNED FOR CONTINUOUS HEAVY DUTY SERVICE
STRAIGHT-FACE TEETH WITH FULL INVOLUTE PROFILES
LARGER PITCH DIAMETERS FOR LOWER TOOTH STRESSES
WIDER SEPARATION OF GEAR SETS
OPTIMUM PERFORMANCE WITH EITHER
NO PARTIAL DISENGAGEMENT OF MATING TEETH
FAST'S EXCLUSIVE END RING DESIGN
TIGHT-FITTING BOLTS IN JIG-REAMED HOLES EQUALIZES BOLT STRESSES
REMOVABLE END RING GEAR TEETH
TYPE CMM MILL MOTOR COUPLING HAS A TYPE MM EXTENDED FLEX HUB WITH A SHORT INNER END TO ACCOMMODATE A TAPERED SHAFT WITH LOCKNUT IN EITHER A FULL-FLEX OR FLEX-RIGID ARRANGEMENT. A TYPE MM EXTENDED RIGID HUB WITH AN INNER END COUNTERBORE IS USED WHEN A RIGID HUB IS SELECTED FOR MOUNTING ON A TAPERED SHAFT WITH LOCKNUT.



	(500 KW)	1500 KW
RATED/NOM. TORQUE	1.93	1.52
ALLOWABLE ANGULARITY	±3°	±3°
MAXIMUM BORE	11.5″	14.5″

Figure 5-45. Main Shaft/Gearbox Flexible Coupling

FLEXIBLE – FLEXIBLE DISC TYPE



	(500 KW)	1500 KW
RATED/NOM. TORQUE	4.24	3.16
MAXIMUM BORE	4.5″	5.5″

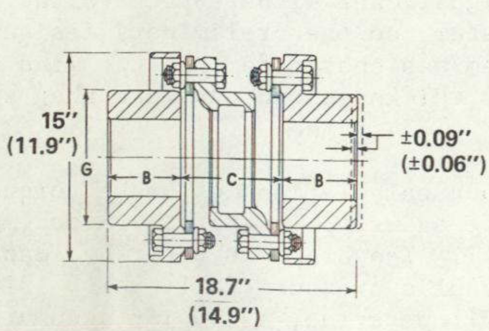


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Figure 5-46. Gearbox/Generator Flexible Coupling

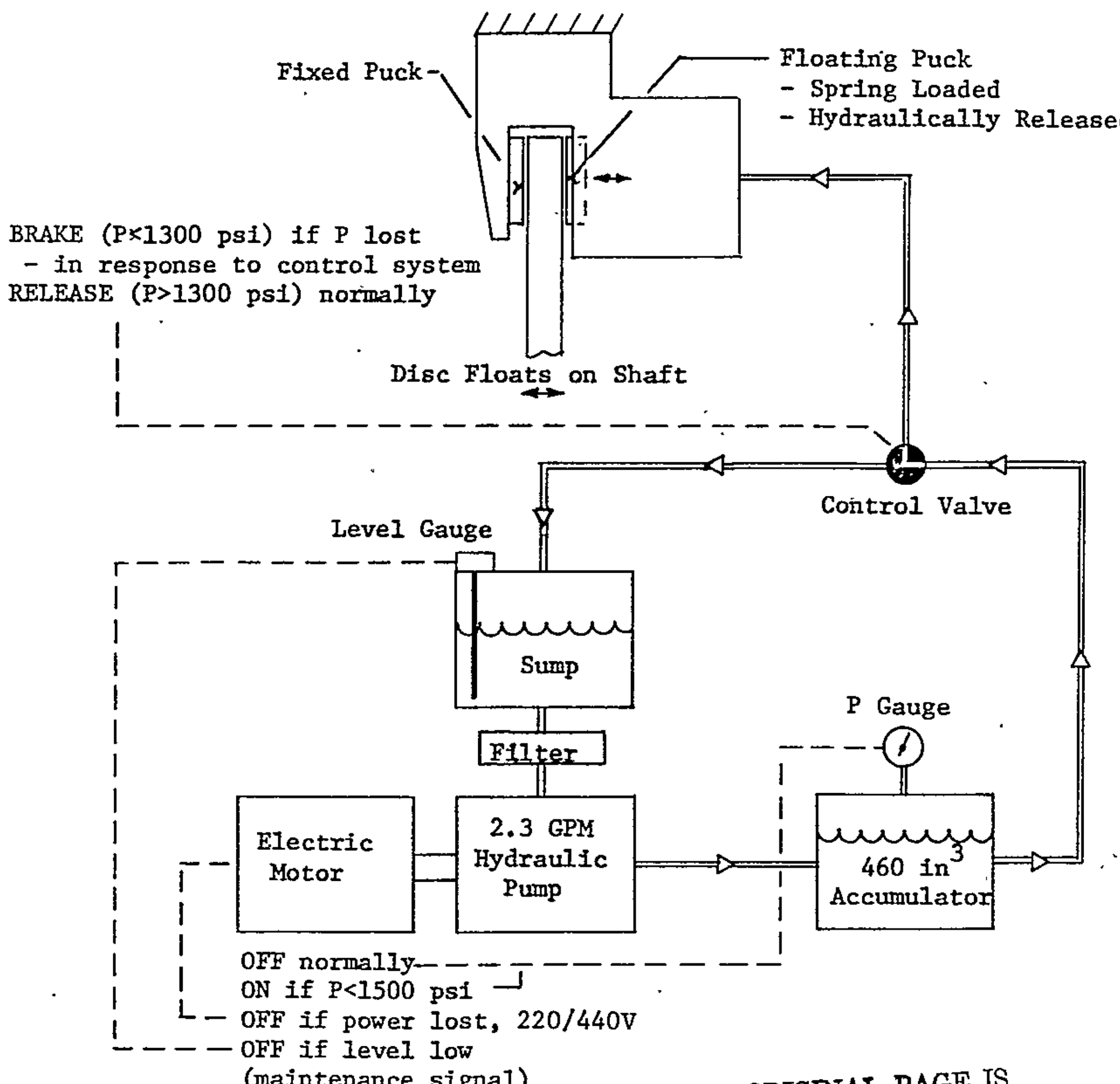
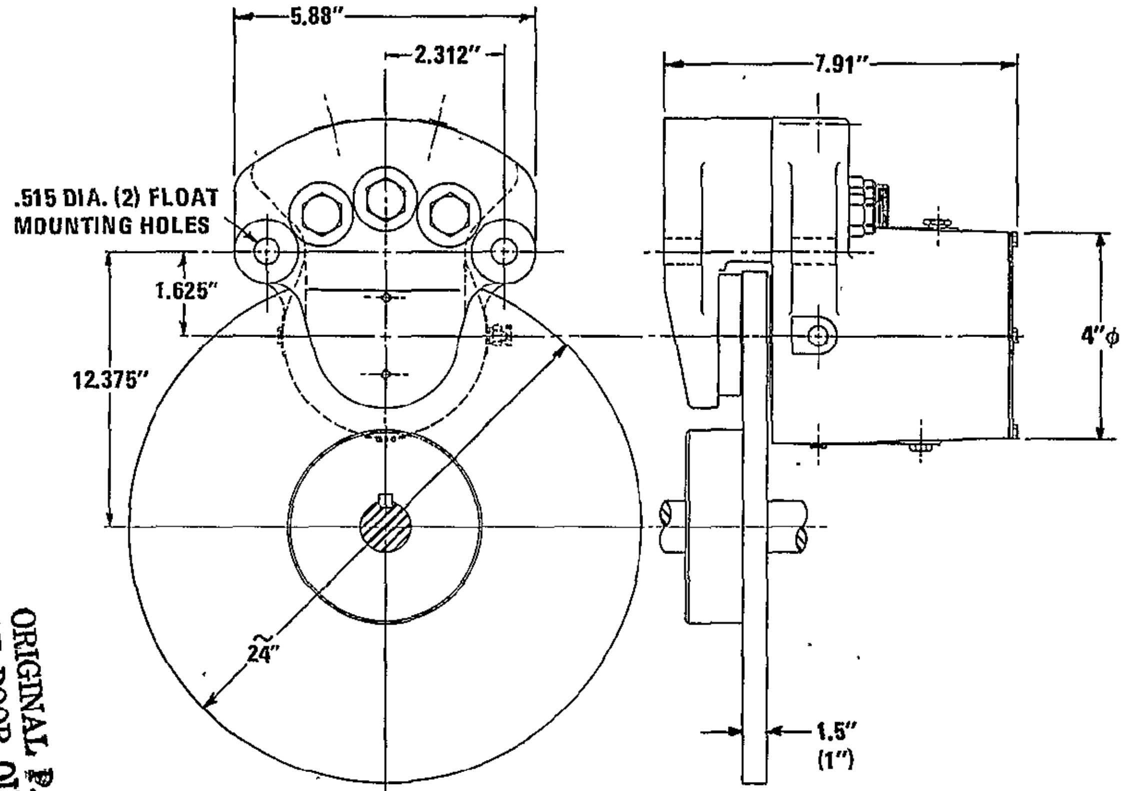
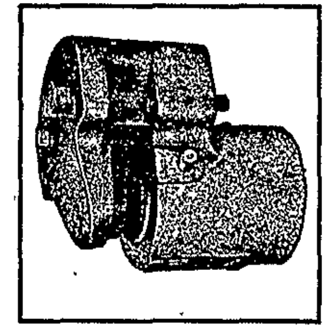


Figure 5-47. Hydraulic Brake System Design





FEATURES

2000 PSI RATING
3.24 .IN^3 WEARABLE MATERIALS
1.5 HP ELECTRIC MOTOR
2.3 GPM HYDRAULIC PUMP
46 .IN^3 ACCUMULATOR
ON/OFF ELECTRICAL CONTROL VALVES

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	500 KW	1500 KW	
WK^2 AT 1800 RPM SHAFT	~ 7,060	~ 13,760 LB-FT^2	
BRAKING/NOM. TORQUE	1.89	0.63	PANIC
STOPPING TIME: 1800 – 0 RPM	10	20 SEC	PANIC
STOPPING CYCLES: 1800 – 0	750	310	PANIC
1800 – 0	960	490	ΔT_{DISC} = 75°F
100 – 0	310K	159K	ΔT_{DISC} = 75°F

Figure 5-48. Hydraulic Brake

incorporate a static rotor support structure in the nacelle machinery support called the rotor locking flange, as shown in Figure 5-49. In addition, the #1 bearing is a split radial bearing with zero thrust capacity enclosed in a split cast pillow block; additional cost investigations may change this selection to a straight radial bearing. The #2 bearing is housed in a split pillow block and has radial and thrust capacity. The flexible gear coupling provides a removable center section to allow for axial (upwind) displacement of the shaft for either bearing or shaft removal. This callout of components and configuration arrangement provides for maximum ease of maintenance involving the bearings, mainshaft and mainshaft coupling.

The placement of the parallel shaft gearbox speed increaser in an over-under configuration was the result of a desire to minimize the thrust couple of the mainshaft on the azimuth bearing and to place the high speed gear pinion in the most accessible position for maintenance of gearbox bearings and hydraulic brake system. In addition, the gearbox case would be reconfigured from a standard unit to allow separation into three sections, split at the high speed pinion shaft centerline and bull gear centerline.

Operational maintenance requirements would include visual inspection for component deformation, loose bolts and misalignment. In addition, lubrication system and fluid levels would be checked as well as brake pad and disc ware. Final design, and consultation with component vendors will result in a detailed maintenance manual and schedule.

5.3.3 ELECTRICAL SUBSYSTEM

The electrical subsystem includes the electrical power generating equipment, the auxiliary and emergency power equipment, the electrical protection equipment, and the heating, ventilating and air conditioning (HVAC) equipment. The preliminary design assumes standard components that are commercially available and that satisfy the electrical subsystem design requirements. The preliminary design also assumes minimum electrical equipment aloft with most of the equipment located in a controlled (HVAC) environment enclosure at the base of the tower.

5.3.3.1 Electrical Subsystem Specifications

Preliminary designs of the electrical subsystems for both the 500 kW and the 1500 kW WTG systems were accomplished in accordance with the utility interface requirements discussed in Section 2.1 with the system design specifications outline in Section 5.1.2, and with the system operational characteristics described in Section 5.2.

In terms of the utility interface requirements, the preliminary designs reflect the following considerations, which could be of concern to the utilities:

1. Voltage fluctuations
2. Power factor
3. Circuit relaying and line reclosing
4. Telephone interference
5. Stability

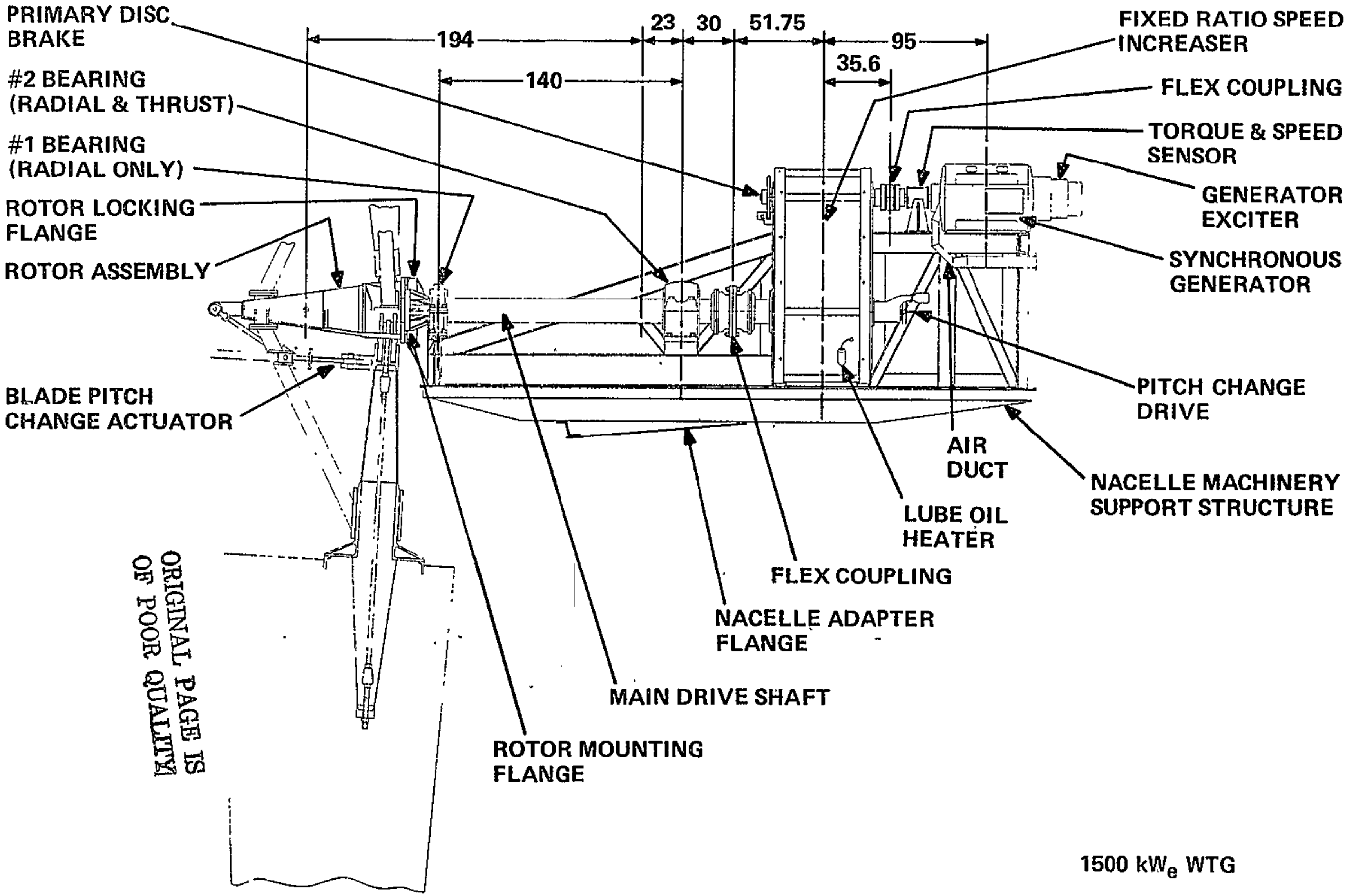

Figure 5-49. Nacelle Component Layout

The electrical subsystem designs provide features which support these utility interfacing considerations.

In terms of the system design specifications both the 500 kW and 1500 kW designs provide (1) an 1800 RPM, 3-phase synchronous generator with an exciter and voltage regulator for AC power generation and regulation, (2) transformers and an engine-alternator set for auxiliary and emergency power requirements, respectively, (3) transfer switches, contactors, relays, resistors, etc. for WTG system and network service protection against both WTG and network faults as well as potential lightning strikes, and (4) a HVAC system to protect and maintain electrical equipment operation and provide a controlled environment for the control system electronics. Also, in terms of system design requirements, it is noted and discussed further in Section 5.3.4 that adequate protection against WTG faults, network faults, and lightning is provided for the control system electronics and control sensors using isolators and an uninterrupted power supply (UPS).

In terms of the system operation characteristics, the design of the electrical system becomes closely allied with the design of the control system to provide semi-continuous operation with adequate stability. The overload capabilities of the generator during gust transients and the design of high performance excitation systems produce rather dramatic reductions in the control system requirements. Also, it is noted that the on-line stability characteristics and, therefore, the design of the power generation equipment is dependent in part on specific utility network reactance characteristics.*

5.3.3.2 Key Features

The preliminary design of the electrical subsystems provides some key features, which include:

(1) Maximum generator protection with a minimum of protection equipment by utilizing the control system electronic capabilities.

(2) An electrical subsystem, which functionally interacts with the electronic control subsystem but is completely isolated electrically from the computer.

(3) An auxiliary power system, using network power for control functions, for which the load can be automatically transferred to a computer initiated emergency power system in the event of network power interruption.

(4) A scheme for the protection of network service using time selectivity and neutral grounding of the generator.

(5) A helical loop cable wrap of 720° to allow tower turning with wind directional changes without the use of slip-rings.

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* A 4160V distribution line of 5-mile length is about 0.13 p.v. impedance on a 500 KVA base.

(6) A maximum amount of equipment mounted at the tower base for ease of maintenance.

5.3.3.3 Power Generation Equipment

The power generation equipment basically includes the brushless generator, the exciter, and the voltage regulator. A 4160V, 1800 RPM 3-phase synchronous generator, of either 500 KVA or 1500 KVA rating with 0.95PF depending on the wind power level, was selected in the preliminary design.

Additional consideration was given to preliminary design in which an induction generator of 700 Hp (500 kW) at 4160 volts, 1820 RPM, and 0.86 P.F. was selected.

5.3.3.4 Auxiliary/Emergency Power Equipment

The auxiliary power equipment is used to supply the WTG system with electrical power, which is drawn from the utility network, to drive control system components and provide other miscellaneous power requirements such as for lighting and maintenance. Auxiliary power from the network is required only when the WTG is not producing adequate power either in the form of electrical or parasitic mechanical power to drive controls such as the blade pitch actuator, the azimuth position actuator and hydraulic pumps. The auxiliary power system is basically a 10 KVA, 3-phase 4160/240V transformer drawing power from a 4160V network potential and providing a 100-amp service at 240V to the WTG system.

In the event the network power is interrupted and auxiliary power is required, an emergency power system is automatically activated to provide any necessary control system power requirements. A 7.5 KVA, 3-phase 120V engine-alternator set is assumed in the preliminary design along with a 10 KVA, 3-phase 120/240V transformer to supply the same service as the auxiliary power system. An 11.2 kW 1800 RPM gasoline engine, which is started using a wet cell battery with a battery keeper, is located in the controlled environment enclosure at the base of the tower and is adequately maintained for operational "reliability".

5.3.3.5 Protection Equipment

The objectives of the protection equipment used in the electrical subsystem are to protect the network service against WTG system faults, to protect power generation equipment against both WTG system faults and network faults, and to protect the electronic control system, including microcomputers and sensors, against both WTG system and network faults. It has been discussed in Section 3.0 that the minimum standard system of protection devices is not cost effective for WTG systems in the 500 to 3000 kW range. Also, it has been stated that utilization of the microcomputer and control system electronics can be implemented with appropriate electrical circuitry to provide a sophisticated, cost effective protection system, whereby the system automatically protects itself and the network without protective relaying hardware.

Such a protection system was assumed in the preliminary design of the electrical system and basically includes one main 4160V contactor, a series of molded-case 120V circuit breakers to protect WTG components, an automatic transfer switch, a neutral ground resistor, and an uninterrupted power supply for the control electronics.

The main contactor (or circuit breaker), as shown schematically in Figure 5-50, is utilized to connect the WTG to the network and serves to protect the network service against WTG system faults. The main contactor is capable of interrupting 350 MVA at 4160V with time selectivity, which allows other WTG 120V circuit breakers to operate first for lesser overloads. Control power is obtained for the auxiliary transformer from this circuit breaker by tapping into the utility before the air-break contactor. This auxiliary transformer is fused for 350 MVA interruption of a nominal 10 KVA load. When the auxiliary transformer is not supplying the control power, only the magnetizing current of the transformer is being supplied by the utility network. Also, the 400-AMP neutral grounding resistor is provided so that generator fault conditions will be limited and will not be propagated beyond the main contactor, the outage being confined to the WTG system.

By utilizing the microcomputer and instrumentation transformers, it is possible to provide the generator protection against both internal faults and network faults, without adding protective relaying. Logic and mathematical algorithms can be developed to protect against (1) generator thermal overload as a result of overcurrent, (2) generator phase unbalance and rotor overheating, (3) anti-motoring, (4) synchronizing, (5) loss of transformers, and (6) ground sensing.

WTG protection against network faults is also provided by a 50 MVA air-break contactor in the main contactor which is commanded (microcomputer) by an automatic transfer switch (Figure 5-51) that transfers system load from auxiliary to emergency power, and by an uninterrupted power supply for the microcomputer and control system operation.

5.3.3.6 HVAC Equipment

Most of the electrical subsystem and control subsystem equipment is located in a control building, or enclosure, at the base of the tower. This enclosure has a controlled environment which is provided by the HVAC equipment. The HVAC equipment required includes a 3 kW base board heater and a 6000 BTU/HR air conditioner.

5.3.3.7 Electrical Subsystem Cost/Weight

The cost of the electrical subsystem is estimated at $37,400 and $60,700, respectively, for the 500 kW and 1500 kW systems. Costs include the following electrical equipment:

1. Generator (Synchronous or induction)
1A. Exciter and voltage regulator (synchronous only)
1B. Capacitor Bank and Capacitor Switch (Induction only)
2. Auxiliary power transformers
3. Engine alternator set
4. Main contactor
5. Transfer switch
6. HVAC equipment
7. Grounding resistor
8. Instrumentation transformers
9. Lightning protection

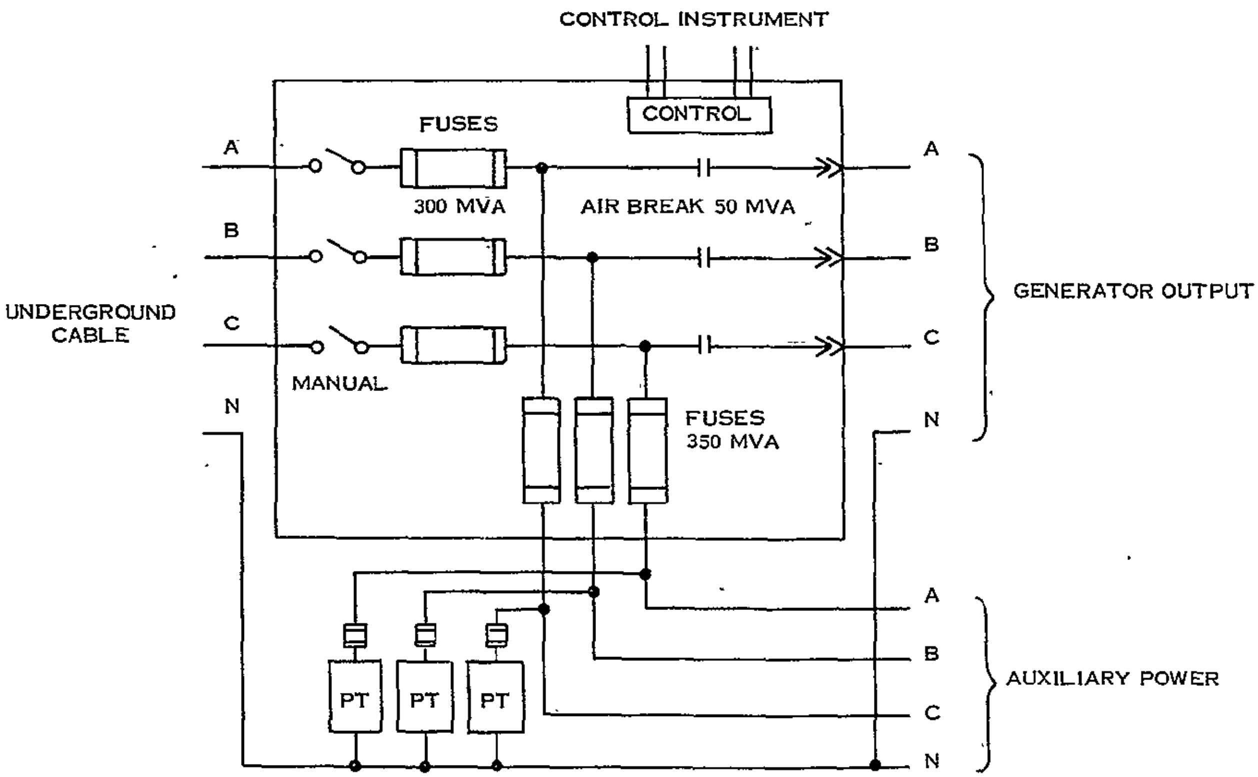


Figure 5-50. Main Contactor Design

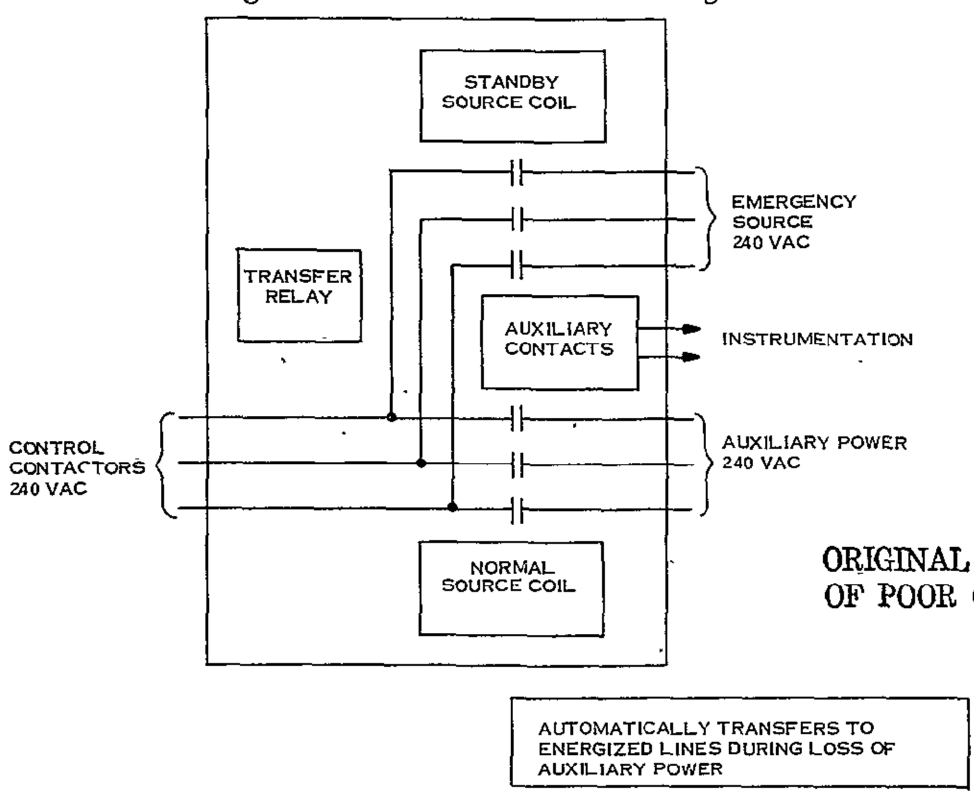


Figure 5-51. Automatic Transfer Switch

10. Small circuit breakers and relays
11. Cabling

Generator weights were determined in the preliminary design studies at 1680 kg and 4045 kg for the 500 kW and 1500 kW systems, respectively. Exciter weights (approximately 182 kg), depend on more detailed design considerations. Other aloft component weights have no significant effect on preliminary design. The weight of a 700 Hp (500 kW) induction machine is 2120 kg. The capacitor bank and capacitor switch are housed in the control enclosure.

5.3.4 CONTROL SUBSYSTEM

The control subsystem consists of all the equipment necessary to sense system operational modes and faults, to perform decision processes, to make functional commands, to actuate controls, and to communicate system data and commands. The objective of the control subsystem is to provide a fully autonomous WTG system operation and to communicate any system faults or maintenance requirements to central locations remote from the WTG installation.

5.3.4.1 Control Subsystem Specification

The preliminary design of the control subsystem assumes functional requirements to provide a blade pitch control, rotor axis azimuth orientation, brake control, generator voltage control (synchronous only), capacitor switching (induction only), component temperature control, generator off/on line technique, monitoring and metering environmental and system performance data, and aircraft warning beacon control.

The control subsystem includes all the components necessary to the above functions except for those components which are included as an integral part of major WTG elements. For example, the pitch control mechanism and motor are a part of the wind rotor; the azimuth drive and motor are a part of the pintle section of the tower; and the wind-rotor brake and transmission-oil heater are part of the transmission subsystem.

All instrumentation and electronics that are not required to be aloft are protected in a control house at the base of the WTG tower. The master control for the logic of operation is provided by a microcomputer in conjunction with data acquisition electronics. For on-off control functions, output signals from the microcomputer are buffered and amplified to drive relays and contactors which apply power to drive at full power as long as the signal is present. For proportional control, coded outputs are decoded to time function for time-ratio-control of silicon-controlled-rectifier circuitry that meter the AC power for the proportional control drives.

The control subsystem is composed of all standard, commercially available components and the hardware requirements for the preliminary design are as follows and which are itemized in Appendix 8.4:

2 MICROCOMPUTERS (8-BIT WORDS), 5-WATTS EACH

4 PROPORTIONAL CONTROLLERS (+ 1 for synchronous WTG only)
15 RELAY DRIVERS (+ 1 for induction WTG only)
20 SENSORS (+ 1 for induction WTG only)
30 EVENT DETECTORS (+ 3 for induction WTG only)
2 SIGNAL CONDITIONERS/ANALOG-TO-DIGITAL CONVERTERS
1 TELEPHONE LINE
1 MODEM
1 AUTOMATIC CALL UNIT

The control subsystem software is developed as a final design requirement for specific WTG applications.

A functional block diagram of the control subsystem that has been assumed in preliminary design is shown in Figure 5-52. The basic components and their operational function is shown to demonstrate the control subsystem operation. It is important to realize that the microcomputer, as the heart of the control subsystem, is electrically isolated from WTG generated and network supplied (auxiliary) power. The sole power source for the microcomputer comes from the uninterrupted power supply; which also provides any power requirements for isolated instrumentation sensors. Subsystem control sensors, both isolated and non-isolated, feed information to the microcomputer through appropriate signal conditioning equipment. The microcomputer absorbs the information, goes through the decision processes dictated by the software, and issues electronic commands through decoders and drivers to activate and regulate control functions.

The two-way communications function is also shown along with an optional on-site portable teletypewriter for interrogating and issuing commands to the microcomputer during maintenance operations.

5.3.4.2 Key Features

The key features of the control subsystem, some of which have already been discussed, are as follows:

(1) Utilization of optical coupling, high impedances and other techniques common to data acquisition technology for isolation of various inputs and outputs of the microcomputer. Isolation is required to protect the electronics from damage which might otherwise be caused by large power switching transients and lightning strikes.

(2) Utilization of redundant microcomputers for providing WTG protection against loss of critical microcomputer functions and for dividing non-critical on-line and off-line functions between microcomputers. In the unlikely event that one microcomputer fails to operate properly, all critical functions are controlled and only half of the non-critical functions are temporarily lost.

(3) Utilization of a random-access-memory (RAM) in the microcomputer to provide data time histories (last 10 points vs. time) so that the microcomputer can deliver recorded data of critical functions for analyses of malfunctioning.

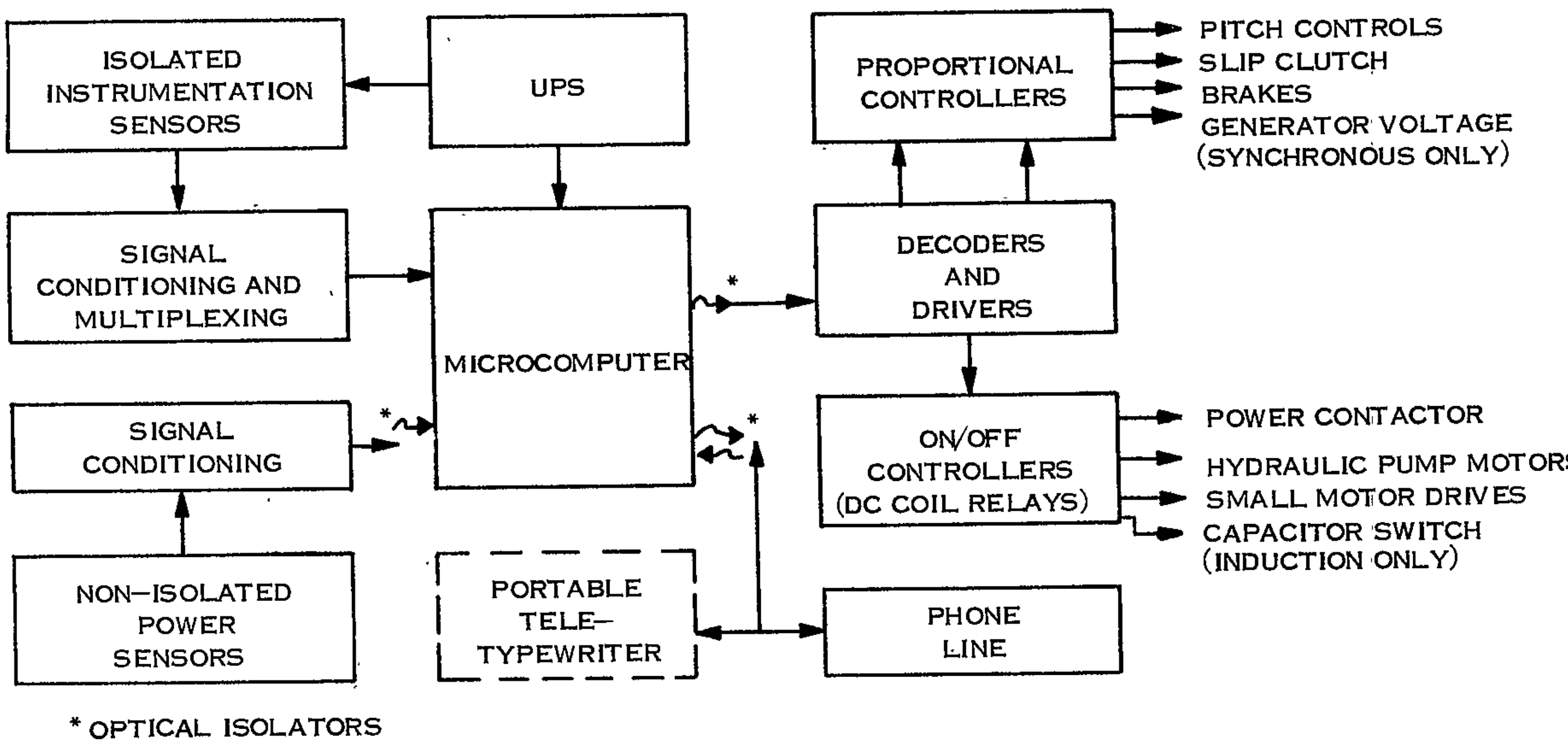


Figure 5-52. Control Subsystem Block Diagram

(4) Utilization of a teletypewriter for interrogation and command signals to the microcomputer. This includes a central teletypewriter used at a utility substation and shared as a dispatcher unit for a complex of WTG's, and an optional portable teletypewriter for maintenance.

(5) Utilization of an uninterrupted power supply to protect the control subsystem electronics. An uninterrupted power supply insures that power sufficient to supply the data acquisition electronics and microcomputer is available even when the auxiliary power from the utility network is temporarily interrupted. In this circumstance, the microcomputer logic will initiate the operation of a standby engine generator set as a temporary measure, so that WTG control is maintained and the condition can be transmitted to the remote headquarters.

5.3.4.3 Microcomputer Description

Two (2) microcomputers are assumed for redundancy and functional load sharing in the preliminary design of the control subsystem. The following description applies to each of the two 8-BIT-WORD (16K-BYTES CORE) microcomputers.

The microcomputer includes:

- A central processor unit, which is capable of more than 100 instructions.
- Four (4) random-access-memories (RAMS) with 4096 BITS each (2K-BYTES)
- Three (3) programmable-read-only-memories (ROMS) with 2048 WORD x 8 BITS - (6K-BYTES).
- A clock driver and level shifter, which is composed of 5 drivers; four low voltage and one high voltage.
- Eleven (11) input/output ports, each with capacity of 8 BITS, to handle intermittent data samples.
- Five (5) input/output priority ports to handle continuous data signals.
- A universal synchronous/asynchronous receiver/transmitter for communications functions.
- A 5-WATT power supply to provide 12 V, 5 V, and -5V driving potentials, each with ± 5% limit on level fluctuations.
- High precision clock/speed reference (for induction only).

5.3.4.4 Actuators

The actuators used in the preliminary design of the control subsystem are categorized into proportional controllers, on/off control contactors, and on/off control relays. A total of sixteen (16) actuators are identified and listed as follows:

- Proportional Controllers (5)
  - Pitch Control (2)
  - Azimuth Slip Clutch
  - Brake
  - Generator Voltage (Synchronous only)

- On/Off Control Contactors (8)
  - Hydraulic Pumps - Pitch, Slip Clutch, Brake
  - Transmission Heaters
  - Azimuth Drive
  - Generator Excitation (Synchronous only)
  - Air Breaker
  - Capacitor Switch (Induction only)
  - Tower Lighting

- On/Off Control Relays (4)
  - Generator Voltage Adjust (Synchronous only)
  - Aircraft Beacons
  - Emergency Power Start
  - Air Louvers

In order to protect the microcomputer from control system actuators, feed-back ripple, optically coupled drivers are used as shown in Figure 5-53.

5.3.4.5 Sensors

The sensors selected in the preliminary design of the control subsystem are categorized as either isolated or non-isolated sensors, as shown in Figures 5-54 and 5-55 with sensor type descriptions and appropriate sampling rates. Thirty isolated sensors and twenty non-isolated sensors are listed for a total of fifty-two (52) sensors.

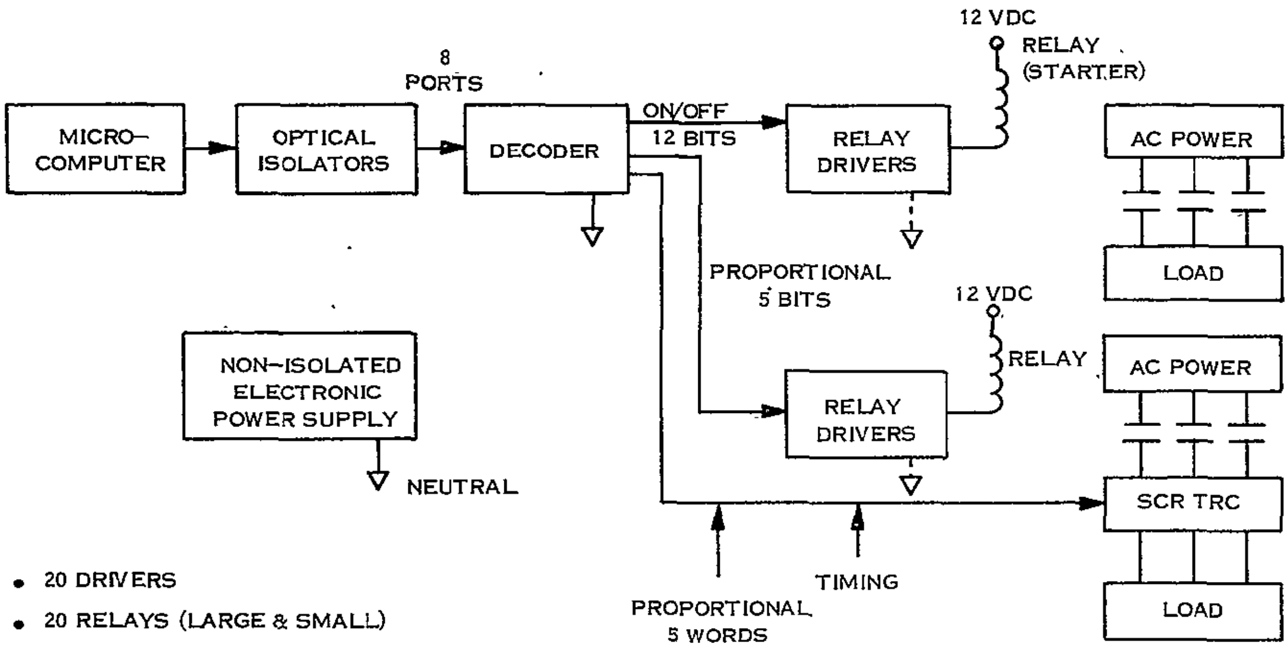
The non-isolated sensors, which are the power sensors, provide signals that must go through signal conditioning, and optical isolation before being processed by the microcomputer. This signal processing for the optically coupled power sensors is shown schematically in Figure 5-56.

Signal conditioning for the isolated instrumentation sensors, shown in Figure 5-57, shows intermittent and event signals being processed through a 32 channel multiplexer and continuous signals being processed by analog-to-digital modules prior to being inputted to the microcomputer.

5.3.4.6 Data and Command Communications

The preliminary design of the control subsystem provides for complete communication capability, as well as a manual-override capability of the WTG system between the microcomputer and the utility dispatcher. This feature closes the link between a fully automatic WTG system and the utility control. A functional flow diagram of the Data and Command communication system is shown in Figure 5-58.

The computed data and status data obtained from the microcomputer is suitable for transmission over a leased phone line. The computer also constructs a coded signal as an input to an automatic call unit which will automatically dial up a predetermined telephone number when conditions warrant outside attention to the WTG. Conversely, data can be obtained from the WTG by dialing the modem at the WTG site and upon receiving proper contact can acoustically couple the calling phone set to the teletypewriter. Once the hook-up is completed, an



- 20 DRIVERS
- 20 RELAYS (LARGE & SMALL)
- 1 DECODER
- 1 SET OPTICAL ISOLATORS
- 1 ELECTRONIC POWER SUPPLY
- 5 PROPORTIONAL CONTROLLERS (SCR/TRC)

Figure 5-53. Optically Coupled Actuator Drivers

NAME	TYPE	SAMPLE RATE
WIND DIRECTION	POTENTIOMETER	I SPS +
WIND SPEED	LIGHT BEAM CHOPPER	I SPS +
GENERATOR RPM	MAGNETIC PICKUP	CONTINUOUS *
PITCH ANGLE	LINEAR VOLTAGE DIFFERENTIAL TRANSFORMER	CONTINUOUS *
GENERATOR TORQUE	STRAIN GAUGE	CONTINUOUS *
ROTOR SHAFT VIBRATION	ACCELEROMETER	CONTINUOUS *
AZIMUTH ANGLE	POTENTIOMETER	I SPS +
#I BLADE POSITION	LIGHT BEAM	4 SPS +
TEMPERATURES (II)	RESISTANCE THERMAL DETECTORS	I SPS +
EVENTS (II)	SWITCHES	I SPS +

* PARALLEL INPUTS TO MICROCOMPUTER THROUGH PRIORITY PORTS

\+ SAMPLE AND HOLD MULTIPLEXED

Figure 5-54. Isolated Sensors

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NAME	TYPE	SAMPLE RATE
NETWORK VOLTAGES (3)*	POTENTIAL TRANSFORMERS (FUSED)	CONTINUOUS
GENERATOR VOLTAGES (3)	POTENTIAL TRANSFORMERS (FUSED)	CONTINUOUS
CAPACITOR NEUTRAL VOLTAGE†	POTENTIAL TRANSFORMER (FUSED)	CONTINUOUS
GENERATOR CURRENTS (3)	CURRENT TRANSFORMER	CONTINUOUS
NEUTRAL CURRENT	CURRENT TRANSFORMER	CONTINUOUS
EVENTS (6)*	PEAK AND ZERO DETECTORS	CONTINUOUS
EVENTS (6)	BREAKERS ON/OFF	1 SPS
EVENTS (6)	POTENTIAL TRANSFORMER FUSE	1 SPS
EVENTS (3)†	CAPACITOR BANK FUSES	1 SPS

* RETAINED FOR INDUCTION MACHINE FOR LINE PLACING TECHNIQUE OF MINIMUM VOLTAGE DIP

† INDUCTION MACHINE ONLY

Figure 5-55. Non-Isolated Sensors

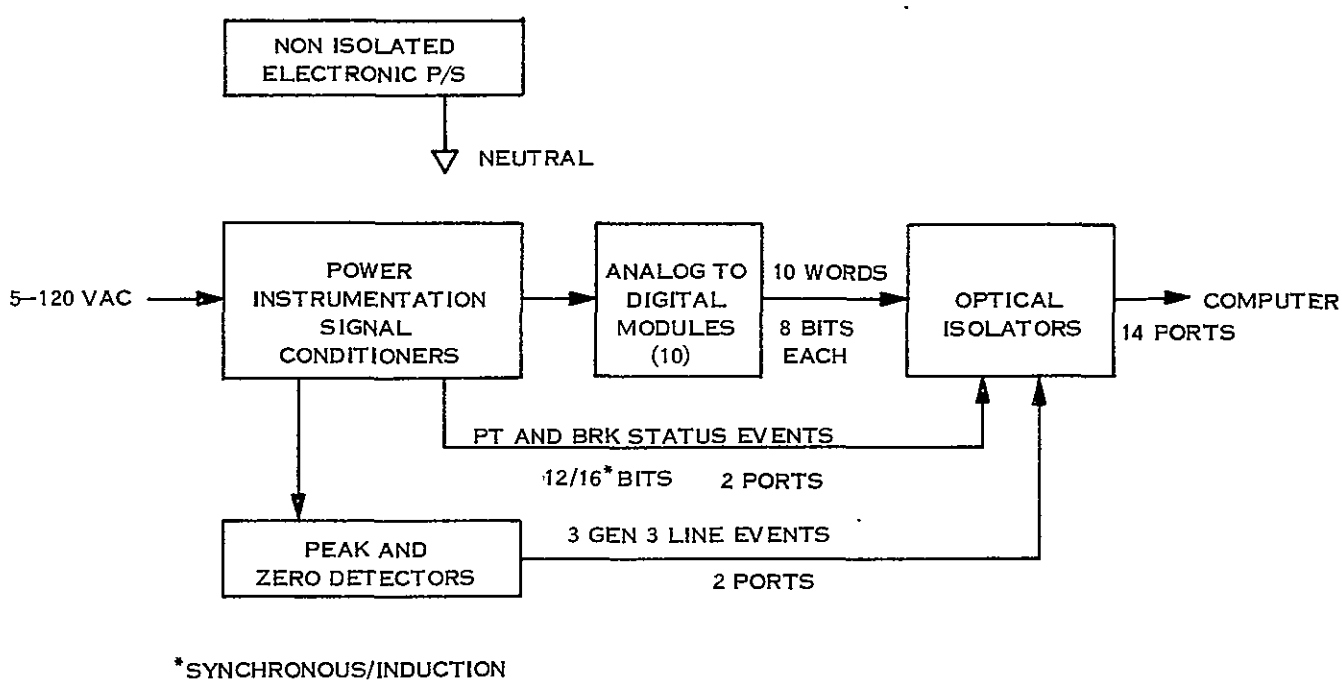

Figure 5-56. Optically Coupled Power Sensors

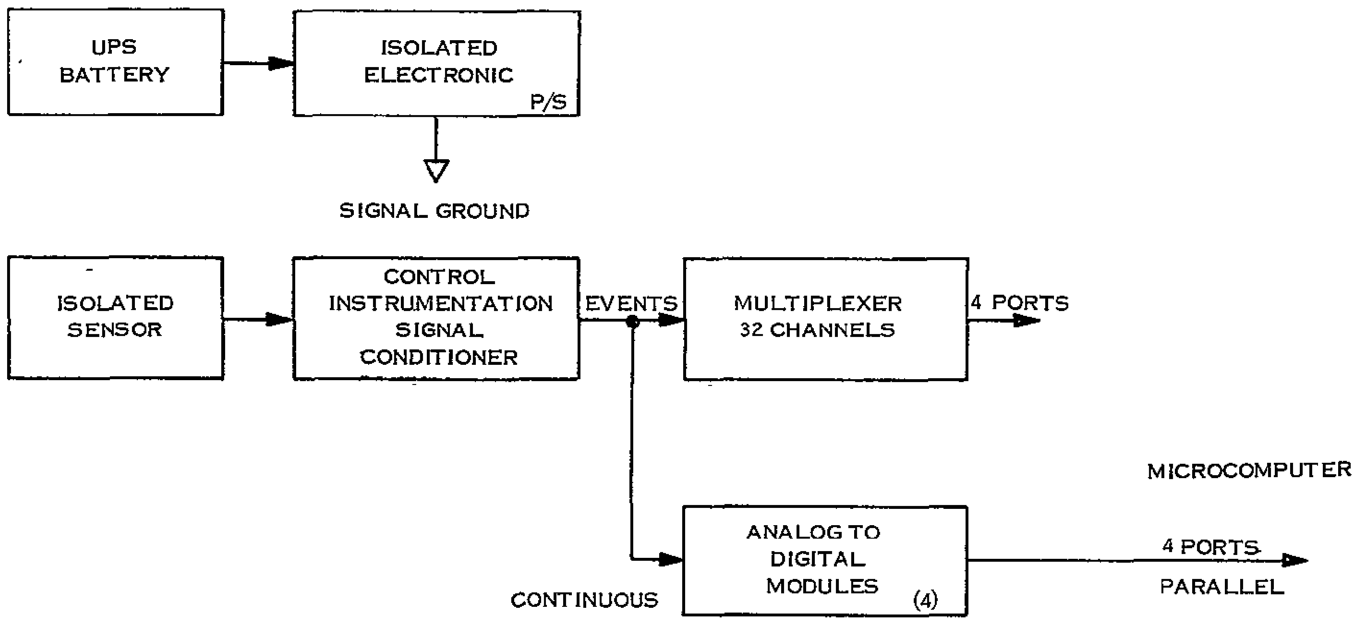

Figure 5-57. Isolated Instrumentation Sensors

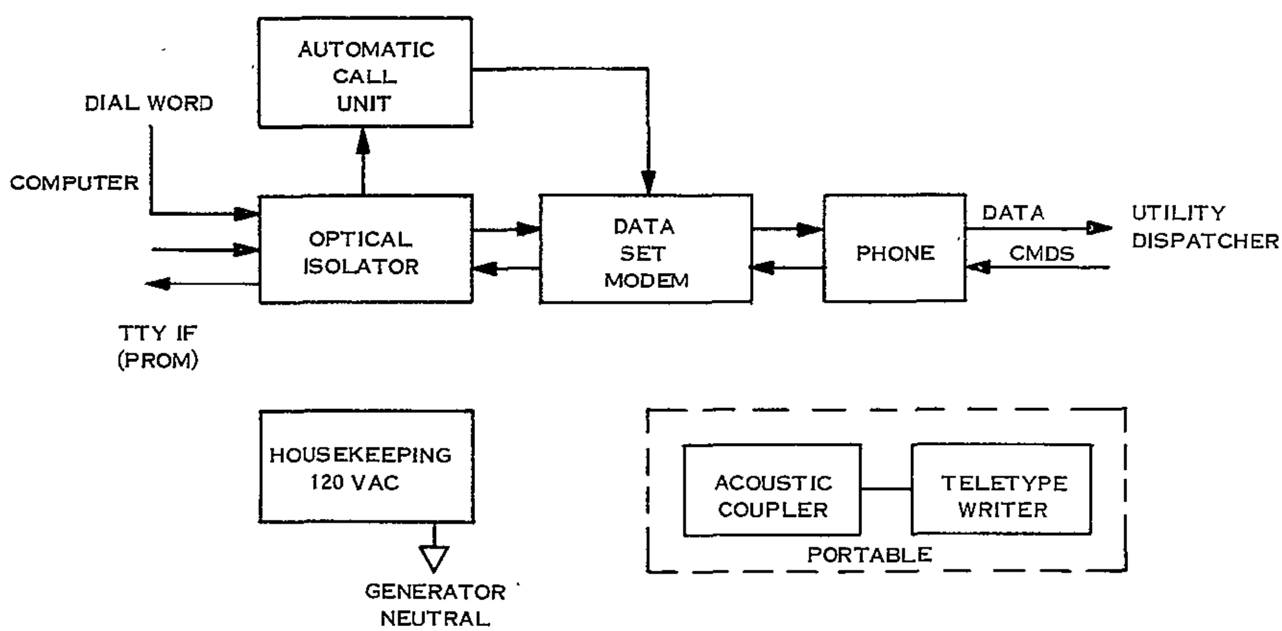

Figure 5-58. Data and Command Communication System

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agent may, from his remote headquarters site, command and interrogate the WTG to determine the status of the WTG and the nature of maintenance, if any, required. A maintenance crew arriving at the WTG site may acoustically couple a portable teletypewriter to the microcomputer and in this manner exercise the WTG and determine additional information on the condition of the WTG.

Of course, it is also possible to connect the WTG microcomputer via phone line, to a larger general purpose computer such that the output of several WTG's can be integrated into the utility network dispatcher's decisions relative to the most efficient manner of regulating supply to demand.

5.3.4.7 Control Subsystem Software

As a part of the preliminary design of the control subsystem, logic flow diagrams of major control functions were generated. A description of the Executive software logic is provided in Figure 5-59. The Executive program logic diagram is presented to aid in understanding the control subsystem and to serve as a model for the determination of memory (core) size and microcomputer architecture for estimating hardware costs.

5.3.4.8 Control Subsystem Costs

The control subsystem costs obtained during the preliminary design are expressed for components, as listed, and for the controls and sensors, as listed. These subsystem costs are applicable to either the 500 kW (synchronous or induction), or the 1500 kW WTG systems, and the total cost of $13,600 for the control subsystem represents a very cost effective method for providing an autonomous, self-protecting, and efficiently operating WTG system, which is appropriately integrated into a utility power network.

CONTROL SUBSYSTEM COMPONENTS

- Microcomputers (2)
- Proportional Controllers
- Relay Drivers
- Decoder
- Optical Isolators
- Analog-to-Digital Modules (14)
- Non-Isolated Power Supply
- Power Instrumentation Signal Conditioner
- Peak and Zero Detectors
- Analog-to-Digital Converter
- Multiplexer
- Control Instrumentation Signal Conditioner
- UPS
- Modem, Automatic Call Unit, Phone Line

CONTROLS - SENSORS

- Wind Direction Transmitter - Potentiometer
- Wind Speed Transmitter - Light Beam Chopper
- Azimuth Angle - Potentiometer
- Pitch - Linear Voltage Differential Transducer

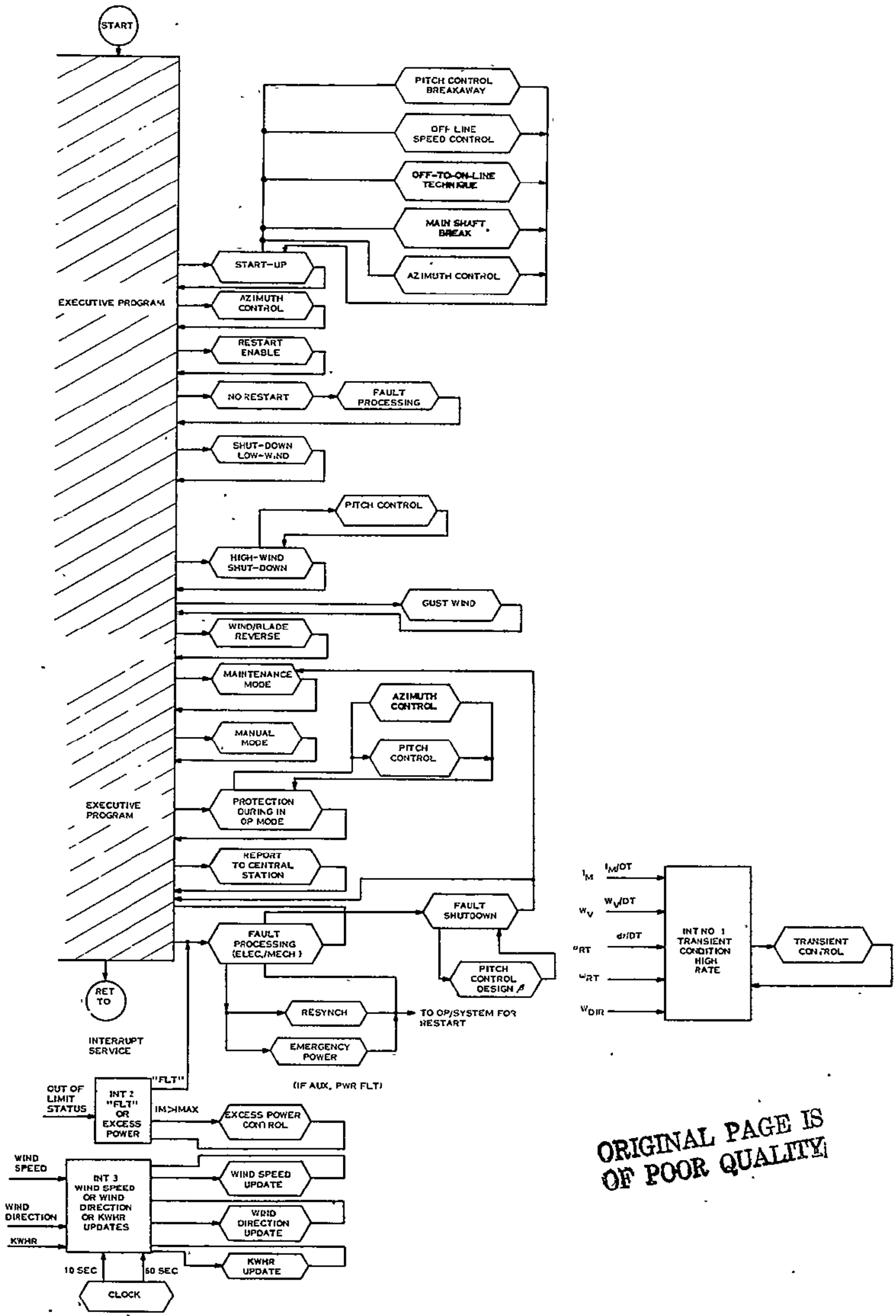

Figure 5-59. Executive Program Logic

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- Rotor Shaft Vibration - Low Frequency Accelerometer
- #1 Blade Position - Light Beam Chopper
- Transmission Oil Level - Switch
- Pitch - Hydraulics Pressure - Switch
- Brake - Hydraulics Pressure - Switch
- Slip - Clutch - Hydraulics Pressure - Switch
- Generator RPM - Magnetic Pick Up.
- Generator Shaft Torque - Strain Gauge

5.3.5 TOWER SUBSYSTEM

This section describes the approach used in the tower designs for the 500 and 1500 kW WTG Systems. Both concrete and truss towers are presented. In addition a description of the nacelle/pintle structure is also given.

5.3.5.1 Tower Loadings

The tower loadings are those listed in Table 5-4, System Load Schedule, Section 5.1.3, and follow the convention established there. Due to the wind shear and tower shadow, tower loads are cyclical on a continuous basis. In addition, changes in azimuth position impose further cyclic loadings. A member that experiences cyclic loads in tension one day may be cycling in compression the next day. Both the reinforced concrete and the truss towers are sized according to the continuous and intermittent load criteria with yields being approached in the 120 mph storm condition. A basic philospshy is to provide adequate capability at the generally low probability storm condition. It has been assumed that the control system will enable avoidance of significant wind inflow angles (i.e. 15°) in high wind conditions such that loadings due to 15° inflow are minimized.

An estimate of the tower shadowing effect has also been included in the load definition. A preliminary analysis was carried out for a typical truss structure to determine the magnitude and duration of blade unloading when encountering the tower wake. The results were utilized as a loading input for a concrete structure. In general, tower shadowing exerts a significant impact on tower designs, especially from a dynamic viewpoint, because it is the shape of the towers which determine the nature of the dynamic forcing functions acting on the towers and the rotor blades while the shapes and structural characteristics in part, are a consequence of the forcing functions. The present tower and blade designs are viewed as a starting point from which to evolve the most cost effective tower and overall WTG final design. Operational system data would then be the final input required for an optimum design definition and large scale utilization. Design iterations and changes would evolve from continued dynamic tuning, load definition, and cost effectiveness studies.

5.3.5.2 Tower Structural Design

The four towers as presently defined have several points of commonality. All of the structures are approximately 140 feet tall with the uppermost portion containing the pintle structure, which houses and meets the particular requirements of the azimuth bearing and drive assembly. The four preliminary tower designs fall into two categories; two reinforced concrete and two steel truss towers at the 500 kW and 1500 kW WTG rated power levels. Schematic drawings of each of these towers are shown in Figure 5-60; specific details and discussion for the particular approaches follow. Preliminary design drawings of the four towers have been delivered to the NASA Wind Power Office at NASA-Lewis.

5.3.5.3 Reinforced Concrete Towers

In the course of the preliminary design effort, General Electric developed analytical tools to provide the physical definition of reinforced concrete towers. On the basis of the external tower loadings, and internal loadings, tower sections are sized on a predetermined set of stress criteria. As presently organized the resultant program, TOWER, does not calculated stress levels based on input loads and assumed section sizes. Since reinforced concrete is a non-homogenous material, the internal mechanics of the program transforms the reinforcing steel rod into equivalent concrete with stresses then calculated according to conventional techniques and compared to input allowable values. The allowable stress values are chosen according to the type of concrete or steel that would be utilized (i.e., high or low strength) and according to the nature of the fatigue characteristics of the tower materials. In the preliminary design a total of thirty nine runs were made, for a variety of both wall thicknesses and material strengths for the three loading conditions, in arriving at the two concrete tower designs.

The two towers each presently have variable O.D's and I.D's with thin walls, and circular structures. The basic motivation behind a thin wall approach is to economize on the materials used and also to maintain better dimensional control during the construction process. The preliminary designs assume that each section may have a different O.D. and I.D. with transitions at the joints. This is a production approach where the tooling costs (forms) are small in relation to the total costs of many tower sections. For a small number of towers the most economical approach would probably be a constant O.D. and I.D. tower where the reduced form costs are more than the additional excess materials costs. It should also be noted that the use of concrete has the potential to be used in various geometries which may be more aesthetically pleasing. Concrete is a very flexible material which can be made architecturally pleasing not only through textured finishing but by contouring and shaping such that both close-in detail and long-range form are appealing. The present approach is to produce the most cost effective design not only from a tower load and aesthetics point of view, but from an overall WTG acceptance point of view.

Ten sections were chosen, because the limit of the truckability is being approached for the larger base sections, characteristic of 500 kW and 1500 kW WTG System towers. Both geometric and weight constraints would generally determine the size of sections amenable to truck transportation. Precasting was chosen over slip forming (continuous casting) strictly from preliminary cost considerations. The sections would be precast either at a central location or possibly on-site, depending on factors such as site location, site accessibility, local materials availability, and local labor conditions. The designs as now defined utilize Grade 40 deformed reinforcing rods and a 28-day compressive strength of 2500 psi for the 500 kW concrete and 3000 psi for the 1500 kW concrete. The trend of the analysis in balancing the load sharing between the concrete and steel suggests utilizing both higher strength steels and concrete aggregate. Design refinement probably will show the higher strength materials as the more cost effective approach. In the analysis to date it has also been assumed that the concrete is incapable of supporting a tensile load.

By "balancing" the design of a reinforced concrete structure, the most economy can be effected. Basically, that process involves working the concrete and steel to stress levels in tension and compression such that one material is not over-stressed in relation to its yield capability while the other is not. The present designs have maximum working stress levels consistent with those allowed by the ACI Code. The towers have been sized based on the intermittent load criteria primarily because of the cyclic nature of almost all the loadings and also the less definable fatigue characteristics of reinforced concrete. The present designs are also approaching yield at the 120 mph storm condition.

Another approach to WTG concrete tower design is to utilize pre-stressed and/or post-tensioned sections. An industry specialist, working along with General Electric, has evolved a preliminary 1500 kW design. The two fundamental variations from the more conventional G. E. approach has been to utilize a constant O.D. and I.D., a more efficient low quantity approach, and to longitudinally post tension the sections together with high strength bridge cable. Again, with the design flexibility of concrete, the combined advantages of several fundamentally different approaches would mature into the most cost-effective and aesthetically pleasing design for any particular site location or requirements.

5.3.5.4 Steel Truss Towers

In the truss tower design effort, several well developed computer codes to design the 500 kW and 1500 kW steel towers were utilized. The computer codes employed nodal finite element techniques and had the advantage of proven reliability. Once geometrically described, and with load inputs, the designs proceeded very quickly. All members specified for the 500 kW and 1500 kW towers are commercially available structural sections. Main supports and lower bays are comprised primarily of higher strength steels while the upper bays and most cross members are lower strength steels. In utilizing bolted construction and minimizing welds the intent has been to duplicate electric transmission line tower technology which is in a mature state of development. Since WTG towers experience greater loadings, however, it has not always been possible to choose convenient member shapes; I-beams were required for their load capability for some of the members. Since those type sections and a symmetrical tower are not convenient for easily bolted joints, some welding will be required. An attempt has been made to reduce the number of joints in the truss tower so as to minimize cost. Consistent with strength designs and the present dynamic definition, the truss towers have no special stiffener bracings which would increase the number of joints. However, depending on the results of a final dynamic analysis and design alterations, some selected stiffening may prove beneficial.

It should also be noted that galvanized steel is proposed for the towers, a three to five thousand dollar repaint being required every seven to ten years.

As in transmission line tower technology, the most efficient construction approach appears to be to prepare the members at a central location, transport, and then erect bays at ground level and hoist the bays aloft with a crane. The advantage, as with pre-cast concrete, is to minimize aloft construction. Secondary design refinements could also perturb the 6.10m (20 ft) bay height presently defined. The truss towers both have the same envelope sizes, a distinct feature that should be maintained. The idea is to maintain as much commonality as possible for the 500 kW and the 1500 kW towers, with only member sizes or strengths changing, the advantage being that for a large number of towers, the same point on a learning curve can be achieved more quickly. As in power towers, deep rooted foundations will be utilized.

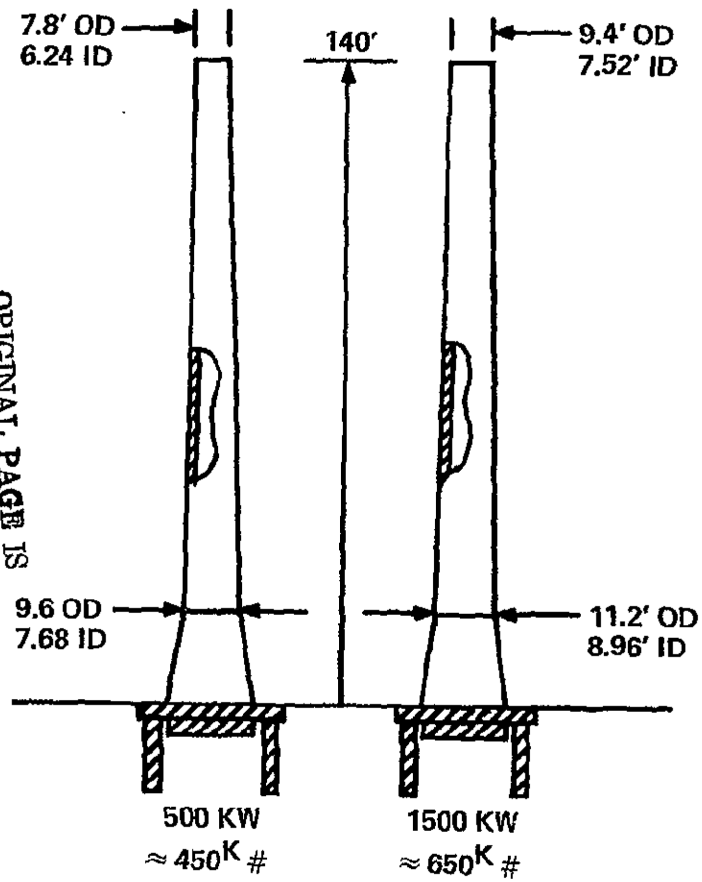
Special consideration is also being given to improving the appearance of the open truss towers. Contouring and shaping, along with painting and total site planning, are but a few of the preliminary ideas.

5.3.5.5 Tower Costs

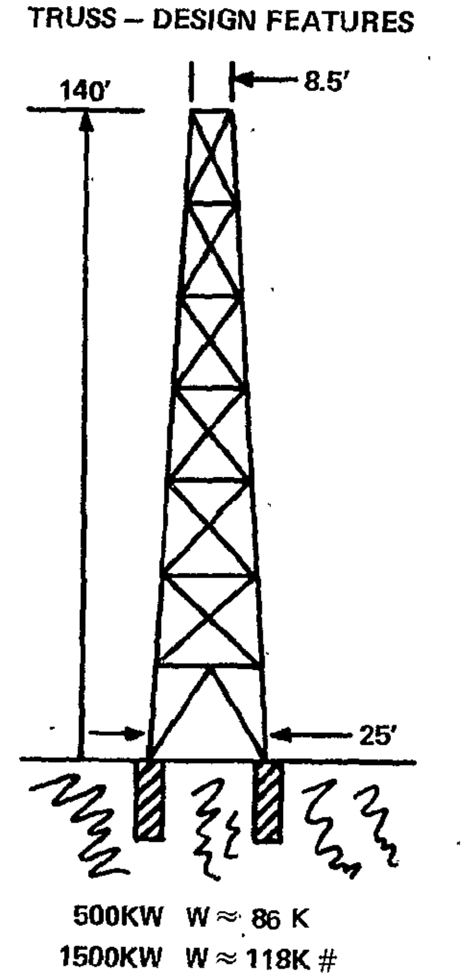
Most of the cost assumptions made during the conceptual design phase have been carried through to the preliminary design. The exception has been that for the reinforced concrete towers, foundation slabs and their corresponding costs have been included in the designs. Truss towers would generally require only deep-rooted pilings. The present costs, shown in Table 5-16, do not include a systems contractor's fee. The costs have been separated into two broad categories: small quantity and production attainable costs. The demarcation between the two categories is not presently defined as better site defintion, tower types, quantities, and location all directly affect materials and labor costs. The purpose of the costing to date has been to enable a comparison of concepts and to define estimated system cost per kilowatt-hr. The cost estimates for the truss tower are believed to accurately reflect current industry pricing, however, the concrete tower costs will be more sensitive to the specific site and the general economy.

TABLE 5-16. TOWER COSTS

		500 kW	1500 kW
Reinforced Concrete	Small Qty.	$45,200	$64,000
	Production Attainable	$39,300	$55,900
Steel Truss	Small Qty.	$90,700	$122,400
	Production Attainable	$73,800	$98,100



WTG RATED POWER LEVEL	ωNI BENDING	ωNI TORSIONAL	2ω EXCITATION
500 KW	.793 CPS	2.03 CPS	.96 CPS
1500 KW	.925 CPS	2.21 CPS	1.3 CPS



WTG RATED POWER LEVEL	ωNI BENDING	ωNI TORSIONAL	2ω EXCITATION
500 KW	1.66 CPS	4.3 CPS	0.96 CPS
1500 KW	1.23 CPS	4.4 CPS	1.3 CPS

Figure 5-60. Schematic Drawings of Tower Concepts

TABLE 5-17. TOWER WEIGHTS

	500 kW	1500 kW
Reinforced Concrete	204,500 kg (450,000#)	295,500 kg (650,000#)
Steel Truss	39,100 kg (86,000#)	53,600 kg (118,000#)

In comparing the two tower types, for the ássumptions made, reinforced concrete would overall, be lowest in price. It is estimated, however, that the actual and particular cost of any specific WTG is a strong function of the assumptions made. Generally speaking, steel structures would cost about $1 per pound while concrete would be about a tenth of that or $0.10 per pound of erected structure; the estimated tower weights are given in Table 5-17.

5.3.5.6 Nacelle/Pintle Structural Design

The nacelle structural assembly as shown in Figure 5-61 is that structure to which all of the power generation components are attached. The nacelle structural assembly contains five major substructures:

- Rotor Locking Flange
- Speed Increaser Structural Frame
- Generator Support Structure
- Structural Support Base
- Nacelle Adapter Flange

The basic design requirements for the nacelle support structure were:

1. Support necessary components and withstand all external and internal loadings.
2. Provide proper configuration layout
3. Insure axial and lateral rigidity between major components
4. Allow for personnel egress and maintenance
5. Use commercially available AISC structural steel members
6. Use AISC weld fabrication
7. Provide load transfer structure to tower pintle structural assembly

The structural support base is of stress skin construction and has an internal I-beam frame. This structure provides the platform to which all other structural members are attached. Located between the aft truss structure and the generator support structure is the speed increaser structural frame. This structure is bolted to both the gearbox and the fore and aft structural frames. Once bolted in place, the gearbox becomes part of the overall structure to provide torque reaction load paths and insure deflection rigidity between the mainshaft and gearbox and generator. In addition, the resulting structural frame reduces the bending loads into the support base and, therefore, more effectively utilizes the strength of the structural members.

Lateral members in the aft portion of the support base provide position guides and load paths for the main bearings. This type of welded structure made from available AISC structural steel can easily be fabricated for under $1.00/lb. The nacelle adapter flange is the only partially machined member in the structure and is fabricated in this manner due to the stiffness, strength, and flatness requirement of the azimuth bearing.

Maintenance access is obtained through the nacelle adapted flange, then through a hole in the structural support base.

The pintle structural assembly as shown in Figure 5-62 is located atop the WTG support structure or tower. The pintle serves 2 primary functions: incorporates an azimuth bearing and azimuth drive unit to align the rotor and nacelle with the wind, and stiff load paths to transfer the rotor-nacelle loads into the tower structure. The azimuth ring gear and bearing are bolted to a stiffening ring which is welded to an octagonal frame. The ring is machined for flatness requirements subsequent to weld fabrication. The azimuth ring gear is driven by the azimuth pinion gear via a drive shaft, double helical worm speed reducer, and an electric motor. The azimuth rate that was established during conceptual design is 1/3 rpm. This value was determined from the maximum tower torsional load capacity resulting from azimuth changes using the gyroscopic moment equation ($M=I\omega\Phi$). Also included in the drive unit is a slip clutch set to a given maximum value to prevent torsional gyroscopic reactions, due to wind shear or inflow, from being sustained.

The pintle structure itself is a weld fabrication of commercially available AISC steel wide flange beams.

Table 5-18 provides a cost and weight summary for the nacelle and pintle structure in both power sizes.

TABLE 5-18. NACELLE AND PINTLE STRUCTURAL ASSEMBLY COST AND WEIGHT SUMMARY

Component	500 kw_e Cost-$	500 kw_e Weight, kgs	(lbs)	1500 kw_e Cost-$	1500 kw_e Weight, kgs	(lbs)
NACELLE	7,000	3,270	(7,200)	13,000	7,730	(17,000)
PINTLE	13,000	4,640	(10,200)	27,000	7,730	(17,000)
TOTALS	$20,000	7,910	(17,400)	$40,000	15,460	(34,000)

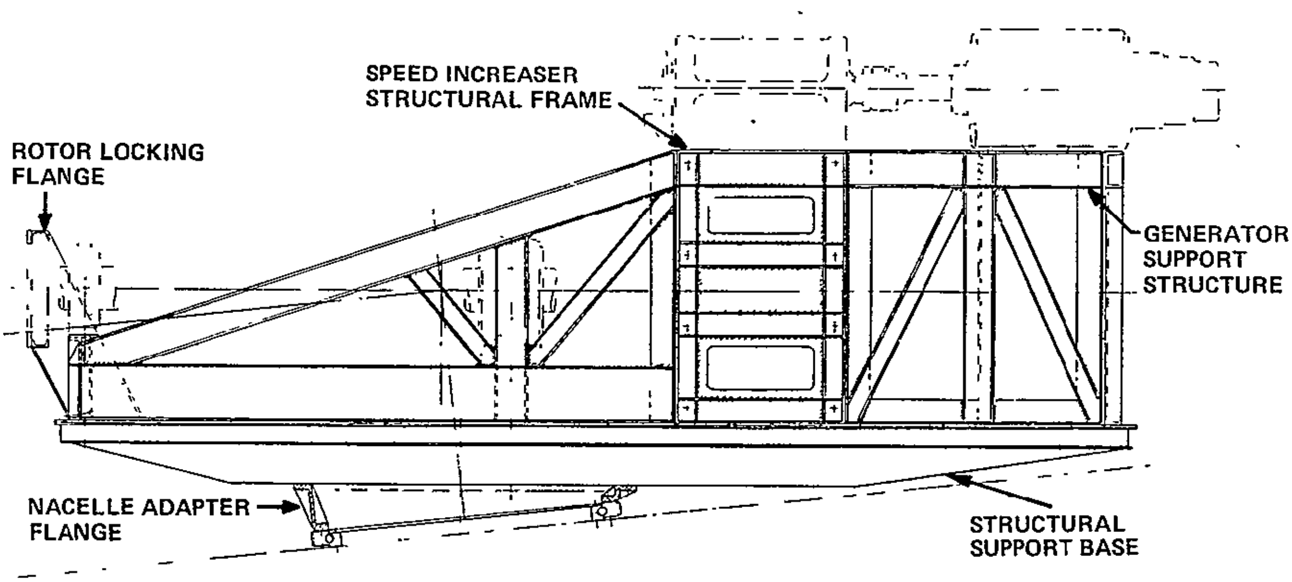


Figure 5-61. Nacelle Structural Assembly

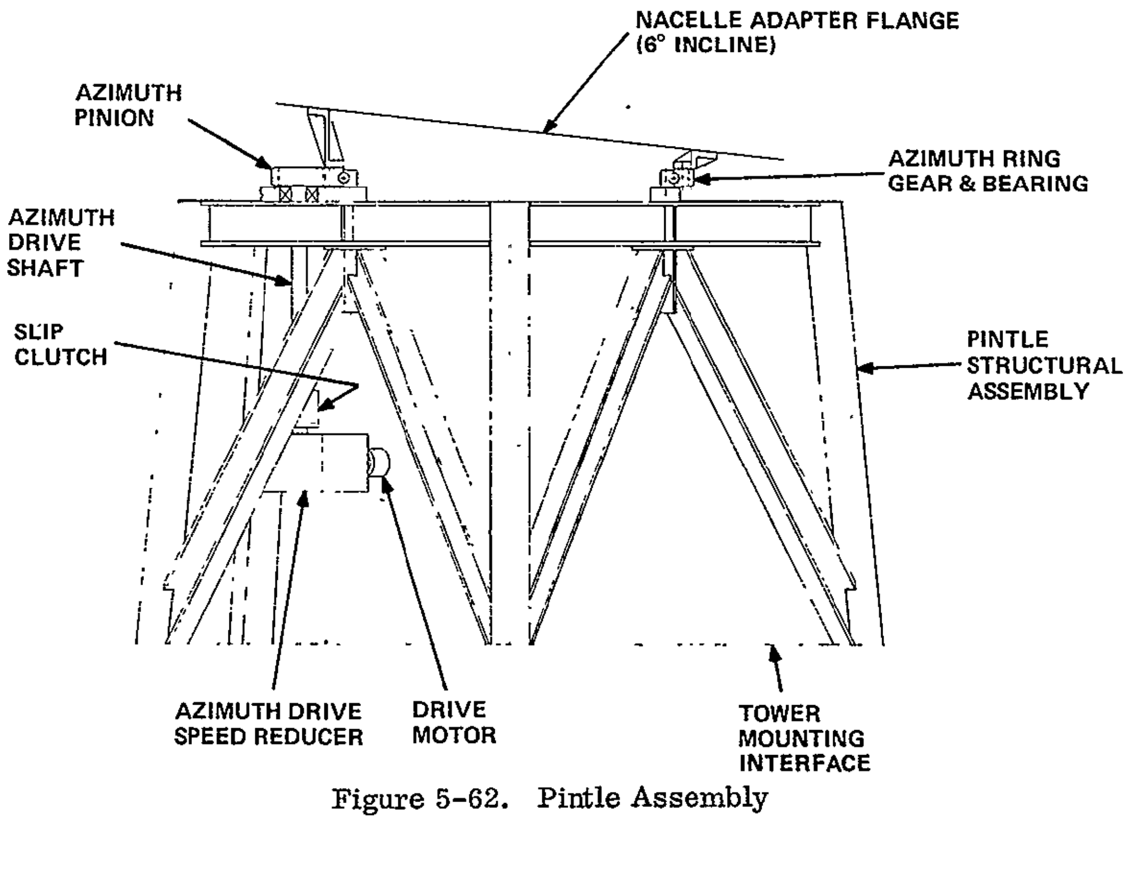


Figure 5-62. Pintle Assembly

SECTION 6.0

STUDY CONCLUSIONS

SECTION 6.0

STUDY CONCLUSIONS

As stated earlier, the principal objective of this study was to evaluate the incentive for utilizing WTG's in broad scale electrical power generation applications. During the program, conclusions were drawn concerning this question as well as more specific design, operating approach and fabrication alternatives. The purpose of this section is to first present the overall study conclusions and secondly, more specific results pertaining to the individual tasks.

6.1 OVERALL RESULTS

The principal results of this study included the following:

Site Selection ➜ • The selection of the site in terms of its median wind speed had an overwhelming effect upon the WTG economics

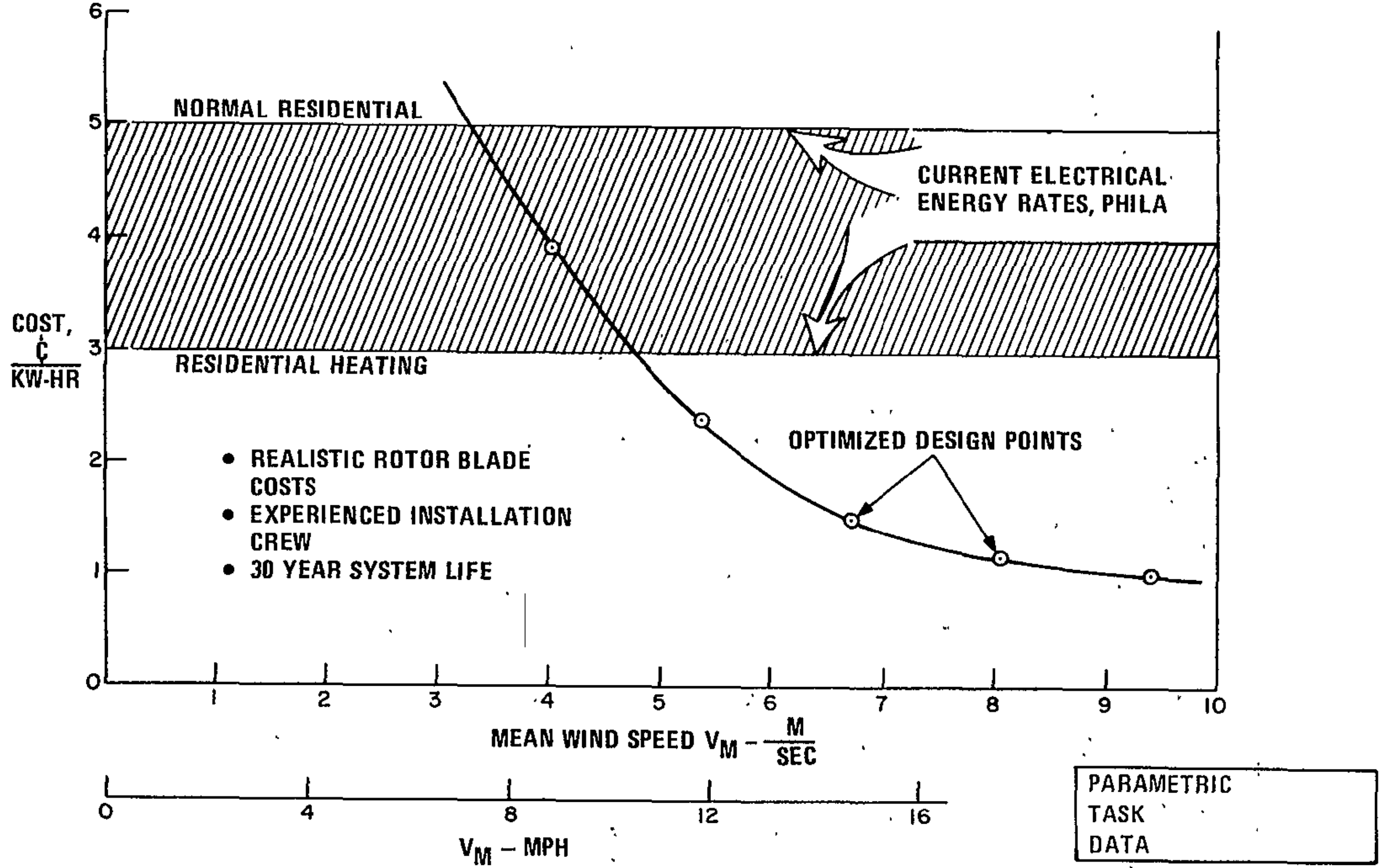
Power Level ➜ • Dramatic cost reductions were available by designing systems at power levels greater than 500 kW and significant cost reductions continued to accrue up to the 1500-2000 kW range

Cost ➜ • Two designs were investigated in detail - a 1500 kW unit in an 18 MPH site and a 500 kW unit in a 12 MPH site. The cost to generate power in the 18 and 12 MPH sites is estimated to be 1.57 and 4.04¢/kW-HR respectively, based on 100 production units. Capital costs were estimated to be $430/kW for the 1500 kW unit and $935/kW for the 500 kW unit.

The rotor assembly was determined to be the largest single contributor to the overall system cost.

Electrical Utility Integration ➜ • No major technical difficulties were uncovered in terms of linking the WTG in an electrical utility network. The WTG can provide electrical stability under steady state as well as gusting wind conditions and provide operational capacity factors in the 40-50 percent range.

The importance of site selection is clearly demonstrated by the data shown in Figure 6-1 where the cost of generating power is plotted against median wind speed. The data shown in this figure was generated by optimizing the WTG for each wind site on the basis of cost; then the locus of minimum cost systems for each wind site was selected to create the parametric data shown in Figure 6-1. On the basis



*POWER DISTRIBUTION COSTS NOT INCLUDED

Figure 6-1. Effect of Median Wind Velocity on Electrical Power Costs of Figure 6-1, primary attention must be given to the accumulation of reliable wind data so that accurate power generation costs can be predicted for candidate sites.

The second conclusion presented above is supported by the results shown previously in Figure 4-3; this data shows the cost trend as a function of power level for various median wind speeds. At power levels up to 500kW a sharp reduction in cost occurs due mainly to the economies of size. Above 500kW further decreases in power level are apparent, however, at a slower rate. The slower rate is due to the increasing expense of the transmission as a percentage of the total system cost; in fact, current limitations in going to higher power levels are dictated by the availability of larger transmissions. This constraint is also shown in Figure 4-3.

The ability to design economically for power levels higher than that permitted by the transmission constraint line is also hampered by unknowns concerning the rotor subsystem design. Figure 4-9 illustrates the optimum blade diameters consistent with the cost/power level data shown in Figure 4-9. This data, which is again parametric, was done for a blade clearance height from the ground of 6.10 m. The major point to be made from Figure 4-9 is the fact that large blade diameters are required to achieve high power level/low cost systems. The participants of this study determined that large blade diameters could best be achieved using propeller as opposed to helicopter technology. This design approach results in a simpler fabrication procedure, a lighter and more reliable blade and has the least impact on the overall system design. Also apparent in Figure 4-9 is a verification of the need to utilize sites having high median wind speeds.

The data presented in Figure 4-9 was generated using a cost expression in which the rotor subsystem cost was proportional to: $D^{2.22}$. An obvious concern arises as to the sensitivity of our large blade diameter/low cost conclusion with respect to the validity of our cost expression. Therefore, parametric data was generated to determine the effect of rotor subsystem cost uncertainties upon our basic conclusion. This analysis showed that low costs continued to be achieved at higher power levels in the range where the rotor subsystem related to diameter between: $D^{2.22}$ and $D^{3.0}$. Later, a detailed cost analysis performed during the Preliminary Design task showed the rotor subsystem cost to be proportional to: $D^{2.33}$. Therefore, the conclusion that lowest cost can be achieved by going to higher power systems having large blade diameters has been verified by a detailed analysis of rotor diameters in the 61.0 meter range.

As a result of the work performed during the Preliminary Design, it has been possible to determine the cost of WTG's with reasonable assurance. The two designs cited above (1500 kW/18 MPH and 500 kW/12 MPH units) have been presented in detail in Section 5.3. The cost estimate quoted above for the 1500 kW system is particularly attractive in terms of its cost due to the high median wind speed assumed, 18 MPH. The 500 kW system, although 2.5 times more expensive than the 1500 kW system on a ¢/kW-HR basis may have longer term potential considering the trend of conventional fuel costs.

The rotor subsystem was estimated to constitute about 35 percent of the total system cost under high production conditions. Therefore, the system design philosophy must be consistent with reducing the cost of this element. A major reason for the high cost of the rotor subsystem is that it must be designed for any unusual wind conditions that may occur over the lifetime of the unit. During the study, it was determined that the use of an intelligent control system which could avoid situations of destructive winds upon the blades would result in appreciable rotor subsystem cost reductions. Further efforts in this area may reduce the rotor subsystem to a smaller fraction of the system cost.

The fourth major conclusion resulting from this study is the fact that WTG's can be successfully integrated with an electrical utility network. The WTG would be linked to the utility by means of a 4160 V line to a transformer which would provide the interface with the network. Voltage fluctuations of less than 3-5 percent can be maintained by the WTG under both steady state and severe gusting wind conditions. Synchronization of the WTG System with the network under a variety of environmental conditions has been investigated in detail and can be performed routinely by modulation of the blade pitch angle. Although both synchronous and induction generators can be utilized in the WTG, the use of a synchronous generator is favored due to the higher power factor offered by this approach. In addition, the synchronous generator is better characterized, making the entire system more amenable to a detailed analysis.

The paragraphs above summarize the overall results obtained in this study; the following subsection reviews those conclusions which resulted from the Conceptual Design portion of the study.

6.2 CONCEPTUAL DESIGN RESULTS

This portion of the study focused on defining the WTG in terms of its overall configuration, operational mode and subsystem concepts. The principal conclusions found are listed below.

WTG Configuration

- The optimum configuration consisted of placing the power transmission/generator equipment atop of the tower. Efforts to place this equipment on the ground were precluded by limitations in existing hydraulic transmissions and the cost ineffectiveness of a mechanical approach, as discussed in Section 3.2.3. The potential cost benefits associated with equipment on the ground are lower maintenance, assembly and tower costs.

- Placing the rotor downwind from the tower is favored due to the inherent stability of this configuration to changes in wind direction. Also of importance is that this configuration minimizes the rotor overhang required to accommodate rotor blade deflections.

- The use of multiple rotors per tower was also considered during the Conceptual Design Phase. In comparing a 100 kW system having one rotor to a 100 kW system having 3 rotors atop a single tower it was estimated that the cost of the 3 rotor system would be 1.5 times that of the single rotor system.

Operating Mode

- Systems operating at constant RPM and at constant velocity ratio* were considered. The constant RPM system was preferred because it resulted in lower tower and blade loads as well as a lower cost transmission or generator. The single advantage of a constant velocity ratio system is the potential for higher energy capture; however, in the investigations performed, the increase in energy capture did not offset the higher system capital costs.

- Both variable pitch and fixed pitch systems were considered. Variable pitch systems using blade rotation was selected due to its excellent response characteristics under changing wind conditions.

Rotor Subsystem

- A two bladed system was selected, as opposed to three, on the basis of lower cost. The technical acceptability of this decision was verified later in the program.

- A rigid as opposed to a teetered hub was also selected on a tentative basis at this point in the study. A dynamic analysis of rotor/tower interactions verified the use of a rigid hub in conjunction with 2 blades; this analysis was performed during the Preliminary Design portion of the study.

- Filament wound blades were selected as the lowest cost approach. This concept is amenable to an efficient fabrication technique, provides a lightweight structure and requires minimum maintenance when formed in a propeller type structure. The use of a helicopter type blade, balanced at the quarter chord point, was unattractive due primarily to the following reasons:

  - higher fabrication costs

  - heavier in weight

  - higher maintenance costs

  In addition, the propeller concept had been demonstrated in European windmills and was judged to have less development risk.

Tower

Many tower concepts including truss, concrete, tube shell and pole designs were evaluated. The truss and concrete approaches appeared to be least costly with the truss having the advantage of better cost predictability while the concrete offers a more aesthetically pleasing design.

* Velocity ratio is the rotor tip speed divided by the wind speed

Transmission Subsystem

- Mechanical, hydraulic and electrical type transmissions were considered. The mechanical concept including: gearbox, belts, chains and combinations of the above were judged to be the least cost approach; of these the gearbox was preferred when maintenance aspects were addressed.

Generator

- Either synchronous or induction generators can be accommodated in the WTG; however, the synchronous generator is preferred due to its higher power factor and the fact that its performance is better characterized.

Control Subsystem

- A microcomputer was selected as the main control element in the system. The microcomputer offers an inexpensive approach to handle the many operating situations which can occur. Design emphasis on the control system permits a relaxation of the design requirements on other system components.

6.3 PARAMETRIC ANALYSIS RESULTS

Results from the Conceptual Design Task were modeled in a design optimization computer code to evaluate the effect of specific design variables on system cost and to select the design conditions for the Preliminary Designs. The principal conclusions of this task, which relate to overall system cost, are presented below.

The parametric analyses were conducted for sites having median wind speeds between 9 and 21 MPH (at a 9 MPH median wind site the wind is assumed to be above this level for one-half of the year). The wind duration curves used in this study are contained in Section 2.0 of this report.

The economic assumptions made in calculating energy costs are believed to be representative for an electrical utility company in today's environment. The assumptions made are:

- Depreciation Method - straight line
- Capitalization Method - 50% bonds, 50% stock
- Interest Rate - 9%
- Return on Equity - 11.5%
- Federal Tax Rate - 48%
- Maintenance cost included

Effect of Power on Cost

Figure 4-3, previously discussed in Section 4.3.1, illustrates parametric system costs (¢/kW-HR) versus system power level (kW) for wind sites having median speeds between 9 and 21 MPH. Figure 6-2 also describes the trend of capital costs, $/kW, as a function of power level; the results shown support the previously mentioned conclusion that higher power levels provide greater cost effectiveness.

Effect of Median Wind Speed on Cost

The fundamental importance of the wind speed at the site is evident from Figure 6-2. At higher wind velocities smaller rotor blade lengths are required for the same power level, thereby reducing rotor subsystem and tower costs. Higher wind speeds also result in a higher design RPM, consequently reducing transmission costs as well. The effect of modest interannual variations in median wind speed will not compromise the cost effectiveness of optimized systems; this conclusion is supported quantitatively in Figures 4-11 and 4-12.

WTG Annual Operating Times

A major concern regarding the use of WTG's in electrical utility applications is the amount of time during the year when power is available. Using the wind duration curves presented in Section 2.0, data was generated for the optimized systems to assess the annual expected operating time. Figure 6-3 illustrates the number of hours per year a WTG can be expected to deliver power as a function of median wind speed. In terms of total hours of operation, 5300 to 7200 hours can be expected for a 9 and 21 MPH median wind speed site, respectively.

Terrain Factors and Altitude

An important element involved in design of a WTG is the velocity gradient of the wind with height. This velocity gradient or "wind shear' affects the design of the WTG in several ways; of concern to this discussion is the effect on the system performance and resulting power generating cost. Figure 6-4 illustrates the effect of the terrain factor, ko, on system cost where ko is defined by the following expression:

$$V = Vo\ (h/ho)^{ko},$$ and:

V = wind velocity at some height

Vo = wind velocity at reference height

h = height

ho = reference height

Terrain factors of 0.15 are characteristic of open, flat land while values of 0.20 are typical for rougher topography with obstructions such as brush and trees. It is important to recognize that while higher terrain factors result in lower

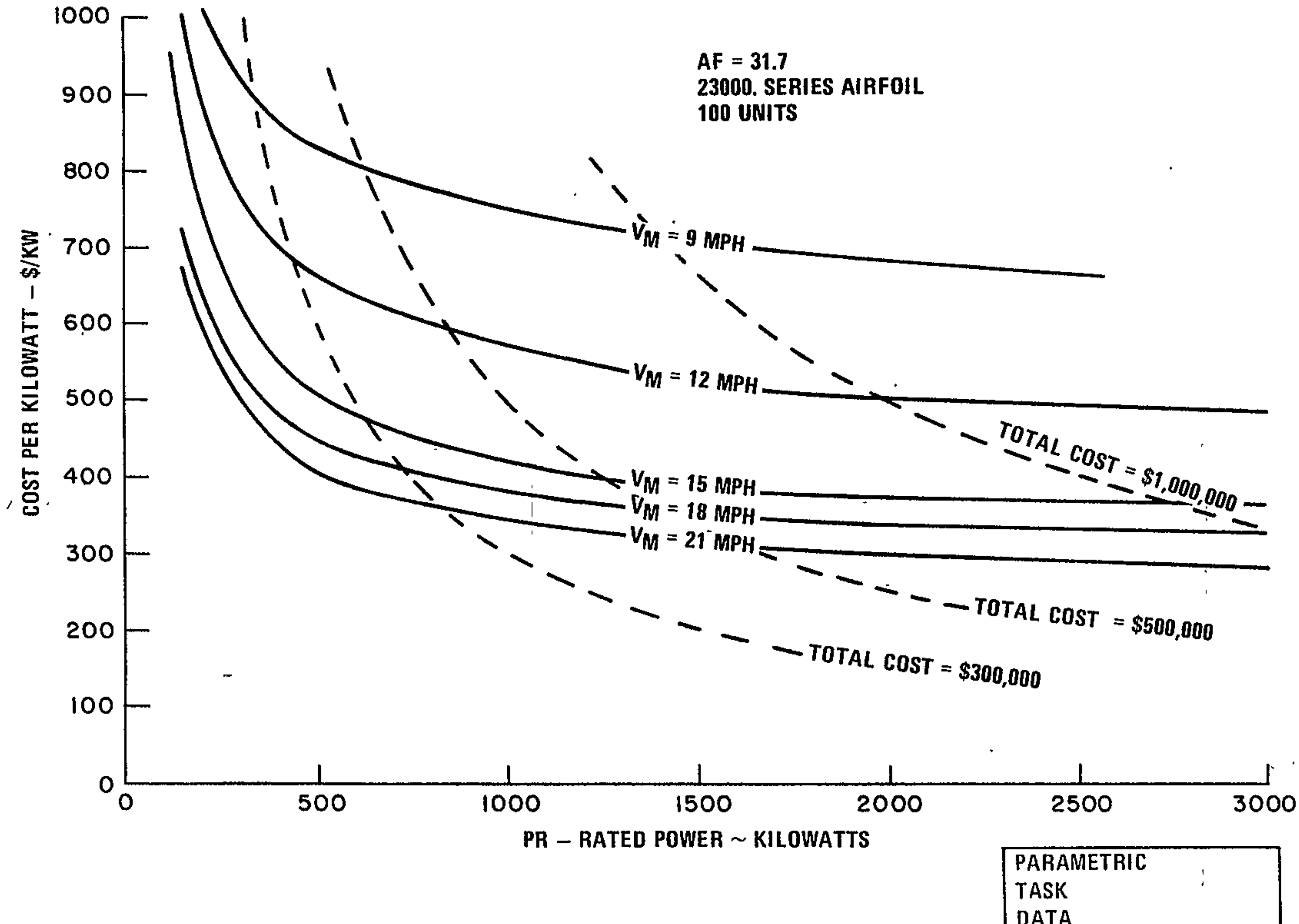


Figure 6-2. Capital Cost per Kilowatt vs. Rated Power

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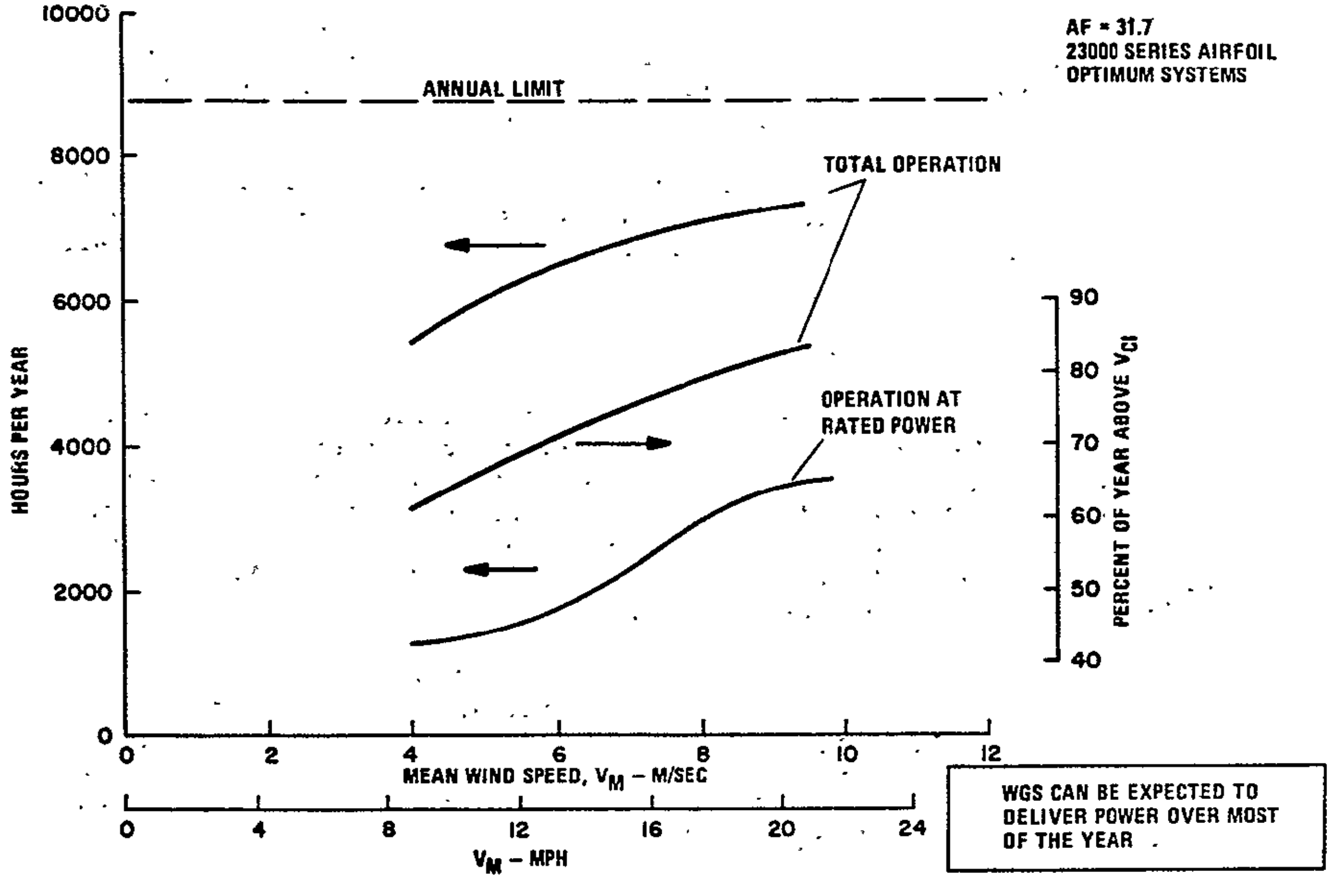

Figure 6-3. Annual Operating Time vs. Median Wind Speed

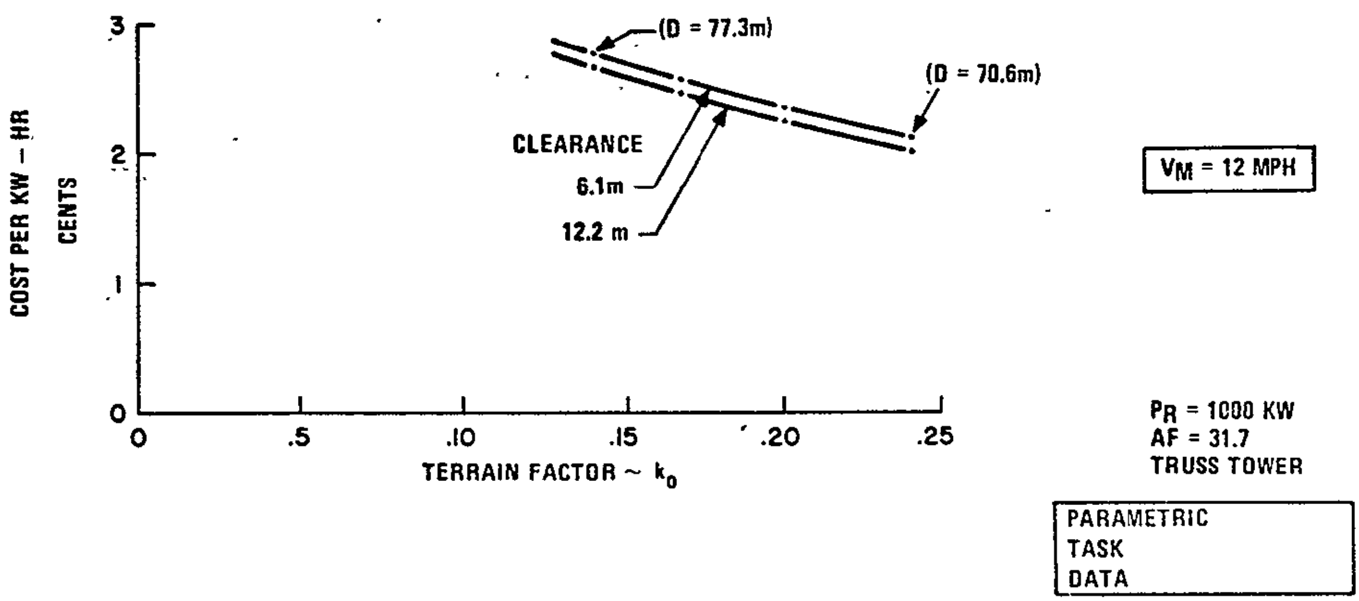

Figure 6-4. Cost Sensitivity of Tower Height
(as a function of wind shear and rotor clearance)

cost systems for the same median wind, it may be more difficult to find sites with higher terrain factors and high median winds. Also shown in Figure 6-4 is the effect on cost of the blade ground clearance height. Some benefit is apparent in increasing the ground clearance to 15.2 meter.

The effect of altitude (between sea level and 1.61km) on system performance and cost was also investigated. The results, shown in Figure 6-5, indicate that decreasing air density at higher altitudes has a minor inpact on overall system cost.

Effects of Lifetime on Cost

A 30 year system life was established to be consistent with other electric utility generation equipment and with utility long range financing methods. The effect of this assumption on system cost was investigated with results shown in Figure 6-6. Over the relevant range of 20 to 50 years the cost per kW-HR is relatively insensitive to assumed system lifetime. The slight rise is due to the diminished value of the depreciation deduction as a tax shield as life increases. This is somewhat offset by recurring equipment overhaul.

System life below 20 years is not realistic for the long term financing assumptions of Figure 6-6 and, in addition, may be constrained by IRS depreciation guidelines. Utilities are often permitted to use a shorter life for tax purposes, but typical values are 22½ years for a 30 year book life item. The 30 year system life assumption was concluded to be reasonable from an economic as well as a technical viewpoint based on utility practice and the cost insensitivity shown in Figure 6-6.

6.4 PRELIMINARY DESIGN RESULTS

Using the component concepts previously identified and the quantitative results of the parametric analyses, 500 kW and 1500 kW units were designed to operate in 12 and 18 MPH median wind sites, respectively. In addition to providing sufficient detail from which to calculate system costs, the Preliminary Design Task made apparent the design philosophy required to build cost effective WTG systems.

A necessary ingredient to designing low cost WTG's is to incorporate an intelligent control system which prevents the WTG from operating under abnormal design conditions.

This design approach, which is explained throughout Section 3.0, became fundamental in determining the operational modes for the WTG. Designing the system to withstand all types of wind condition/blade altitude situations or system component failures eliminates the possibility of a low cost system. The preferred approach is to design adequate sophistication into the WTG so as to minimize the probability of adverse operating conditions occurring.

The specific results of the Preliminary Design Task are outlined below.

Cost

Costs to generate power were estimated on the basis of 100 production units and include the system contractors overhead and fee. The costs for the 500 and

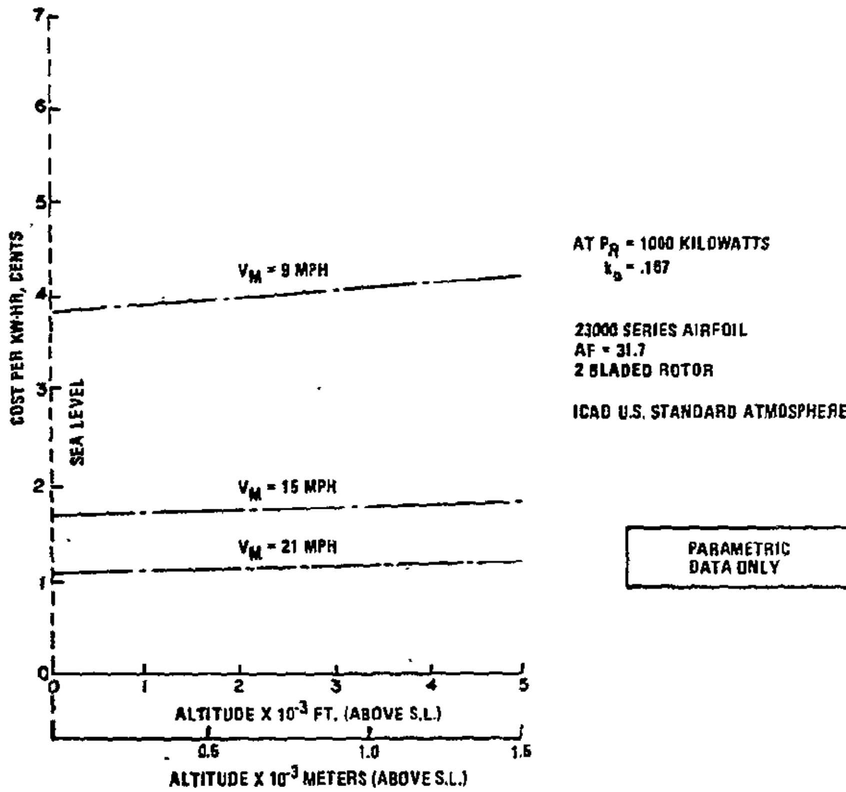

Figure 6-5. Cost Sensitivity to Altitude

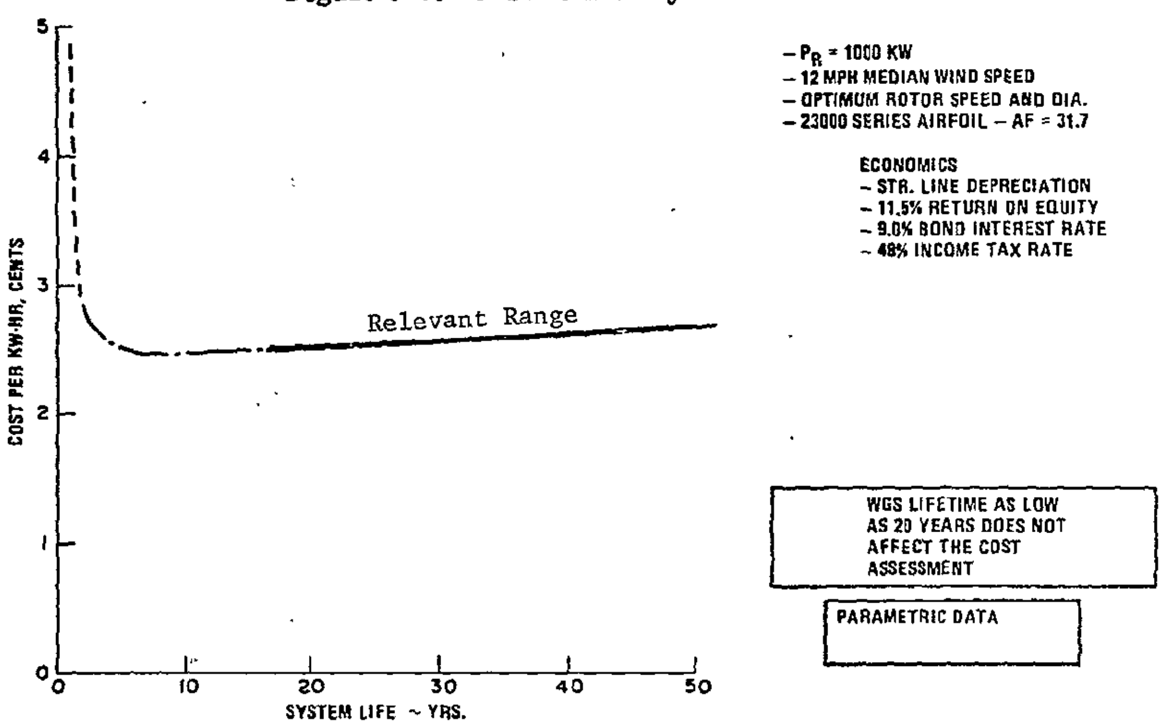

Figure 6-6. Effect of Lifetime Variation on Energy Cost

1500 kW units, including a 15 percent system contractors fee, are $466,670 and $643,655 respectively, or $935/kW and $430/kW. These capital cost figures translate into 4.04 and 1.57 ¢/kW-HR.

If the Preliminary Design costs quoted above are coupled with the parametric trends shown in earlier curves, it is possible to estimate costs for power levels and median wind sites other than the two Preliminary Design conditions.

Final Design and Fabrication Schedule

Figure 6-7 illustrates the proposed schedule to perform a final design, fabrication and assembly, and check-out of the 1500 kW WTG; the schedule for the 500 kW unit would be similar.

Fabrication Approach

The approach used in erecting and checking-out the WTG could have a significant impact on overall system cost. The logistics outlined during the Preliminary Design for these procedures included the following:

- Delivery of the rotor subsystem by the manufacturer direct to the site
- Assembly and check-out of the mechanical power transmission, generator, pintle and nacelle structure at an assembly plant
- Tower erection at the site by a local contractor under system contractor supervision
- Delivery of the nacelle to the site and placement atop the tower
- Mating of the rotor subsystem with the nacelle atop the tower

Variations of these procedures are possible depending upon the size of the WTG and the site location and condition.

Rotor/Tower Dynamic Interactions

Tower and rotor costs can be minimized by analyzing these two major subsystems simultaneously. Proper selection of operating speed and natural frequencies of the rotor and tower can result in less cost yet higher reliability by reducing system weight and strength requirements. Therefore, the rotor/tower system should be dynamically "tuned" so as to provide the most cost effective and technically reliable design.

Rotor Subsystem Design

Two rotor blades mounted on a rigid hub was found to offer the least cost rotor subsystem. Both 2 and 3 bladed systems and teetered as well as rigid hubs were evaluated. The increased cost and sophistication of the teetered hub was found to be unnecessary if large overturning moments due to wind directional changes can be avoided by the control system.

Tower

Analyses of both truss and concrete towers for the Preliminary Design indicates that for sites with good soil conditions, the concrete tower is lower in cost. Also of importance is the fact that concrete towers appear to be more aesthetically pleasing to the majority of people.

System Stability

Both the electrical and mechanical stability of WTG's were investigated under a variety of conditions. Electrical stability can be maintained by generator field control under most wind gusting conditions. In situations where there is a high impedance line to the network, pitch control may be required in addition to field control for low frequency wind gusts.

Mechanical stability must be provided in order to synchronize the WTG. Gain factor and speed loop stability analyses indicate that synchronization can be achieved under steady state and gusting situations by varying the blade pitch angle.

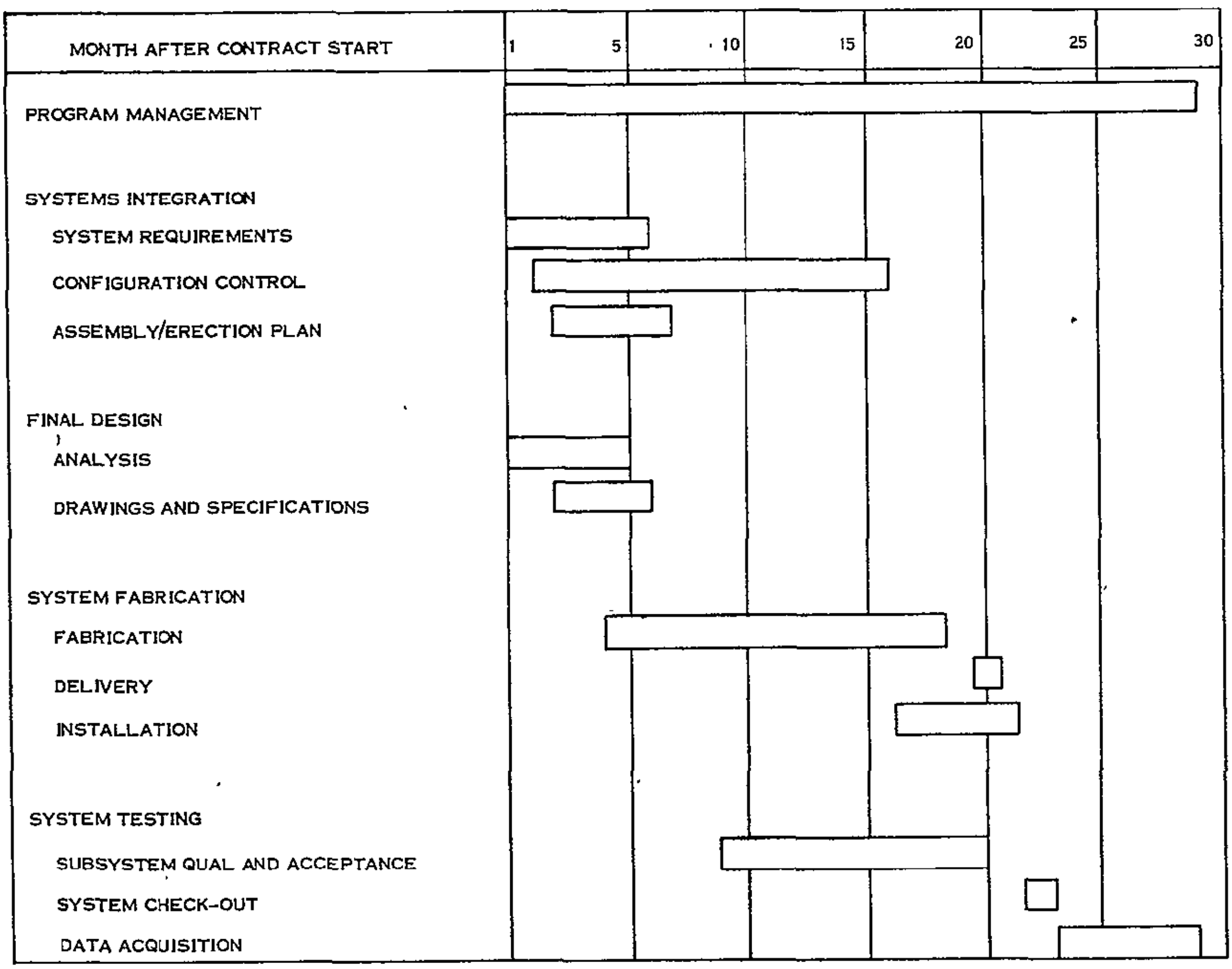

Figure 6-7. WTG Fabrication Schedule for Prototype Unit

SECTION 7.0

REFERENCES

SECTION 7.0

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SECTION 8

APPENDICES

APPENDIX 8.1

TERMS AND DEFINITIONS

8.1.1 NOMENCLATURE

AF	-	Rotor/blade activity factor
b	-	Blade chord length
C_1	-	Airfoil section lift coefficient
C_d	-	Airfoil section drag coefficient
C_p	-	Power Coefficient
C_Q	-	Torque Coefficient
C_T	-	Thrust Coefficient
C_L	-	Operating Lift Coefficient
D	-	Rotor Diameter
E	-	Energy
F	-	Force
H_R	-	Rotor axis height above ground level
H_0	-	Wind site data reference height
H_c	-	Rotor ground clearance height
h	-	Blade thickness
h_T	-	Tower height
I	-	Inertia
K	-	Spring constant
k	-	Exponential constant
L/D	-	Lift-to-Drag ratio
L		moment reference length
M	-	Moment
N_R	-	Rotor Shaft speed, RPM

N_G	-	Generator Shaft Speed, RPM
n	-	Number of units
PR	-	Power Ratio
P	-	Power
P_R	-	Rated power
P_E	-	Electric Power generated
Q	-	Torque
Q_R	-	Rated Torque
R	-	Blade radius
r	-	Blade radial location
T	-	Thrust
T_e	-	Equivalent Thrust
t	-	Temperature
V_O	-	Reference wind speed (@ Ho)
V_T	-	Blade tip speed
V_W	-	Wind speed
V_R	-	Rated wind speed
V_M	-	Annual mean wind speed
V_{BA}	-	Rotor Breakaway wind speed
V_{CI}	-	Generator cut-in wind speed
V_{CO}	-	Generator cut-out wind speed
W	-	Weight
X	-	Rotational reference axis
Y	-	Azimuth reference axis
Z	-	Lateral reference axis

Greek Symbols

α	-	Blade angle of attack
β	-	Blade pitch angle (@ 3/4R)
δ	-	Blade deflection
η_t	-	Transmission Efficiency
η_g	-	Generator Efficiency
θ	-	Blade twist
λ	-	Velocity Ratio
π	-	const. (3.14159)
ρ	-	Atmos. air density
σ	-	Stress
τ	-	Torque (electrical)
ϕ	-	Inflow angle
ψ	-	Azimuth angle
ω	-	Rotational Rate (Rad/sec)
$\dot{\Phi}$	-	Azimuth rotational rate

8.1.2 ACRONYMS

AC (VAC)	-	Alternating Current (volts-AC)
APS	-	Auxiliary Power Supply
BRK	-	Circuit Breaker
DC (VDC)	-	Direct Current (volts-DC)
FPS	-	Feet per second
HVAC	-	Heating, Ventilating and Air Conditioning
KVA	-	kilo-volt-amperes
KW	-	kilowatts
KW-HRS	-	kilowatt-hours
MPH	-	miles per hour
MPS	-	meters per second
MVA	-	mega-volt-amperes
P/S	-	power supply
PF	-	power factor
RPM	-	revolutions per minute
RAM	-	random access memory
ROM	-	read-only-memory
UPS	-	uninterrupted power supply
WTG	-	wind turbine generator

8.1.3 DEFINITION OF TERMS AND PARAMETERS

The following list of terms and parameters are defined as used in the text of this report.

1. Median Wind Speed, V_M - is the wind speed, characteristic of the site, for which actual wind speeds exceed that value for one-half of the year and are less than that value for one-half of the year. (NOTE: Statistically and meteorologically defined as the median wind speed.)

2. Rated Wind Speed, V_R - is the minimum wind speed at which rated power of the wind turbine generator is achieved.

3. Breakaway Wind Speed, V_{BA} - is the minimum wind speed at which the rotor absorbs sufficient power to overcome friction and begin to rotate.

4. Cut-In Wind Speed, V_{CI} - is the minimum wind speed at which the generator may be cut-in and produce power. It also represents the lower limit of the wind speed spectrum for generator cut-out.

5. Cut-Out Wind Speed, V_{CO} - is the maximum wind speed at which the generator power output can be adequately regulated.

6. Maximum Wind Speed, V_{MAX} - is the maximum wind speed for which the wind turbine system is designed to survive in a shut-down mode.

7. Wind Duration Curve, (V_O vs. HRS./YR) - is a statistical representation of the total amount of time in one year that the wind is at or above any given speed, which is characteristic of the site.

8. Wind Shear, V_W/V_O - is the wind speed gradient, characteristic of the site, with height referenced to ground level (as opposed to sea level).

9. Blade Tip Speed, V_T - is the local in-plane wind velocity at the tip of the blade due to the rotational speed of the rotor.

10. Velocity Ratio, V_T/V_W - is the ratio of the rotor blade tip speed to the wind speed and represents a performance parameter which relates the rotor speed (RPM), the rotor diameter and the wind velocity to rotor blade performance.

11. Design Velocity Ratio, λ_{DES} - is the velocity ratio at which a given rotor design achieves peak aerodynamic efficiency (CP_{MAX}).

12. Rated Velocity Ratio, λ_R - is the velocity ratio at which a given rotor design achieves rated power output.

13. Power Coefficient, C_p - is a non-dimensional performance parameter that characterizes the rotor design in terms of its capability to extract power from the winds: (Also, the same as ROTOR AERODYNAMIC EFFICIENCY and POWER RATIO, which are numerically equal to shaft power extracted divided by the power available in the wind.)

14. Rated Power, P_R - is the power level rating of the system and is governed by the capabilities of the generator. It also represents the power level that is maintained by power regulation at wind speeds above the rated wind speed.

15. Blade Airfoil - is the aerodynamic contour of the blade cross-section, which provides the blade performance characteristics in terms of lift and drag. (Lift and drag forces acting on the blades are resolved into the resultant torque and thrust absorbed by the rotor when integrated over the blade length.)

16. Blade Thickness Ratio, h/b - is the ratio of maximum airfoil thickness to airfoil chord length at any given radial station along the blade. The blade thickness ratio dictates the airfoil section lift-to-drag ratio and allows the compromises between aerodynamic performance and blade structural requirements.

17. Blade Pitch Angle, β - is the angle between the blade setting and the plane of rotation and is referenced at the 3/4-blade radius station.

18. Blade Twist, θ - is the variation in the blade pitch angle from the root to the tip of the blade. Twist is incorporated into the blade design to enhance high aerodynamic performance over the entire length of the blade and prevent blade tip stall at high wind speeds.

19. Blade Taper, b/D - is the variation in blade chord length with the radial location along the blade. Taper is incorporated in the blade design to enhance the load distribution on the blades and the blade structural requirements.

20. Blade Activity Factor, AF - is a non-dimensional performance parameter, which characterizes blade planform area and area distribution in terms of both aerodynamic and structural performance. Activity factor incorporates the effects of taper as well as blade solidity (area ratio of blade planform to rotor disc)

21. Blade Conding - is the leeward inclination of the blades from the rotor hub. Either a variable or a constant coning angle may be utilized to relieve root bending stresses in the blades, and to minimize rotor dynamic moments with variable coning.

22. Rotor Teetering - is the variable inclination of the rotor disc to absorb the asymmetric loading due to wind shear and tower shadowing and to minimize rotor dynamic moments.

23. Rotor Tilt - is a constant inclination of the rotor shaft axis and is often incorporated in the design for blade-tower clearance.

24. Rotor Azimuth (Yaw) Angle, ψ - is the polar orientation angle of the rotor axis.

25. Rotor Inflow Angle, ϕ - is the angle between the rotor rotational axis and the free stream wind direction.

APPENDIX 8.2

WTG TOWER/ROTOR DYNAMIC ANALYSIS

This Appendix contains the results of a preliminary dynamic analysis performed by the General Electric Co. during both the parametric analyses and Preliminary Design Phases of NASA Contract NAS3-19403. The purpose of the study was to analyze the generalized wind generator system with specific emphasis on tower/rotor interactions. The analysis was performed with the aid of a Beckman 2200 Analog Computer which enabled convenient manipulation of the model parameters.

The wind generator systems simulated were generalized in nature and did not represent any particular or specific designs, as the intent was to determine the best overall approach to alleviating potential adverse tower/rotor dynamic interactions. General Electric Co.'s 500KW reference design, evolved under the above mentioned contract during the preliminary design phase, was used as a reference, however, against which to determine typical values and ratios of the many parameters studied.

The basic options studied were two and three bladed propellor type rotors mounted on vertically cantilevered towers. The two bladed rotors were either of the hingeless or teetering hub type. The models were three of four degrees of freedom analogs which focused on the lateral bending characteristics of both the blades and the WTG towers. That so called flapping mode was considered most critical. All the models were limited in that torsional couplings and higher order bending modes were neglected.

The range of model parameters varied during the study enabled significant insight into many controversial questions on WTG operating philosophies and basic design approaches. The addressing of such questions as whether towers should be designed on the basis of strength or stiffness, whether the teetered hub was necessarily dynamically better than the hingeless or three bladed approach was a primary study objective. Such issues have such a large bearing on both operational and cost feasibility that an early design analysis, even though limited in nature, was considered very important.

8.2.1 SUMMARY

The following general conclusions are a result of this study:

1. The totally undamped teetering hub is unstable.

2. The teetering hub approach with damping can reduce maximum mainshaft moments by 20% to 45% over the 2 blade rigid hub design.

3. The teetering hub approach with damping will generally reduce both blade and tower peak displacements and oscillating range displacements over the 2 blade rigid hub design, but not to a significant degree.

4. Teetering motions of approximately 1^{o} or less are sufficient; further increases are of little benefit.

5. The three blade rotor design is dynamically superior to either the hingeless or teetering 2 blade design.

6. Towers designed on the basis of strength rather than stiffness are a more cost effective approach and can be dynamically more attractive for certain wind generator systems.

8.2.2 ANALYTICAL MODEL DESCRIPTION

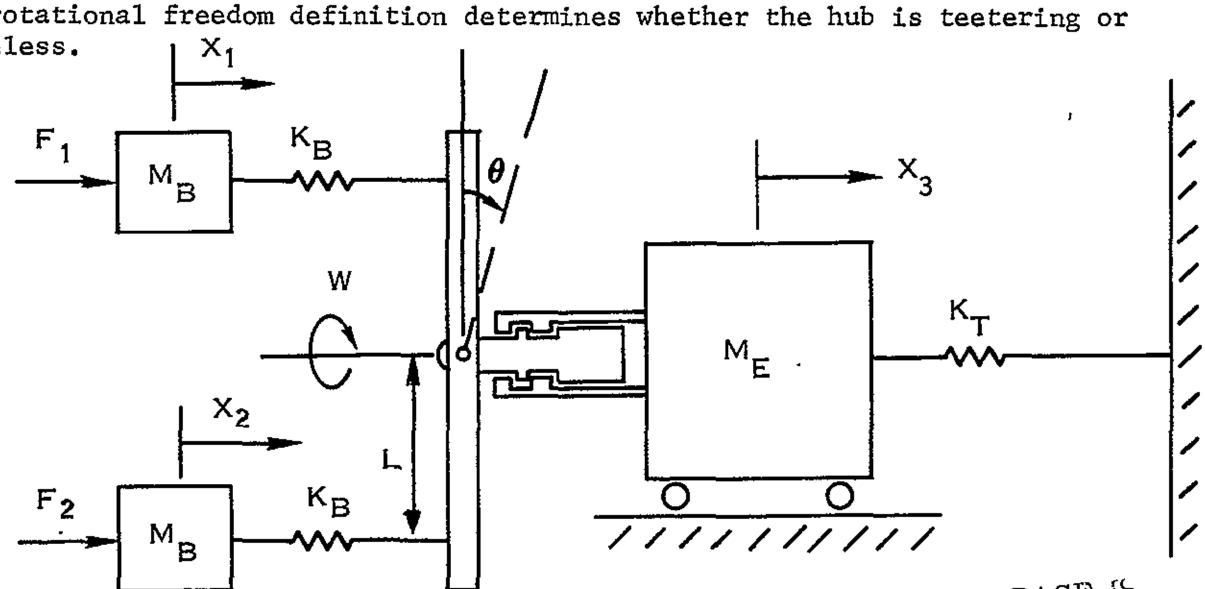
A basic analytical approach was to model the WTG as a collection of springs and masses. All the blade weight was assumed to be concentrated at the blade center of mass. The weight attached to the representative tower spring was assumed to be comprised of the weight of all the equipment atop the tower, excluding the blades. Both the blade and tower model spring rates would be analogous to the actual blade and tower net lateral bending stiffness. The out of plane (of the model) rotational effects were included in formulating the equations of motion. Thus the increase in blade stiffness due to centrifugal action was included.

The model equations of motion were derived by utilizing Lagrange's Equations. The two bladed teetering hub model became a rigid hub model in the simulation be restricting the angular motion of the blades. The analog simulation was wired such that a single switch controlled the option of a teetering or a rigid two blade hub. Further development of the simulation allowed easy changeover to the three bladed simulation.

In developing the models the usual assumptions of frictionless surfaces and small oscillations (i.e. $\sin\theta \approx \theta$) was followed. All runs were made with the initial conditions of the springs and masses in the undistrubed and unforced static condition.

8.2.2.1 Two Bladed Rotor

The following diagram illustrates the dynamic model of a two bladed windmill. The rotational freedom definition determines whether the hub is teetering or hingeless.



Two Bladed Analog Model

The following definition of parameters applies:

$X_1 X_2$ - blade lateral bending displacement - ft.

X_3 - tower lateral bending displacement - ft.

θ - rotation (i.e. teetering motion)

ω - rotor rpm

$F_1\ F_2$ - forcing functions due to vertical wind shear, tower shadow and/or wind gusting

M_b - blade mass

M_e - upper equipment mass

l - distance from hub to blade center of mass

K_b - blade bending net spring rate

K_t - tower bending net spring rate

C - damping

The following equations of motion apply:

(1) $$M_b(\ddot{X}_1 + \ell\ddot{\theta} + \ddot{X}_3) + M_b \ell \omega^2 (X_1/\ell + \theta) + K_b X_1 = F_1$$

(2) $$M_b(X_2 - \ell\ddot{\theta} + \ddot{X}_3) + M_b \ell \omega^2 (X_2/\ell - \theta) + K_b X_2 = F_2$$

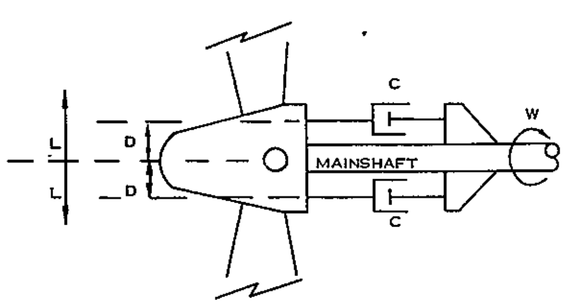
(3) $$M_b(\ddot{X}_1 + \ddot{X}_2) + (M_e + 2M_b)\ddot{X}_3 + K_t X_3 = F_1 + F_2$$

(4) $$M_b(\ddot{X}_1 - \ddot{X}_2 + 2\ell\ddot{\theta}) + M_b \omega^2 (X_1/\ell - X_2/\ell + 2\theta) = F_1 - F_2$$

or

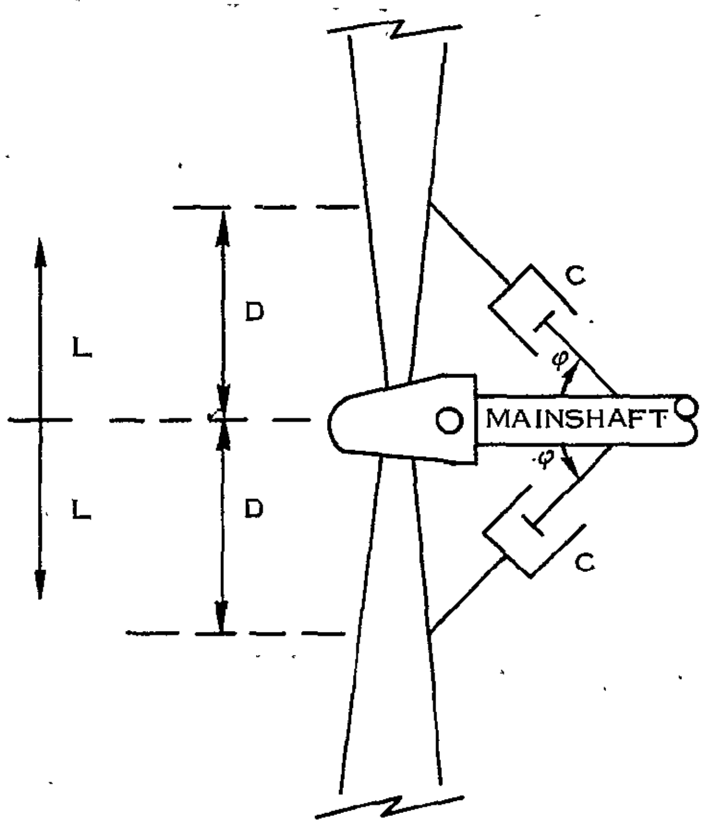
(4a) $$M_b(\ddot{X}_1 - \ddot{X}_2 + 2\ell\ddot{\theta}) + M_b \omega^2 (X_1/\ell - X_2/\ell + 2\theta) = F_1 - F_2 - 2C\dot{\theta}d/\ell$$

Equation (4a) applies when the teetering motion is damped. In that condition, the following schematic would describe a typical hub mechanism.



Two Bladed, Teetered Rotor with Damping on Hub

Another type of damped teetering and the corresponding equations are shown below. This approach was not investigated in the present study but would be recommended for future study:



Two Bladed, Teetered Rotor, Blades Damped

(5) $$M_b(\ddot{X}_1 + \ell\ddot{\theta} + \ddot{X}_3) + M_b \ell\omega^2(X_1/\ell + \theta) + K_b X_1 = F_1 - C\dot{X}_1/\ell \cos\varphi \; d/\ell$$

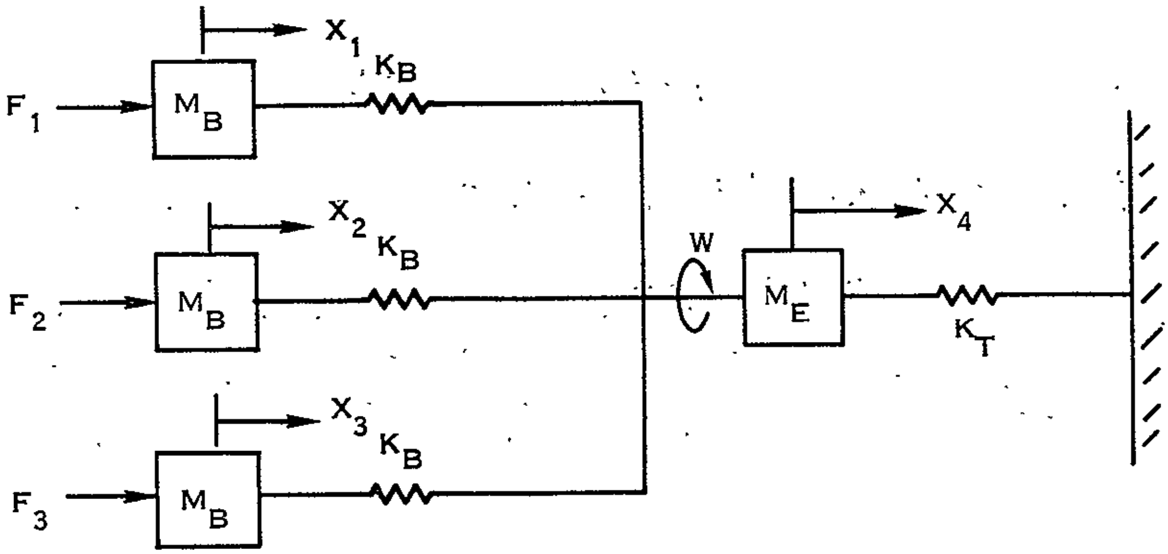
(6) $$M_b(\ddot{X}_2 - \ell\theta + \ddot{X}_3) + M_b \ell\omega^2(X_2/\ell - \theta) + K_b X_2 = F_2 - C\dot{X}_2/\ell \cos\varphi \; d/\ell$$

(7) $$M_b(\ddot{X}_1 + \ddot{X}_2) + (M_e + 2M_b)\ddot{X}_3 + K_t X_3 = F_1 + F_2$$

(8) $$M_b(\ddot{X}_1 - \ddot{X}_2 + 2\ell\ddot{\theta}) + M_b \ell\omega^2(X_1/\ell - X_2/\ell + 2\theta) = F_1 - F_2 - 2C\dot{\theta} \cos\varphi \; d/\ell$$

8.2.2.2 Three Bladed Rotor

The following diagram illustrates the dynamic model of a three bladed windmill.



The definition of parameters for the three bladed windmill is:

$X_1\ X_2\ X_3$-	blade lateral bending displacements - ft.
X_4-	tower lateral bending displacement - ft.
ω -	rotor rpm
$F_1\ F_2\ F_3$-	forcing functions due to vertical wind shear, tower shadow, and/or wind gusting
M_b-	blade mass
M_e-	upper equipment mass
K_b-	blade bending net spring rate
K_t-	tower bending net spring rate

The following equations of motion apply:

(9) $M_b \ddot{X}_1 = -K_b (X_1 - X_4) + F_1 - \omega^2 X_1 M_b$

(10) $$M_b \ddot{X}_2 = -K_b(X_2 - X_4) + F_2 - \omega^2 X_2 M_b$$

(11) $$M_b \ddot{X}_3 = -K_b(X_3 - X_4) + F_3 - \omega^2 X_3 M_b$$

(12) $$M_e \ddot{X}_4 = -K_b(X_4 - X_1) - K_b(X_4 - X_2) - K_b(X_4 - X_3) - K_t X_4$$

The only damping introduced in the simulation was that to control the teetering motion. There was no attempt to simulate structural or aerodynamic damping. Thus the time to reach steady state conditions in the model would be somewhat greater than with the actual windmill. It is well to note that there is a small amount of motion damping always present due to the inherent nature of analog machinery with the actual amount difficult to determine. It is also well to note that the simulation was utilized as a design tool as far as damping was concerned. A value was set on a potentiometer to produce a desired result with the physical significance of that value to be determined later.

Several parameters were held constant between the two and three bladed models. The blades masses were 139#m and the value of M_e was 1950#m except where noted. In parametrically varying the spring rates no attempt was made to correspondingly change the blade masses, as would be the actual physical case. Instead, the simulation could be viewed as a study of blades with different bending characteristics and materials but all of the same weight. Also in the models, ℓ was taken to be approximately 62 feet. A recheck reveals that the distance is too great for the analogous physical windmill, but since the model is concerned with relative changes in response with parametric variation, the basic results and conclusions are unaffected. The vertical wind shear was described as follows:

Two bladed rotor

$$F_1 = (a_0 + a_1 \sin \omega t)^2 \quad (13)$$

$$F_2 = (a_0 + a_1 \sin(\omega t + \pi))^2 \quad (14)$$

Three bladed rotor

$$F_1 = (a_0 + a_1 \sin \omega t)^2 \quad (15)$$

$$F_2 = (a_0 + a_1 \sin(\omega t + 2\pi/3))^2 \quad (16)$$

$$F_3 = (a_0 + a_1 \sin(\omega t + 4\pi/3))^2 \quad (17)$$

In other words the wind velocity power function was linearized over the range of interest of the model. In each case it was assumed that the mean wind speed was 12 mph so that the total thrust of 12500# was constant between the models. Although not strictly correct, but for the purposes of better comparison, the three bladed rotor was assumed to turn at one half the rate of the two bladed system.

The effect of the tower shadow was in most cases superimposed on the wind shear function. Preliminary calculations showed about a 20% decrease in blade loading for about a 20° duration while the blade was within the tower shadow. The magnitude of unloading was varied in some of the runs.

The effect of wind gusting was also superimposed on the other forcing functions. A power spectral density function was derived from the NASA-Lewis wind model and simulated by electronic circuitry. Gusting was generally not included in most runs because the random nature of that function made comparisons among runs more difficult.

Seismic forcing functions were not run. They would be much more meaningful were the simulation used to study the response of a specific design with know parameters.

8.2.3 DISCUSSION OF RESULTS

The overall results, even with only three or four degrees of freedom studied in the flapping mode, clearly confirm trends that have resulted from much more rigorous analysis, especially in the rotor area. Differences between two bladed hingeless and teetering and three bladed designs are very evident.

Prior to discussing specific results, the following key and table are introduced.

KEY

K_t - tower stiffness

ω_{nt} - tower natural frequency;

X_{ts} - maximum static tower displacement

X_{td} - max. steady state dynamic tower displacement

$Range_t$ - max. and min. steady state dynamic tower displacements

MF_t - tower displacement magnification;

K_t - blade stiffness

ω_{nb} - blade natural frequency

X_{bs} - max. static blade displacement

X_{bd} - max. steady state dynamic blade displacement

$Range_b$ - max. and min. steady state dynamic blade displacements

MF_b - blade displacement magnification;

rpm - rotor speed

Moment - out of plane net maximum steady state mainshaft bending moment

C - potentiometer setting (i.e. damping)

θ - teetering rotation

Except where noted, all runs were made with vertical wind shear and tower shadow effect as the two superimposed forcing functions.

It was learned in the simulation that the totally undamped teetering hub is unstable (Table 8-1, run #15). Many other runs were made at different spring rates and the same situation holds true. The teetering motion will increase at the rotor turns such that the rotor disc would have a net wobbling motion of destructive magnitude. A rewriting of equation (4),

$$(18) \quad \ddot{\theta} + \omega^2\theta = \frac{F_1}{2\ell M_b} - \frac{F_2}{2\ell M_b} - \frac{\ddot{X}_1}{2\ell} + \frac{\ddot{X}_2}{2\ell} - \frac{\omega^2 X_1}{2\ell} + \frac{\omega^2 X_2}{2\ell}$$

reveals that the natural frequency of teetering is always at the exciting frequency. A solution to the homogenous form of (18) reveals that

$$(19) \quad \ddot{\theta} = -A\,\omega^2 \sin \omega t - B\,\omega^2 \cos \omega t$$

Comparison with (13) and (14) clearly illustrates the point. As soon as damping is introduced, however, the teetered natural frequency changes enough to stabilize the system. For a purely teetering hub there can be no moments transmitted to the mainshaft due to wind shear, tower shadow, inflow angle or whatever loadings. When damping is introduced, however, reaction forces are reintroduced into the system such that the moments are no longer eliminated. The approach then is to allow enough teetering to stabilize the motion but to minimize moments. When the teetering motion was allowed to vary during the simulation, it was found that there was no change in the moments even for teetering angles of up to 15°. The net result then is that a damped teetering motion of about 1° or less is all that is required. Less damping would increase the teeter amplitude but not effect the moments. It was for that reason that in runs #2 through #18 only approximately .5° of teetering was studied. An examination of those runs reveals that, over a wide range of tower and blade stiffnesses, that one could expect about a 20% to 45% reduction in moment with a teetered hub of less than 1° motion, depending on the relative tower and blade stiffnesses. The reduction would also depend on the wind shear and tower shadow magnitudes. Greater teetering action through less dampening would be academic. It is also probable, but not confirmed in the present analysis, that aerodynamic damping would be insufficient to limit the teetering oscillations to less than 1°. A mechanical implementation would probably be necessary as illustrated in previous figures. Teetering of up to 1° also has the benefit of reducing displacement magnification factors for both the blades and tower. That is undoubtably due to reduced inertial loadings, again over a wide range of combined stiffnesses. For heavier blades that reduction would probably be even greater. It is well to note that not only are the peak dynamic deflections reduced but the displacement

TABLE 8-1. RESULTS OF ANALOG COMPUTER ANALYSIS

	K_t #/ftx10^{-3}	W_{nt} r/s	X_{ts} ft.	X_{td} ft.	$Range_t$ ft.	MF_t	K_b #/ft	W_{nb} r/s	X_{bs} ft.	X_{bd} ft.	$Range_t$ ft.	MF_b	RPM	Moment ft-#x10^{-3}	C	θ
#1	110	7.52	0.118	0.16	0.16 & 0.06	1.36	7902	7.54	0.822	0.80	0.80 & 0.45	0.97	28.8	150 & -115	∞	0
	65	6.60	0.153	0.23	0.23 & 0.05	1.5	7902	7.54	0.822	0.926	0.926 & 0.36	1.12	28.8	± 200	∞	0
	60	5.55	0.217	0.26	0.26 & 0.14	1.20	7902	7.54	0.822	1.10	1.10 & 0.20	1.34	28.8	± 170	∞	0
	35	4.24	0.371	0.38	0.38 & 0.30	1.02	7902	7.54	0.822	0.90	0.90 & 0.35	1.09	28.8	± 180	∞	0
	15	2.77	0.867	0.875	0.875 & 0.725	1.01	7902	7.54	0.822	1.0	1.0 & 0.25	1.22	28.8	± 220	∞	0
#2	110	7.52	0.118	0.16	0.16 & 0.06	1.36	7902	7.54	0.822	0.80	0.80 & 0.45	0.97	28.8	150 & -115	∞	0
	110	7.52	0.118	0.16	0.16 & 0.06	1.36	7902	7.54	0.822	0.80	0.80 & 0.45	0.97	28.8	± 175	1.	<0.5°
#3	60	5.55	0.217	0.26	0.26 & 0.14	1.20	7902	7.54	0.822	1.10	1.10 & 0.20	1.34	28.8	± 170	∞	0
	60	5.55	0.217	0.26	0.26 & 0.14	1.20	7902	7.54	0.822	1.10	1.10 & 0.20	1.34	28.8	± 130	1.	<0.5°
#4	15	2.77	0.867	0.875	0.875 & 0.725	1.01	7902	7.54	0.822	1.0	1.0 & 0.25	1.22	28.8	± 220	∞	0
	15	2.77	0.867	0.875	0.875 & 0.725	1.01	7902	7.54	0.822	0.95	0.95 & 0.30	1.156	28.8	± 120	1.	<0.5°
#5	110	7.52	0.118	0.24	0.24 & -0.025	2.03	2845	4.524	2.28	2.8	2.8 & 0.10	1.23	28.8	± 180	∞	0
	110	7.52	0.118	0.22	0.22 & -0.01	1.86	2845	4.524	2.28	2.6	2.6 & 0.30	1.14	28.8	± 120	1.	<0.5°
#6	60	5.55	0.217	0.32	0.32 & 0.06	1.47	2845	4.524	2.28	2.1	2.1 & 0.65	0.92	28.8	± 180	∞	0
	60	5.55	0.217	0.28	0.28 & 0.11	1.29	2845	4.524	2.28	1.9	1.9 & 1.0	0.83	28.8	± 120	1.	<0.5°
#7	15	2.77	0.867	1.10	1.10 & 0.40	1.27	2845	4.524	2.28	3.3	3.3 & -0.40	1.45	28.8	± 200	∞	0
	15	2.77	0.867	0.95	0.95 & 0.65	1.10	2845	4.524	2.28	2.6	2.6 & 0.40	1.14	28.8	± 120	1.	<0.5°
#8	110	7.52	0.118	0.32	0.32 & -0.12	2.71	1264	3.016	5.14	6.5	6.5 & -1.75	1.26	28.8	± 160	∞	0
	110	7.52	0.118	0.37	0.37 & -0.10	3.14	1264	3.016	5.14	7.0	7.0 & -2.5	1.36	28.8	± 120	1.	<0.5°
#9	60	5.55	0.217	0.65	0.65 & -0.275	2.99	1264	3.016	5.14	4.25	4.25 & -0.50	0.83	28.8	± 160	∞	0
	60	5.55	0.217	0.575	0.575 & -0.175	2.65	1264	3.016	5.14	3.50	3.50 & 0.50	0.681	28.8	± 120	1.	<0.5°
#10	15	2.77	0.867	0.90	0.90 & 0.66	1.04	1264	3.016	5.14	3.5	3.5 & 1.0	0.681	28.8	± 160	∞	0
	15	2.77	0.867	0.85	0.85 & 0.75	0.98	1264	3.016	5.14	3.10	3.1 & 1.5	0.603	28.8	± 120	1.	<0.5°
#11	17738	3.016	0.704	2.2	2.2 & 2.0	3.13	7902	7.54	0.822	0.90	0.90 & 0.40	1.09	28.8	± 120	∞	0
	17738	3.016	0.704	2.2	2.2 & 2.0	3.125	7902	7.54	0.822	0.90	0.90 & 0.40	1.09	28.8	+ 120 & -130	1.	<0.5°
#12	20266	3.22	0.617	0.90	0.90 & 0.40	1.46	7902	7.54	0.822	0.90	0.90 & 0.40	1.09	28.8	± 120	∞	0
	20266	3.22	0.617	0.85	0.85 & 0.35	1.38	7902	7.54	0.822	0.85	0.85 & 0.60	1.03	28.8	± 120	1.	<0.5°
#13	81065	6.44	0.154	0.18	0.18 & 0.12	1.17	7902	7.54	0.822	0.80	0.80 & 0.45	0.973	28.8	± 150	∞	0
	81065	6.44	0.154	0.18	0.18 & 0.12	1.17	7902	7.54	0.822	0.80	0.80 & 0.50	0.973	28.8	± 120	1.	<0.5°
#14	110	7.52	0.118	0.20	0.20 & 0.0	1.695	5057	6.032	1.29	1.4	1.4 & 0.60	1.085	28.8	± 130	∞	0
	110	7.52	0.118	0.20	0.20 & 0.0	1.695	5057	6.032	1.29	1.4	1.4 & 0.60	1.085	28.8	± 120	1.	<0.5°
#15	110	7.52	0.118	∞	∞	∞	7902	7.54	0.822	∞	∞	∞	28.8	-	0	∞
	110	7.52	0.118	∞	amplitude	∞	7902	7.54	0.822	∞	amplitude	∞	28.8	-	1.	<0.5°
#19	110	7.52	0.118	0.082	0.082 & 0.047	0.695	7902	7.54	0.527	0.664	0.664 & 0.3125	1.26	14.4	48400 & 0	∞	0
	65	6.60	0.153	0.14	0.14 & 0.039	0.915	7902	7.54	0.527	0.586	0.586 & 0.117	1.11	14.4	48400 & 0	∞	0
	60	5.55	0.217	0.148	0.148 & 0.102	0.682	7902	7.54	0.527	0.625	0.625 & 0.273	1.19	14.4	49200 & 0	∞	0
	35	4.24	0.371	0.243	0.243 & 0.188	0.656	7902	7.54	0.527	0.609	0.609 & 0.273	1.16	14.4	49200 & 0	∞	0
	15	2.77	0.867	0.55	0.55 & 0.429	0.634	7902	7.54	0.527	0.609	0.609 & 0.273	1.16	14.4	49200 & 0	∞	0
#20	110	7.52	0.118	0.25	0.25 & -0.135	2.12	2845	4.524	1.46	2.96	2.96 & -0.761	2.03	14.4	38,500 & 0	∞	0
	60	5.55	0.217	0.215	0.215 & 0.081	0.991	2845	4.524	1.46	1.56	1.56 & 0.76	1.07	14.4	29000 & 10485	∞	0
	15	2.77	0.867	0.555	0.555 & 0.406	0.64	2845	4.524	1.46	1.33	1.33 & 0.937	0.911	14.4	27950 & 12500	∞	0
#21	110	7.52	0.118	0.094	0.094 & 0.046	0.797	1264	3.016	3.296	2.97	2.97 & 1.48	0.901	14.4	23550 & 12180	∞	0
	60	5.55	0.217	0.181	0.181 & 0.083	0.834	1264	3.016	3.296	2.97	2.97 & 1.31	0.901	14.4	25000 & 12180	∞	0
	15	2.77	0.867	0.586	0.586 & 0.39	0.675	1264	3.016	3.296	2.34	2.34 & 1.64	0.71	14.4	24800 & 11860	∞	0
#22	17738	3.016	0.704	tower resonance			7902	7.54	0.527	tower resonance			14.4	resonance	∞	0
	20266	3.22	0.617	0.469	0.469 & 0.261	0.76	7902	7.54	0.527	0.625	0.625 & 0.25	1.186	14.4	49400 & 0	∞	0
	81065	6.44	0.154	0.32	0.32 & -0.156	2.08	7902	7.54	0.527	1.25	1.25 & 0.625	2.37	14.4	48400 & 0	∞	0
	110	7.52	0.118	0.094	0.094 & 0.039	0.797	5057	6.032	0.824	0.898	0.898 & 0.398	1.09	14.4	52290 & 0	∞	0

2

range is similarly reduced. That is a very important factor in designing long life blades, the relative value depending on the reversing blade dead weight loads.

In comparing the three bladed system with either case of the two bladed rotor, it is clear that shaft moments, dynamic magnification and range oscillations all generally decrease for the three bladed rotor. The tower especially benefits from the three bladed system. The loads are considerably alleviated. When remembering that each blade in a three bladed rotor is likely to be lighter than each in a two blade system, the three blade advantage to the rotor would seem to be even greater than indicated by the models.

An overall examination of the data also reveals, as suspected, that there is a fair sensitivity to tower stiffness on blade loadings. Significant overall advantage can be gained from tuning the tower and blades in relation to any particular rotor speed (excitation frequency).

A particularly interesting result is that for blades that are approximately 2.5P (see runs 1-4) tower dynamic magnifications tend to increase with increasing tower stiffness for 2 blade rotors. The implication is that as the tower is stiffened, at additional cost, it must absorb greater loads so that a stiffness design may be no better from a strength standpoint than a tower designed on the basis of strength to begin with. Since the tower "sees" only 2 as the fundamental excitation, it would seem as practical and more economical to design a tower with a natural frequency less than the 2 harmonic exciting frequency if the start up and synchronizing sequence allows powering through the resonance speed. Such an approach would make the use of high strength steels more attractive in lattice towers as much less weight would be required for the same strength. Reinforced concrete towers would also become more attractive as would pre-loaded steel towers. A pre-loaded steel tower approach would utilize an initial compressive preload in the members such that liveloads would not result in a tensile stress condition. In that way, much greater utility could be made of the total strength of the steel on a long term fatigue basis. Also with a softer tower, the time for resonant amplitudes to reach the same degree of severity is greater, so that more time could be allotted for powering through the critical region. It is also well to keep in mind that for larger systems, there is a greater percentage increase in tower cost to attain the same degree of additional stiffening.

Further examination of runs 1-4 and 15 also reveal some interesting trends. For the two blade case (run 15), it is clear that the tower shadow effect has a large bearing on tower loads for the rigid hub design. Its reduction also slightly decreases blade loads and shaft moments. Teetering again does not significantly alleviate the situation. An increase of wind shear magnitude difference (i.e., less WTG ground clearance) tends to adversely affect every motion. The addition of wind gusting to the basic wind shear and tower shadow did not produce any surprises. The overall response was stable and followed the overall excursions.

For the three bladed rotor (runs 19-22), the effect of increasing the wind shear is also shown to be a worst condition. The analysis indicates the tower shadow effect to be significant. Both the magnitude and duration of the blade unloading are important parameters in determining the harmonic content of the tower shadow forcing function. Since that should be known in relation to both blade and tower stiffnesses, it becomes an especially difficult problem to design a tower whose own size and shape not only determines its own dynamic characteristics, but whose size and shape determine the nature of the forcing function acting on itself. It would seem desirable to undertake a further Fourier analysis of several tower shadow situations in order to determine if there is a consistent harmonic nature to the excitations as a function of blade unloadings and shadow duration.

Several runs were made with the weight atop the tower reduced by a little less than 50% for the stiff blade and tower values for both the two and three blade systems. The tower displacements were about 25% less for the two blader while the three blader entered a resonance condition. More runs would have to be made of course, but mounting the generating equipment on the ground appears to dynamically be a step in the right direction.

Overall, the simulations have shown that not only are there significant differences among the three rotor types studied, but significant gain is possible within any particular approach by properly tuning the rotor/tower combination to not only avoid the usual resonances, but also to reduce the magnitude and range of displacements and moments. The importance of the dynamic interactions has been underscored. The results of this report would have to be combined with a cost analysis in order to provide the most cost effective dynamic approach.

APPENDIX 8.3

GUST MODEL

A significant portion of the design investigations conducted during the course of this study related to the electrical and mechanical stability of the WTG. As a means of proceeding with these studies assumptions were made concerning the behavior of the wind. This appendix describes the Gust Model which was used; it was obtained from the Wind Power Office at the NASA Lewis Research Center.

Preliminary Wind Model for Wind Energy Machine Dynamics and Control Analyses

The environments provided herein are intended for application to the dynamics and control problems associated with the design of wind energy machines. These environments consist of two parts; namely, mean wind or steady-state wind environments and discrete gust inputs. It is intended that these environments be used for wind conditions associated with mean wind speeds at the 10 meter level which are greater than or approximately equal to 8 m sec^{-1}.

1. Mean wind profile - For the altitudes involved and for the wind machines to be used at the LeRC Plumbrook facility, a reasonable representation of the steady-state wind profile for engineering applications is given by

$$\bar{u}(z) = \bar{u}_r \left[\frac{\ln(z/z_o)}{\ln(z_r/z_o)}\right] \qquad (1)^*$$

   where $\bar{u}(z)$ is the mean wind at z above natural grade, $\bar{u}_r$ is the mean wind at a reference level z_r and z_o is the surface roughness length. The surface roughness length for the Plumbrook site will be determined from analyses of Plumbrook wind profile data; however, until that task is accomplished a recommended value for engineering applications is $z_o \approx 0.2$ m.

2. Discrete gust environments - The recommended discrete longitudinal, lateral or vertical gusts $V_i(t, z)$ $(i=1,2,3)$ at altitude z as a function of time t to be used with the mean wind profile given in item 1 above are given by

$$V_i(t,z) = 0, \qquad -\infty \le t \le 0$$

$$V_i(t,z) = \frac{A_i(T, z)}{2}\left(1-\cos\frac{2\pi t}{T}\right), \qquad 0 \le t \le T \qquad (2)$$

$$V_i(t,z) = 0, \qquad T \le t \le \infty$$

* ln is the natural logarithm.

where A_i is the gust amplitude and T is the gust period. The gust amplitude is a function of the gust period T and the altitude z, namely

longitudinal: $$A_1 = z \frac{\overline{u}_r}{\ln(z_r/z_o)} \left[1-\exp\left(-\frac{\overline{u}(z)T}{1.48z}\right) \right]^{1/2} \tag{3}$$

lateral: $$A_2 = 2.4 \frac{\overline{u}_r}{\ln(z_r/z_o)} \left[1 - \left(1 - \frac{\overline{u}(z)T}{3.79z}\right)^2 \exp\left(-\overline{u}(z)T/0.95z\right) \right]^{1/2} \tag{4}$$

vertical: $$A_3 = 1.5 \frac{\overline{u}_r}{\ln(z_r/z_o)} \left[1-\left(1-\frac{\overline{u}(z)T}{1.48z}\right)^2 \exp\left(-\overline{u}(z)T/0.37z\right) \right]^{1/2} \tag{5}$$

3. Combination gust and wind profile - For engineering purposes, the discrete gusts proposed herein are statistically independent and thus should not be applied simultaneously. To apply the discrete gusts in item 2 above, each gust should be considered separately. A combined mean wind/discrete gust environment is obtained by vertical addition of the gust in question with mean wind at height z.

4. Application to wind mills - To assess the dynamics and control of wind mills, it is recommended that as a first approximation that the mill be completely immersed in the gust and that gust gradients (across disk of rotation) effects be neglected. An estimate of the discrete gust impinging upon the blades can then be obtained by evaluating the discrete gusts at the axis of rotation, z_a say. These gusts would then be applied over the complete disk or rotation of the blades. In the cause of the longitudinal gust we have

$$A_1 = 3 \frac{\overline{u}_r}{\ln(z_r/z_o)} \left(1-\exp\left(-\overline{u}\,(z_a)\;T/1.48z_a\right)\right)^{1/2} \tag{6}$$

with similar expressions for the lateral and vertical gusts. The quantity T should be selected over a range of values which encompasses the significart periods of response of the wind mill. The wind mill dynamics and control characteristics should then be assessed for a sufficient number of values of T over this range of period T to guarantee that all frequencies of concern are taken into account.

Discrete gust models which account for the gradient of gust velocity across the disk or rotation of a wind mill should also be taken into account. However, further discussions with the wind energy personnel at LeRC will be required to determine the kind of input which is needed.

APPENDIX 8.4

SYSTEM COMPONENT LISTING

This section lists the major subsystems and subsystem components which are embodied in the Preliminary Designs developed during the study. This list served as a basis for calculating the anticipated hardware costs of the WTGs presented.

8.4.1 ROTOR ASSEMBLY

8.4.1.1 Rotor Blades

1. Blade Shell
2. Spar Support
3. Hub Adapter Flange
4. Blade Lightning Conductor Strips
5. Blade Tip Aircraft Beacon
6. Beacon Operation/Fault Indicator

8.4.1.2 Rotor Hub

1. Blade Adapter Flange
2. Blade Pitch Bearings (Radial & Thrust)
3. Blade Pitch Actuator
4. Blade Pitch Control Mechanism (shaft/gearing)
5. Main Shaft Adapter Flange
6. Hub Structure and Braces

8.4.2 POWER TRANSMISSION DRIVE ASSEMBLY

8.4.2.1 Main Shaft

1. Hub Adapter Flange/bolts
2. Main Shaft
3. Flexible Coupling Adapter Flange/bolts
4. Main Shaft Position Lock Mechanism
5. Internal Pitch Control Shaft Bearings

8.4.2.2 Main Shaft Bearings (split)

1. Main Support (radial) Bearing (fwd.)
2. Main Support (thrust/radial) bearing (aft.)
3. Bearing Pillow Blocks
4. Main Shaft Bearing Temp. Sensors (^{o}F)
5. Main Shaft Motion Sensors (accelerometers)

8.4.2.3 Main Shaft Flexible Coupling

8.4.2.4 Gear Box - Speed Increaser (incl. housing, lub. system)

1. Low Speed Shaft (flex. coupl.) adapter flange
2. High speed shaft (flex. coupl.) adapter flange
3. Gear box heating system (oil sump)
4. Gear box oil temp. sensor (2-way) (^{o}F)
5. Gear box oil level indicator (green/red)

8.4.2.5 Primary Brake (hydraulic disc type)

1. High speed shaft adapter flange (bolts)
2. Structural mounting brackets
3. Hydraulic pump
4. Hydraulic Reservoir
5. Hydraulic pressure sensor (psi-green/red)

8.4.2.6 Generator (high speed) Shaft

1. Flexible coupling
2. Flex. coupl. adapter flange (bolts)
3. Generator shaft torque/speed sensor

8.4.3 ELECTRICAL SYSTEM ASSEMBLY

8.4.3.1 AC Synchronous Generator (1)

1. Resistance temp. detectors (6)
2. AC Brushless exciter

3. Voltage regulator
4. 1 KVA Transformer (voltage regulator)
5. Excitation switch (voltage regulator)
6. Surge Capacitors (3)
7. Lightning arrestors (3)
8. Grounding Transformer and resistor
9. Power connection box

8.4.3.2 <u>Generator Power Instrumentation</u>

1. Generator current sensors
2. Generator peak current sensors
3. Generator voltage sensors
4. Generator peak voltage sensors
5. Instrument transformers, voltage (3)
6. Instrument transformers, current (4)
7. Instrumentation/control connection box

8.4.3.3 <u>Power Cable Conduit</u>

1. Main power cable
2. Control power cable
3. Relay power cable
4. Junction boxes (4)
5. Clamps

8.4.3.4 <u>Power Cable Wrap-Up Loop</u>

1. Flexible power cable
2. Helical loop former (snubbers)
3. Junction box (1)

8.4.3.5 <u>Electrical Control Cable Conduit</u>

1. Multi-conductor shielded cable

2. Connector boxes (2)

3. Clamps

8.4.3.6 <u>Auxiliary Power Transformer</u> (6 KVA)

1. AP circuit breaker

2. AP contactor (shunt coil & overload elements)

3. AP sensor/monitor (on-off)

4. AP signal conditioner

8.4.3.7 <u>Emergency Power Generating System</u> (6 KVA)

1. Gasoline engine

2. Gasoline tank

3. 3 Phase AC generator/excitor/volt regulator

4. Battery and battery charger

5. EP circuit breaker (molded case)

6. EP automatic throwover (transfer)

8.4.3.8 <u>Control Building</u>

1. House Keeping load center (circuit breakers, bus)

2. Circuit breaker position indicators (on-off)

3. Equipment Racks for electronics

4. Power Connection

    - Pot heads (3)

    - Instrument Transformers, current (3)

    - Instrument Transformers, Voltage (3)

5. Main power contactor compartments

6. Main Power contactor (to utility)

    - Shunt control

    - Sensor/monitor

7. HVAC unit (air conditioning/ducting)

8.4.4 CONTROL SYSTEM ASSEMBLY

8.4.4.1 Data Handling System

1. Microcomputer
2. Multiplexer
3. Input/output interfaces
4. Optical isolators
5. Isolated sensors
6. Decoders
7. Relay drivers

8.4.4.2 Electronic Power Supply

1. Analog to digital devices
2. Signal conditioner
3. Electronic power indicator (on-off)

8.4.4.3 Uninterrupted Electronic Power Supply

1. UPS battery, keeper
2. UPS trickle charger, line filter
3. UPS electronic power indicator (on-off)
4. UPS electronic signal ground

8.4.4.4 Data Set Phone

1. Data auxiliary set (phone)
2. Data set interface
3. Dial up phone
4. Automatic call unit & interface (microcomputer)
5. Acoustic coupler
6. Utility Monitor/control
   - Teletypewriter interface
   - Teletypewriter

- Portable teletypewriter interface

8.4.4.5 Blade Pitch control System

1. Blade pitch control motor
2. Blade pitch angle sensor
3. Blade pitch angle lock
4. Blade pitch drive power indicator (on-off)
5. Blade pitch control relay

8.4.4.6 Azimuth Orientation Control System

1. Azimuth drive motor
2. Azimuth drive power indicator (on-off)
3. Azimuth drive motor relay

8.4.4.7 Primary Brake Control System

1. Hydraulic Pump Motor
2. Hydraulic brake operation indicator (on-off)
3. Hydraulic Pump Motor Relay

8.4.4.8 Gear Box Heater Control System

1. Gear box heater power indicator (on-off)
2. Gear box heater power relay

8.4.4.9 Generator cooling control system

1. Air duct louver actuators (solenoids)
2. Air duct louver actuator relays
3. Air duct louver position indicators (open-closed)

8.4.4.10 Main Shaft Position Lock Control System

1. Main shaft position lock actuator
2. Main shaft position lock indicator (on-off)
3. Main shaft position lock actuator relay

8.4.4.11 Power Actuation Control System

1. Main circuit breaker actuator
2. Auxiliary power circuit breaker actuator
3. Emergency power circuit breaker actuator

8.4.5 TOWER ASSEMBLY

8.4.5.1 Foundation Assembly

1. Concrete forms
2. Concrete reinforcement
3. Concrete
4. Anchor bolts
5. Control building
6. Security fence

8.4.5.2 Truss Structure Assembly

1. Main structural members (4)
2. Cross bracing members
3. Attachments (gussets, rivets, bolts, etc.)
4. Pintle access ladder
5. Tower platform/railings below pintle
6. Azimuth drive mounting structure/seals

8.4.5.3 Azimuth Drive Assembly

1. Azimuth Drive bearing/seals
2. Azimuth drive ring gear/weather seals
3. Azimuth drive gear box (incl. worm gear)
4. Azimuth drive shaft/bearings/slip clutch
5. Azimuth angle sensor
6. Azimuth angle lock pin (maintenance)

8.4.5.4 Pintle Structural Assembly

1. Pintle base structure
2. Pintle access hatch
3. Pintle structure/working platform
4. Pintle nacelle (latches, hinges, windows)
5. Cooling air inlet ducts/air filters/leuvers
6. Pintle service hoist
7. Nacelle compartment temp. sensor
8. Fire control system (extinguisher)

8.4.5.5 Instrumentation Boom Assembly

1. Boom
2. Wind direction sensor
3. Wind velocity sensor
4. Secondary aircraft beacon

APPENDIX 8.5

WTG-OPT COMPUTER CODE DESCRIPTION

This appendix provides additional definition of the WTG-OPT computer code described in Section 4.0. Section 8.5.1 describes the list of Inputs to this design optimization tool, while Section 8.5.2 describes the program Outputs. A complete presentation of the program structure, subroutines and equations is given in Section 8.5.3.

8.5.1 PROGRAM INPUT

The first series of cards make up the table information for the wind velocity profile, aerodynamic power coefficient, C_p, thrust coefficient, C_t, and velocity ratio, λ.

Table Input

The first card describes the input for the wind velocity profile input as shown below; a similar card precedes the data input for each of the 3 remaining tables.

Column	Variable Name	Format	
1-10	LVEC	I10	length of input vector (i.e. no. points in table) (must be .LE. 90)
20	IPRNT	I10	printing option IPRNT = 1 = print table = 0 = no printing
21-80	TITLES	30 A2	table description

Cards 2-11 provide the data for the selected wind duration curve in the 8F10.4 format. Data for the wind duration curve is input in meters/second for eighty six (86), 100-hour increments. Therefore, the first number would be the maximum wind velocity experienced at time = 0.

In the manner described above, Cards 12-15, 16-19 and 20-23 provide the data for power coefficient, thrust coefficient and velocity ratio curves. These curves consist of a total of 30 data points each, and the values for C_p and C_t must correspond to the data for λ. The data input begins with the value of C_p corresponding to a λ of 0.

Case Input

The next set of cards (cards 24 through 34) contains the data for the "first run".

Cards 24-26: These are Header Cards to identify the specific run. Format is 40A2. If the first 4 colums of a Header Card contains the word STOP, the program will call EXIT.

Card 27: Contains 4 values which refer to base case calculation in the 3F10.4, I5 format

PR	=	rated power
LAMDAR	=	rated velcotiy ratio
NR	=	rotor RPM
NITER	=	maximum number of iterations permitted in diameter calculation loop

Card 28: Describes Wind Regime Data, 5 inputs in format 5F10.4

VMEAN	=	mean (median) wind speed on input curve Table 1 (i.e. wind speed @ 4380 Hrs.) (meters/sec)
HO	=	reference height for wind curve calculation (meters)
CLRNCE	=	rotor blade ground clearance (meters)
TERFAC	=	terrain factor
RHO	=	mass density (Kg/M^3) of air

Card 29: Upper and Lower Limits of Independent Variables, 6 inputs in format 6F10.4

PRMAX	=	upper limit on rated power
LMDRMX	=	" " " velocity ratio @ rated power
NRMAX	=	" " " rotor RPM
PRMIN	=	lower limit on rated power
LMDCMN	=	" " " velocity ratio @ rated power
NRMIN	=	" " " rotor RPM

Card 30: Constants for Input to Optimization Routine, 6 inputs in format 3I10, 3F10.4

NOPT	=	no. of independent variables in model
KM	=	max. no. of base cases to be generated
IVCON	=	1 hold PR const & optimize on LAMDAR & NR
	=	2 " LAMDAR " " " PR & NR
	=	3 " NR " " " PR & LAMDAR

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= 4 hold all 3 independent variables constant & determine cost @ this pt. (no optimization)

= 0 optimize on all 3 independent variables

TOL = percent improvement of objective function between base cases Sprint stops

DELX = determines perturbation size for calculation of partial derivatives (10^{-4})

DELY = determines initial step size along the steepest ascent vector (.05)

Card 31: Transmission and Generator Constants 2 input in Format 2F10.4

NG = generator RPM

Card 32: Constants for Investment Analyses, 2 inputs in format 2I10

LIFE = life of investment (yrs)

IDEPM = 1 straight line depreciation

= 2 sum of years digits depreciation

= 3 double declining balance depreciation

Card 33: Constants for Investment Analyses, 7 inputs in format 7F10.4

NU = no. of units

DEBT = debt fraction in capital structures

EQTY = equity fraction in capital structures

DINT = average interest on debt

EINT = return on equity

TAX = corporate income tax rate

TXCR = investment tax credit

Card 34: Tower Option Card, 1 input in format I10

IOPT = 1 truss tower option

= 2 concrete tower option

8.5.2 PROGRAM OUTPUT

The first set of output provides a listing of the wind velocity profile and power coefficient and thrust coefficient values versus velocity ratio if requested by the IPRNT input option.

The second set of output repeats the input listing so that the values can be checked for accuracy.

The third set of output lists the value of the payoff function (¢/KW-HR), the value of the constraint and the optimized values of the independent variables for each base case. When an optimum has been successfully determined, a message is printed out followed by the final values of the optimum objective, payoff and constraint functions.

If an optimum has not been found after going through the specific maximum of base cases (KM), a message is printed followed by the final (not optimum) objective, payoff and constraint functions. Assuming an optimized case has been found, the performance characteristics, weight breakdown and cost information is presented.

The performance characteristics are given in the folloiwng order.

PRATED	=	rated power, KW
LAMDAR	=	velocity ratio at rated power
NROTOR	=	rotor RPM
DIAMETER	=	rotor diameter, M
VRATED	=	rated velocity, m/sec
VCO	=	cut-out velocity, m/sec
VCI	=	cut-in velocity, m/sec
PMIN	=	minimum power for generator cut-in, KW
ENERGY	=	electrical energy generated, KW-HRS
PDENSITY	=	power density, KW/m^2
VR/VM	=	ratio of rated wind velocity to mean wind velocity
VW/VO	=	ratio of wind velocity at rotor axis to wind velocity at reference
LSTITD	=	no. of iterations gone through for power convergence the last time through the diameter iteration loop
TR	=	time at rated velocity, HRS

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TCO	=	time at cut-out velocity, HRS
TCI	=	time at cut-in velocity, HRS
CPMIN	=	power coefficient for PMIN
EWIND	=	total energy available from the wind, KW-HRS
ETASYS	=	system efficiency; ratio of electrical energy generated to total energy available from the wind stream
CPPR	=	power coefficient at rated power
CTPR	=	thrust coefficient at rated power
RTRTHRUST	=	wind force on tower, newtons
QGENRTR	=	generator torque, netwon-m
QROTOR	=	rotor torque, newton-m
LSTITP	=	no. of iterations gone through for minimum power convergence the last time through the PMIN iteration loop
VTIP	=	rotor tip speed, m/sec

Following the performance characteristics, fabrication and delivery costs for each of the major subsystems are given as well as the estimated installation and systems engineering cost. The result is the total capital cost for the system.

<u>Fabrication Costs</u>

Tower

Rotor

Generator (Electrical Subsystem)

Transmission

Controls

Total Fabrication Cost

Delivery Costs

Tower

Rotor

Generator (Electrical Subsystem)

Transmission

Controls

Total Delivery Cost

Installation

Total System Installation Cost

Total Capital Cost

The next set of output relates to the subsystem weight.

Tower, kgs

Rotor, kgs

Generator, kgs

Transmission, kgs

Controls, kgs

Total System Weight, kgs

Next, a cost and weight breakdown of the tower, generator and transmission is given.

Tower Pintle Cost

Tower Structure Cost

Pintle Weight

Structure Weight

Generator Cost

Electrical Accessory Cost

Generator Weight

Electrical Accessory Weight

Transmission Gear Box Cost

Brake Cost

Clutch Cost

Main Shaft Cost

Main Bearings Cost

Flexible Couplings Cost

Gear Box Weight

Transmission Accessory Weight

Next, subsystem maintenance costs are given.

Tower

Rotor

Generator

Transmission

Controls

Total

Following the maintenance costs is a summary of the annual WTG System performance which relates to the specific wind duration curve assumed. The information listed below is given in 100 hour time increments.

TIME	=	point on wind curve (hours)
WIND O	=	wind velocity profile at reference
WIND	=	wind velocity profile at altitude
CP	=	power coefficient
LAMBDA	=	velocity ratio
PE	=	electrical power generated
PW	=	power available in wind stream
PLOAD	=	percent load on generator
ETAG	=	generator efficiency
ETAT	=	transmission efficiency

The final set of output is a yearly investment analysis over the depreciated life of the WTG. These outputs are as follows:

Capital investment

Start-up costs (non-capitalized)

Annual costs (maintenance)

Debt fraction

Equity fraction

Life of investment

Corporate income tax rate

Tax credit

Debt interest

Equity return

Method of depreciation being used

The yearly information is listed below:

YR	=	year of life
CASH REQ.	=	recurring capital costs
DEPR.	=	depreciation
GROSS	=	gross income
TAXES	=	income taxes
NET	=	net income
CAPFD	=	sinking fund

8.5.3 PROGRAM STRUCTURE

The following pages describe the structure and logic flow for each of the subroutines in the WTG-OPT code.

```
CSPRINT

DIMENSION YX(3),XO(3),Z(3),FX(3),GX(3),DX(3)

COMMON C(6,6),WT(6),MILPKW,ANNREV,NU,WTOTHR(10),COTHR(15),
       DEP(50),GS(50),TX(50),XNET(50),SF(50),CR(50),CAP,CINS,
       LIFE,DEBT,EQTY,DINT,EINT,TAX,TXCR,IDEPM,XFIX

COMMON NITER,D,VRDVM,CPPR,ETAT,NG,VR,QR,GR,QG,ETAGO,HR,CLRNCE,
       TERFAC,HO,RHO,VMEAN,VTIP,NERROR,PD,LSTITD

COMMON VW(90),LAMBDA(90),CP(90),ETAGEN(90),PLOAD(90),PE(90),
       CPMIN,PGMIN,
       VCI,VCO,VWDVO,TR,TCO,E,IVCO,IVCI,TCI,LSTITP,EW,ETASYS,
       PW(90),ETATR(90)

COMMON VO(90),CP1(30),CT1(30),LAMDA1(30),LVW,LCP1,LCT1,LLAM1

COMMON VAL(3),VALMAX(3),VALMIN(3),
       N,KM,K,TOL,DELX,DELY,X(3),Y(4),F(4),G(4),IVCON

COMMON HT,CTPR,TRTR,IOPT
REAL MILPKW,NU
REAL NG
REAL LAMBDA
REAL LAMDA1

C  READ INPUT TO SUBROUTINE MODEL

CALL IO(1)

A1  2   13
    13

5
CALL IO(4)

GG = 1.0
K = 1

CALL SCALE(1)

CALL MODEL (F(1),G(1))

IF (NERROR.NE.0) GO TO 5
IF (IVCON.NE.4) GO TO 8
Y(1)=0.0
```

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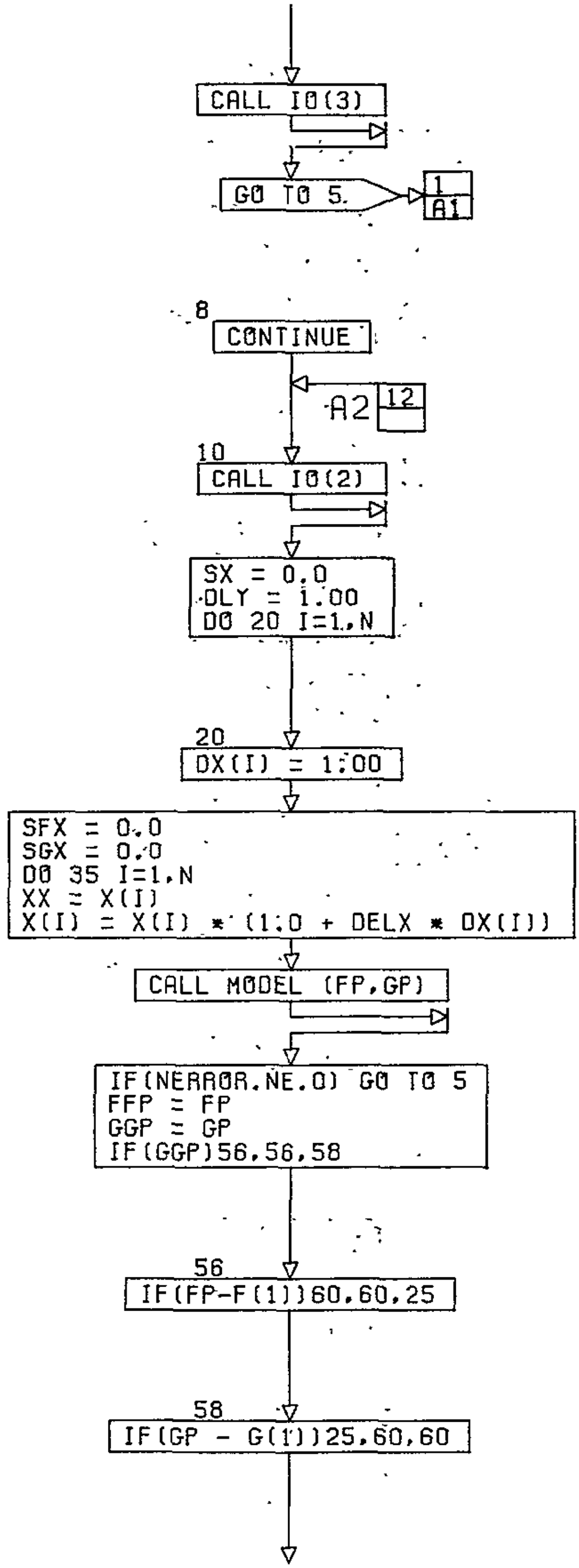
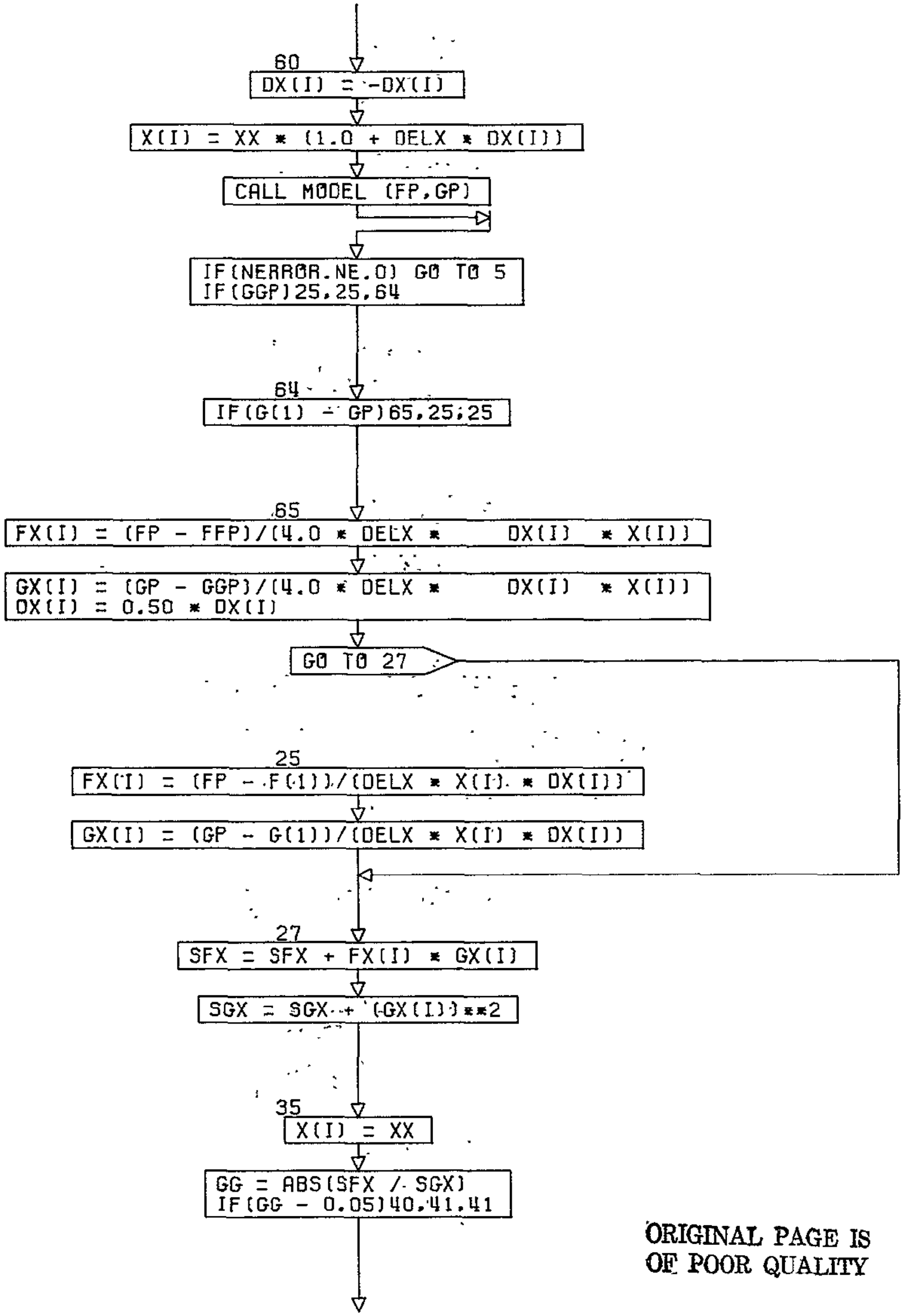
CONT. ON PG 2


CALL IO(3)
GO TO 5.
1
A1
8
CONTINUE
A2
12
10
CALL IO(2)
SX = 0.0
DLY = 1.00
DO 20 I=1,N
20
DX(I) = 1.00
SFX = 0.0
SGX = 0.0
DO 35 I=1,N
XX = X(I)
X(I) = X(I) * (1.0 + DELX * DX(I))
CALL MODEL (FP,GP)
IF(NERROR.NE.0) GO TO 5
FFP = FP
GGP = GP
IF(GGP)56,56,58
56
IF(FP-F(1))60,60,25
58
IF(GP - G(1))25,60,60

CONT. ON PG 3



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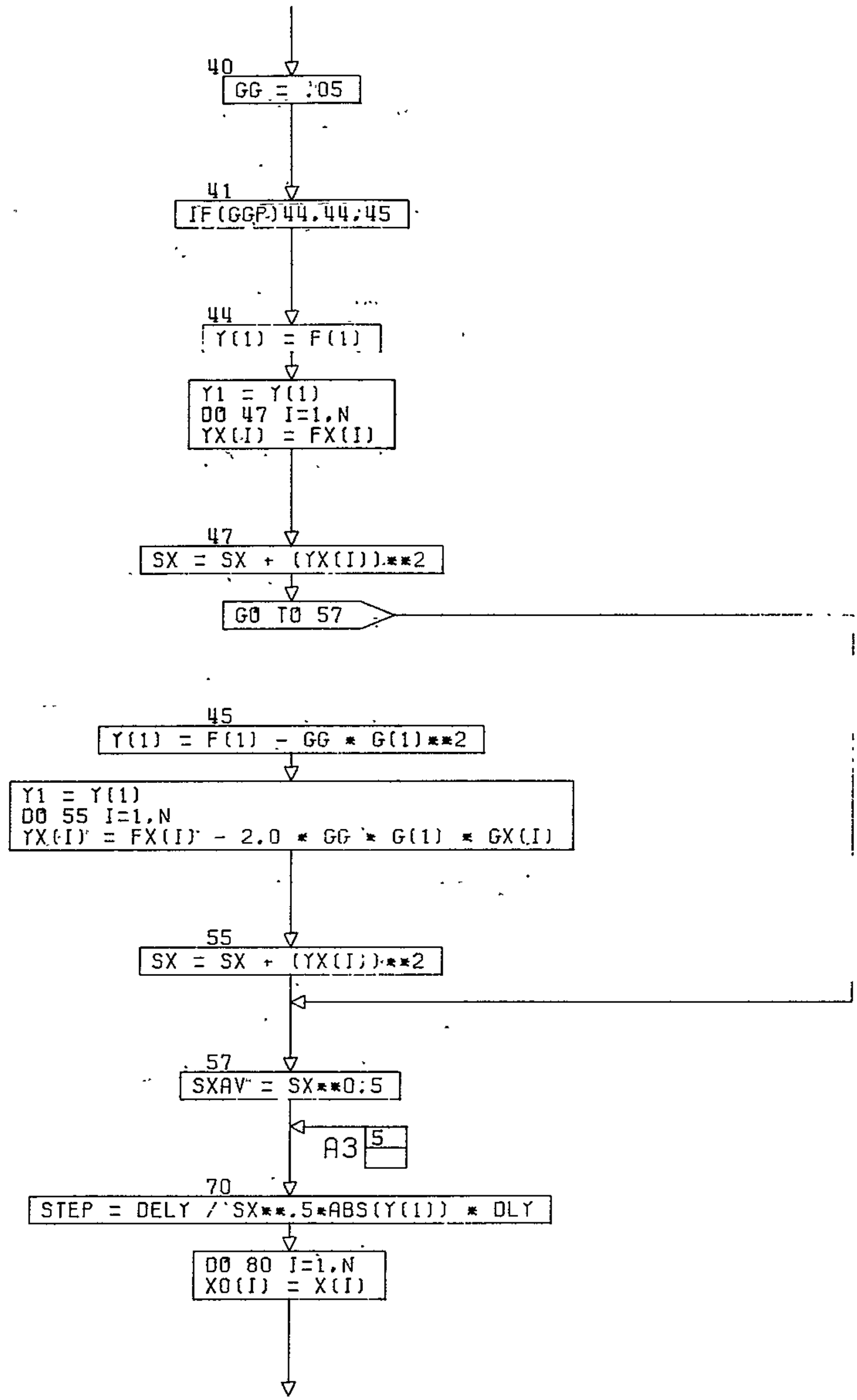
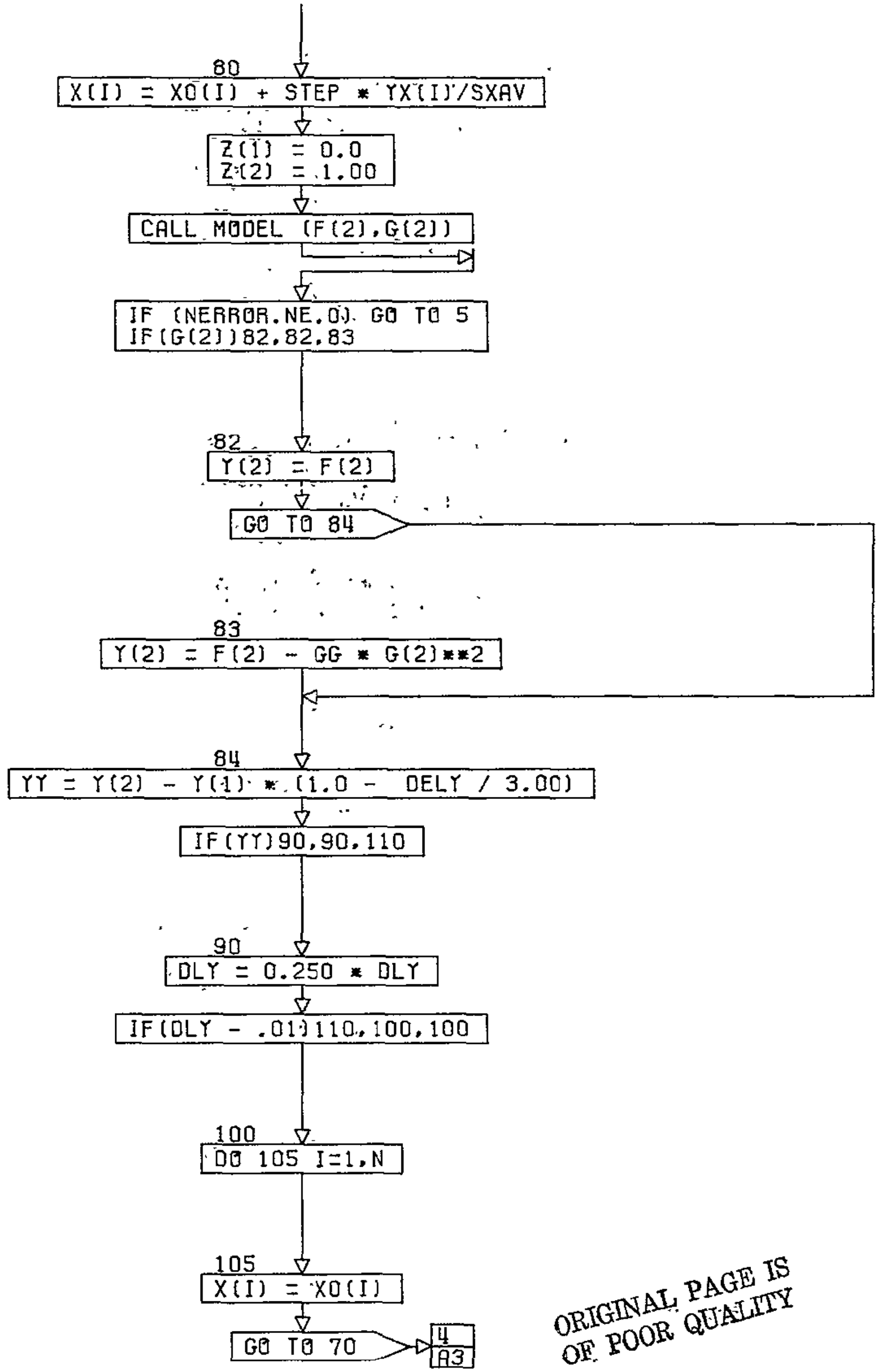
CONT. ON PG 4

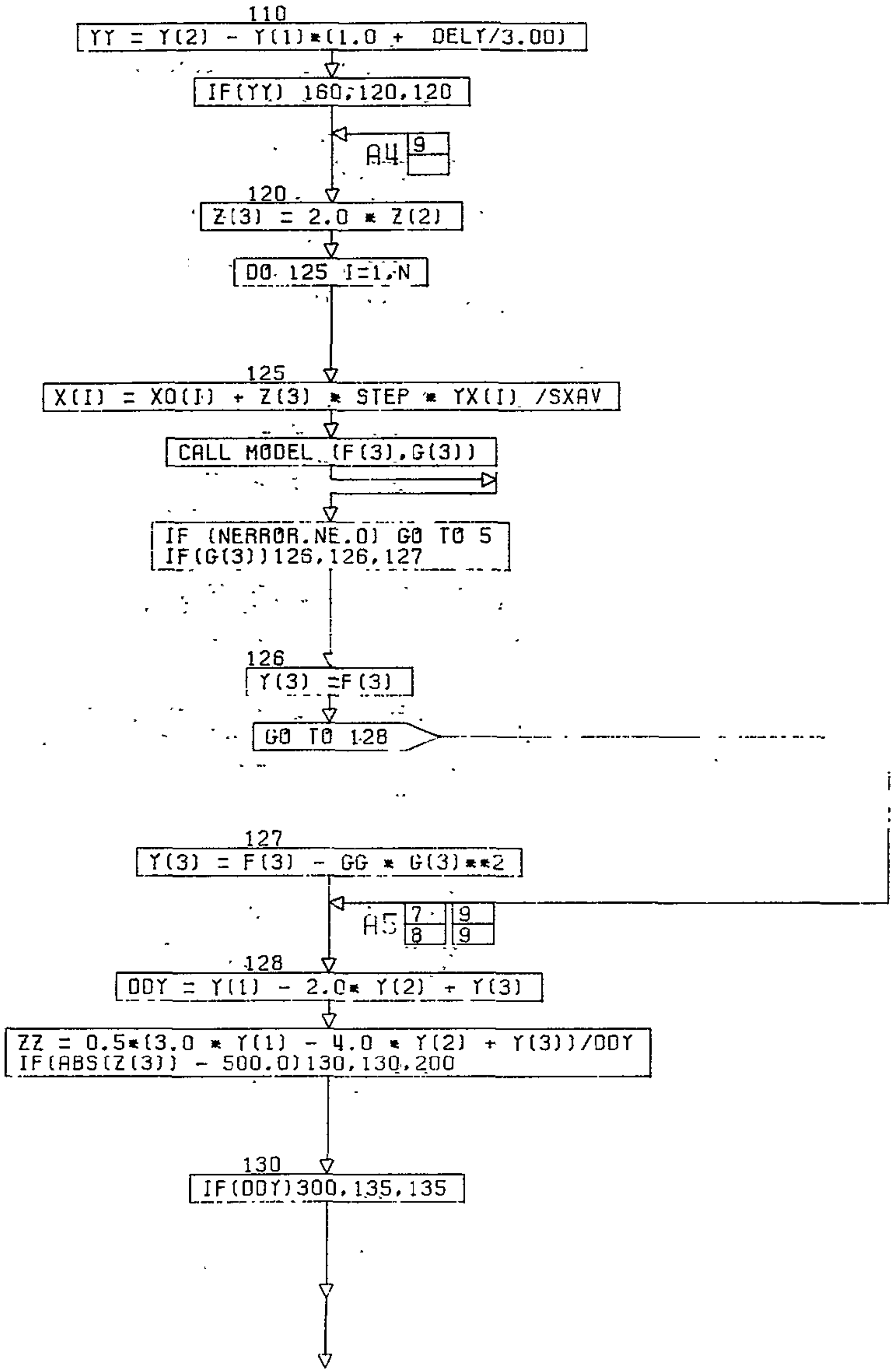


40
GG = .05
41
IF(GGP)44,44,45
44
Y(1) = F(1)
Y1 = Y(1)
DO 47 I=1,N
YX(I) = FX(I)
47
SX = SX + (YX(I))**2
GO TO 57
45
Y(1) = F(1) - GG * G(1)**2
Y1 = Y(1)
DO 55 I=1,N
YX(I) = FX(I) - 2.0 * GG * G(1) * GX(I)
55
SX = SX + (YX(I))**2
57
SXAV = SX**0.5
A3 5
70
STEP = DELY / SX**.5*ABS(Y(1)) * DLY
DO 80 I=1,N
XO(I) = X(I)

CONT. ON PG 5

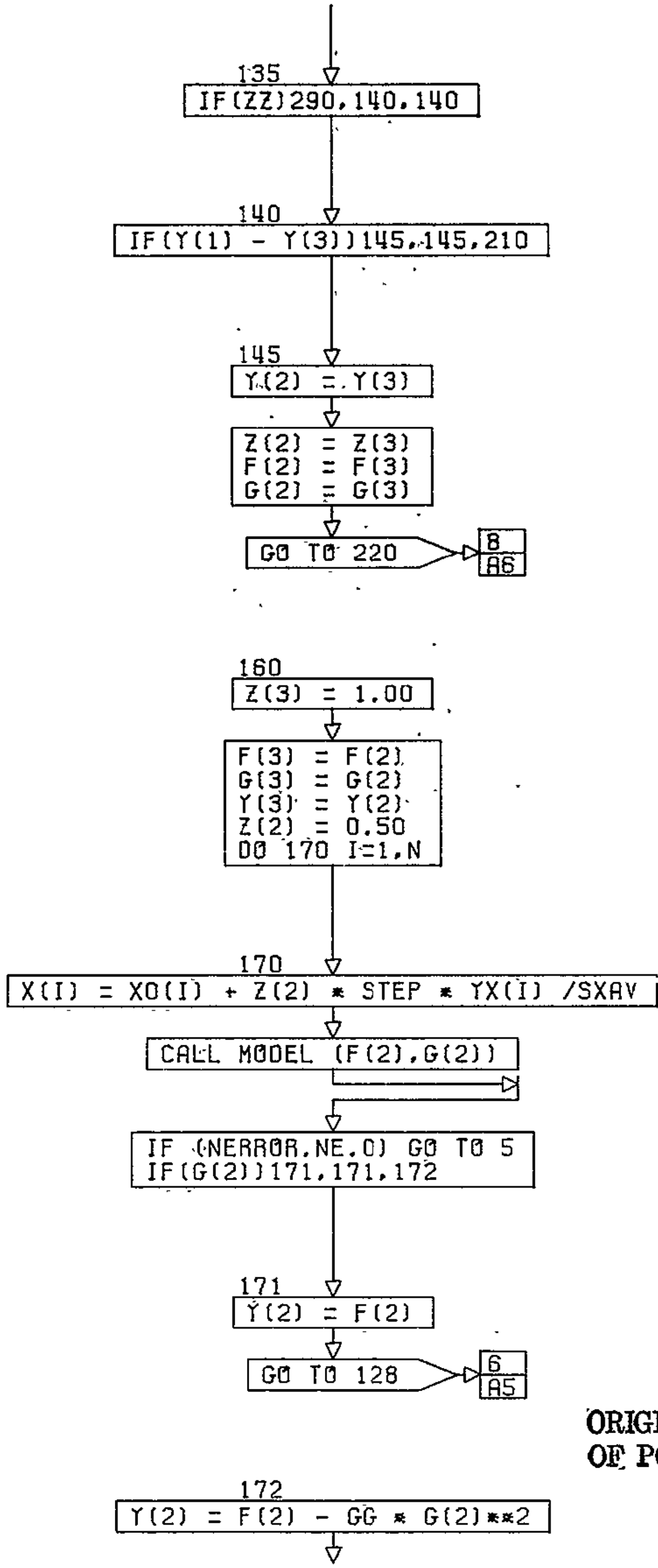

80
X(I) = XO(I) + STEP * YX(I)/SXAV
Z(1) = 0.0
Z(2) = 1.00
CALL MODEL (F(2),G(2))
IF (NERROR.NE.0) GO TO 5
IF(G(2))82,82,83
82
Y(2) = F(2)
GO TO 84
83
Y(2) = F(2) - GG * G(2)**2
84
YY = Y(2) - Y(1) * (1.0 - DELY / 3.00)
IF(YY)90,90,110
90
DLY = 0.250 * DLY
IF(DLY - .01)110,100,100
100
DO 105 I=1,N
105
X(I) = XO(I)
GO TO 70
4
A3

CONT. ON PG 6

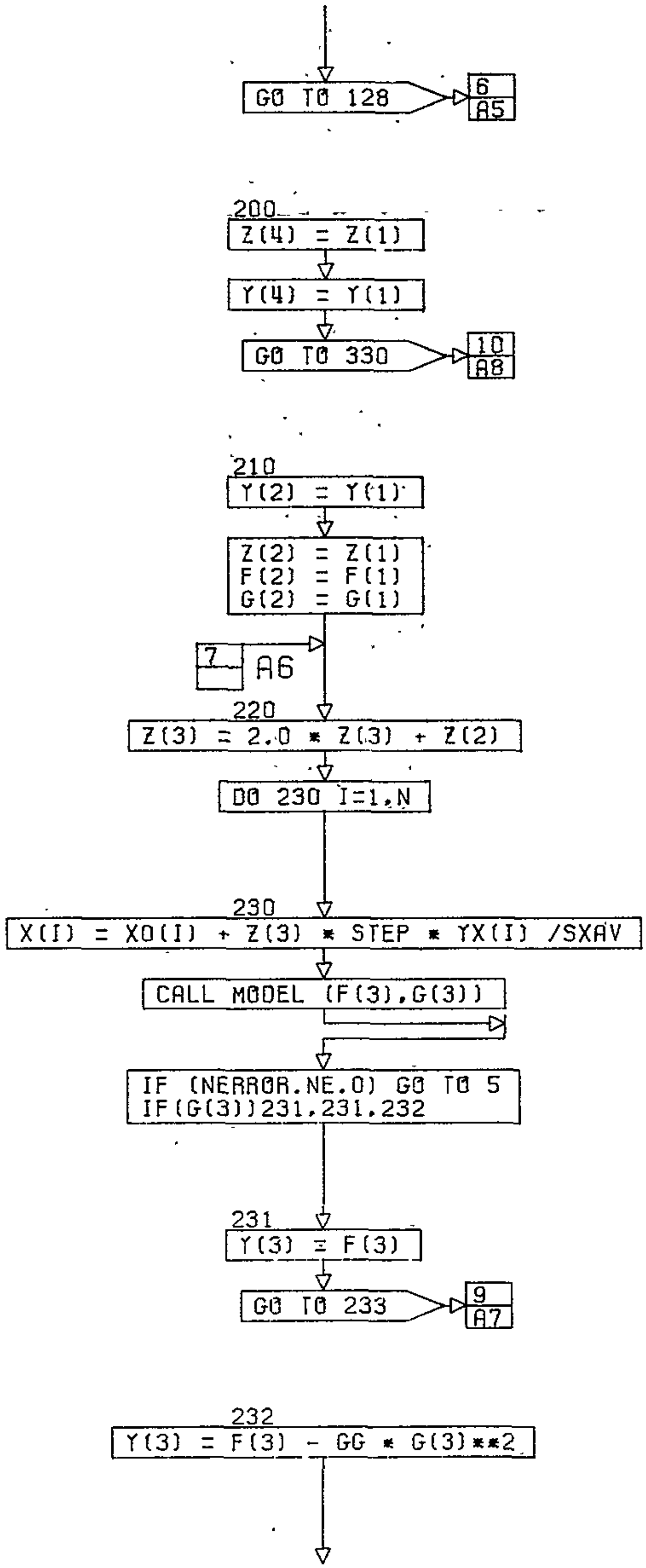


110
YY = Y(2) - Y(1)*(1.0 + DELY/3.00)
IF(YY) 160,120,120
A4 9
120
Z(3) = 2.0 * Z(2)
DO 125 I=1,N
125
X(I) = XO(I) + Z(3) * STEP * YX(I) /SXAV
CALL MODEL (F(3),G(3))
IF (NERROR.NE.0) GO TO 5
IF(G(3))126,126,127
126
Y(3) =F(3)
GO TO 128
127
Y(3) = F(3) - GG * G(3)**2
A5 7 9
8 9
128
DDY = Y(1) - 2.0* Y(2) + Y(3)
ZZ = 0.5*(3.0 * Y(1) - 4.0 * Y(2) + Y(3))/DDY
IF(ABS(Z(3)) - 500.0)130,130,200
130
IF(DDY)300,135,135

CONT. ON PG 7


135
IF(ZZ)290,140,140
140
IF(Y(1) - Y(3))145,145,210
145
Y(2) = Y(3)
Z(2) = Z(3)
F(2) = F(3)
G(2) = G(3)
GO TO 220
8
A6
160
Z(3) = 1.00
F(3) = F(2)
G(3) = G(2)
Y(3) = Y(2)
Z(2) = 0.50
DO 170 I=1,N
170
X(I) = XO(I) + Z(2) * STEP * YX(I) /SXAV
CALL MODEL (F(2),G(2))
IF (NERROR.NE.0) GO TO 5
IF(G(2))171,171,172
171
Y(2) = F(2)
GO TO 128
6
A5
172
Y(2) = F(2) - GG * G(2)**2

CONT. ON PG 8


GO TO 128
6
A5
200
Z(4) = Z(1)
Y(4) = Y(1)
GO TO 330
10
A8
210
Y(2) = Y(1)
Z(2) = Z(1)
F(2) = F(1)
G(2) = G(1)
7
A6
220
Z(3) = 2.0 * Z(3) + Z(2)
DO 230 I=1,N
230
X(I) = XO(I) + Z(3) * STEP * YX(I) /SXAV
CALL MODEL (F(3),G(3))
IF (NERROR.NE.0) GO TO 5
IF(G(3))231,231,232
231
Y(3) = F(3)
GO TO 233
9
A7
232
Y(3) = F(3) - GG * G(3)**2

CONT. ON PG 9

```
8
-   A7

233
Z(1) = 2.0 * Z(2) - Z(3)

DO 240 I=1,N

240
X(I) = XO(I) + Z(1) * STEP * YX(I) /SXAV

IF(G(1))241,241,242

241
Y(1) = F(1)

GO TO 128        6
                 A5

242
Y(1) = F(1) - GG * G(1)**2

GO TO 128        6
                 A5

290
Z(2) = Z(3)

Y(2) = Y(3)
F(2) = F(3)
G(2) = G(3)

GO TO 120        6
                 A4

300
IF(ZZ + 0.50)305,310,310

305
ZZ = -0.50
```

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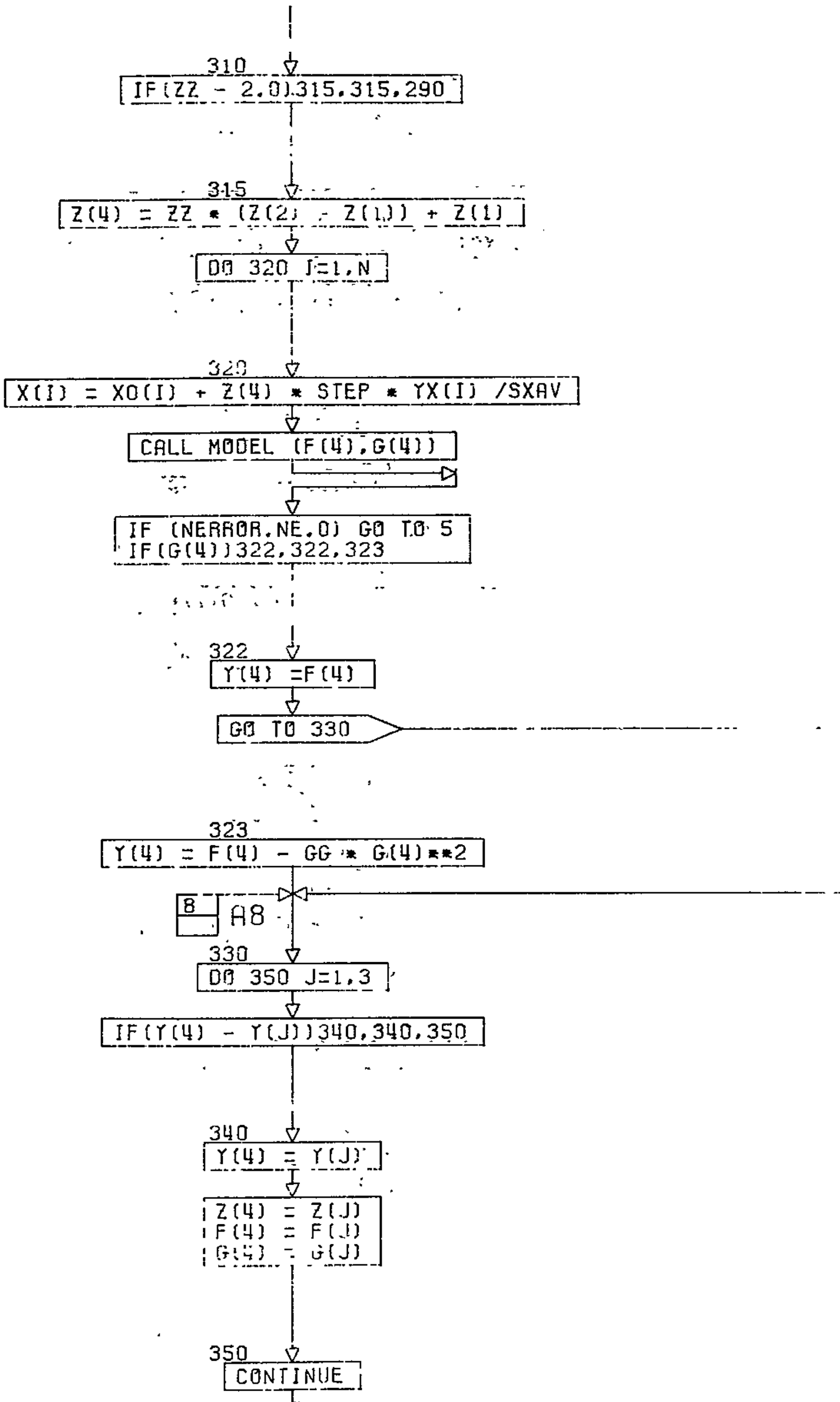
CONT. ON PG 10


310
IF(ZZ - 2.0)315,315,290
315
Z(4) = ZZ * (Z(2) - Z(1)) + Z(1)
DO 320 I=1,N
320
X(I) = XO(I) + Z(4) * STEP * YX(I) /SXAV
CALL MODEL (F(4),G(4))
IF (NERROR.NE.0) GO TO 5
IF(G(4))322,322,323
322
Y(4) =F(4)
GO TO 330
323
Y(4) = F(4) - GG * G(4)**2
B
A8
330
DO 350 J=1,3
IF(Y(4) - Y(J))340,340,350
340
Y(4) = Y(J)
Z(4) = Z(J)
F(4) = F(J)
G(4) = G(J)
350
CONTINUE

CONT. ON PG 11

```
IF(Z(4) - Z(3))390,360,360

360
Y(1) = Y(2)

Y(2) = Y(3)
F(1) = F(2)
G(1) = G(2)
F(2) = F(3)
G(2) = G(3)
Z(3) = 2.0 * Z(3) - Z(2)
Z(1) = Z(2)
Z(2) = 0.50 * (Z(3) + Z(1))

DO 370 I=1,N

370
X(I) = XO(I) + Z(3) * STEP * YX(I) /SXAV

CALL MODEL (F(3),G(3))

IF (NERROR.NE.0) GO TO 5
IF(G(3))371,371,372

371
Y(3) = F(3)

GO TO 373   12
            A9

372
Y(3) = F(3) - GG * G(3)**2

11   A9

373
IF(ABS(Z(3)) -.500.0)128,390,390

390
YYY = (Y(4) - Y1)/Y1
```

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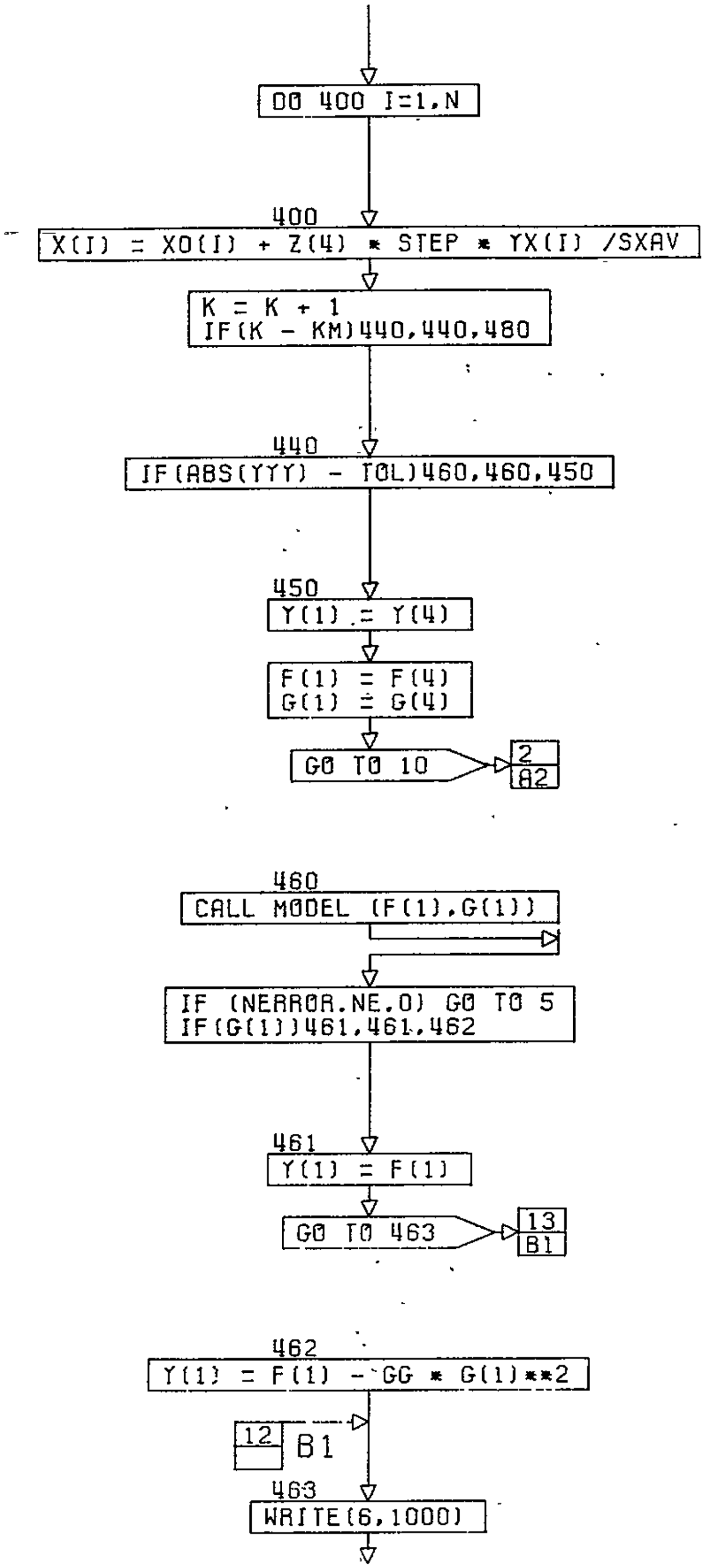
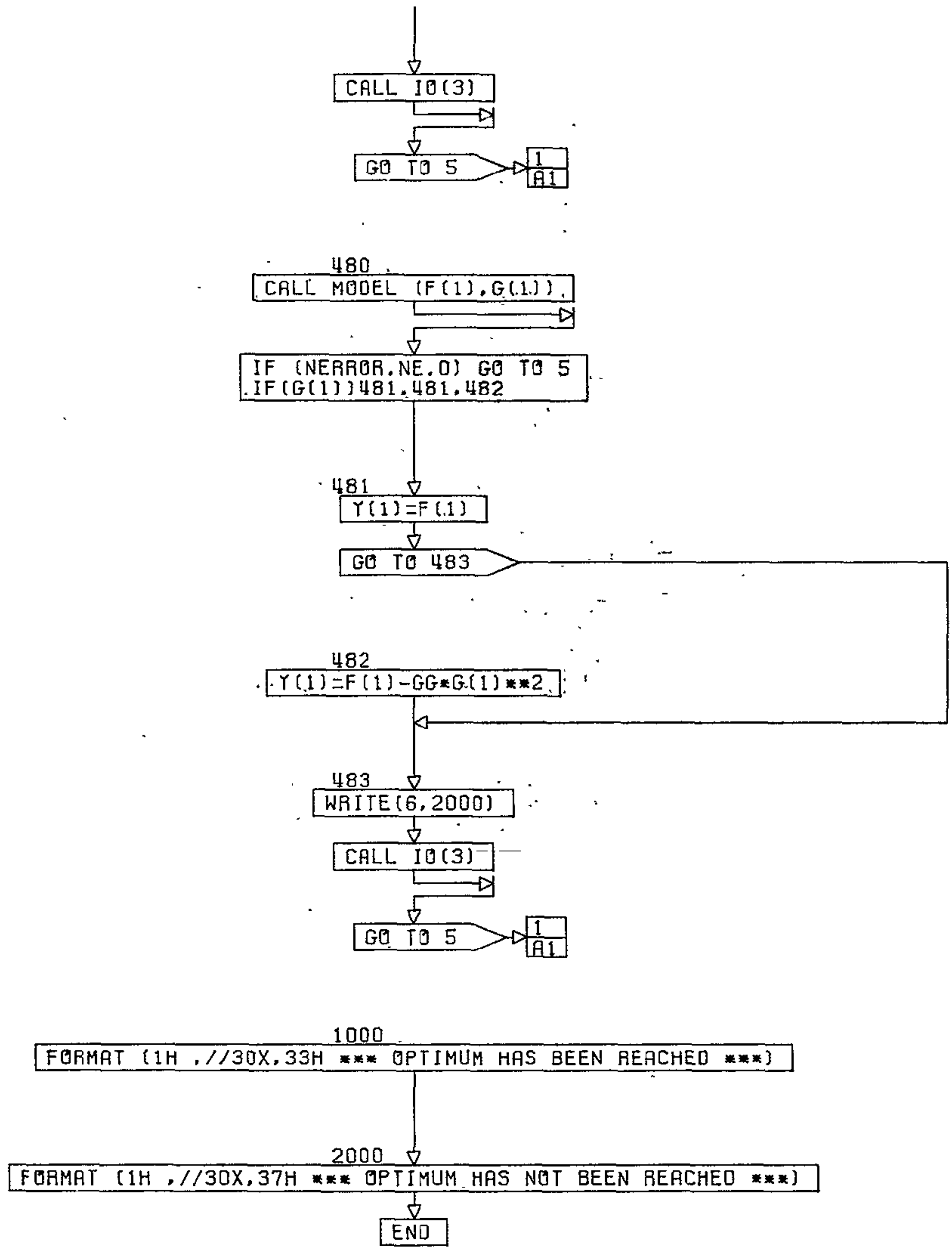
CONT. ON PG 12


DO 400 I=1,N
400
X(I) = XO(I) + Z(4) * STEP * YX(I) /SXAV
K = K + 1
IF(K - KM)440,440,480
440
IF(ABS(YYY) - TOL)460,460,450
450
Y(1) = Y(4)
F(1) = F(4)
G(1) = G(4)
GO TO 10
2
A2
460
CALL MODEL (F(1),G(1))
IF (NERROR.NE.0) GO TO 5
IF(G(1))461,461,462
461
Y(1) = F(1)
GO TO 463
13
B1
462
Y(1) = F(1) - GG * G(1)**2
12
B1
463
WRITE(6,1000)

CONT. ON PG 13


CALL IO(3)
GO TO 5
1
A1
480
CALL MODEL (F(1),G(1))
IF (NERROR.NE.0) GO TO 5
IF(G(1))481,481,482
481
Y(1)=F(1)
GO TO 483
482
Y(1)=F(1)-GG*G(1)**2
483
WRITE(6,2000)
CALL IO(3)
GO TO 5
1
A1
1000
FORMAT (1H ,//30X,33H *** OPTIMUM HAS BEEN REACHED ***)
2000
FORMAT (1H ,//30X,37H *** OPTIMUM HAS NOT BEEN REACHED ***)
END

C

SUBROUTINE MODEL (PAYOFF,CONSTR)

```
COMMON C(6,6),WT(6),MILPKW,ANNREV,NU,WTOTHR(10),COTHR(15),
       DEP(50),GS(50),TX(50),XNET(50),SF(50),CR(50),CAP,CINS,
       LIFE,DEBT,EQTY,DINT,EINT,TAX,TXCR,IDEPM,XFIX
```

```
COMMON NITER,D,VRDVM,CPPR,ETAT,NG,VR,QR,GR,QG,ETAGO,HR,CLRNCE,
       TERFAC,HO,RHO,VMEAN,VTIP,NERROR,PD,LSTITD
```

```
COMMON VW(90),LAMBDA(90),CP(90),ETAGEN(90),PLOAD(90),PE(90),
       CPMIN,PGMIN,
       VCI,VCO,VWDVO,TR,TCO,E,IVCO,IVCI,TCI,LSTITP,EW,ETASYS,
        PW(90),ETATR(90)
```

```
COMMON VO(90),CP1(30),CT1(30),LAMDA1(30),LVW,LCP1,LCT1,LLAM1
COMMON PR,LAMDAR,NR,PRMAX,LMDRMX,NRMAX,PRMIN,LMDRMN,NRMIN,
```

```
COMMON HT,CTPR,TRTR,IOPT
       N,KM,K,TOL,DELX,DELY,X(3),Y(4),F(4),G(4),IVCON
```

```
REAL MILPKW,NU
REAL NG
REAL LAMBDA
REAL LAMDA1
REAL LAMDAR,LMDRMX,LMDRMN,NR,NRMAX,NRMIN
```

```
C  ANALYTICAL MODEL OF WIND GENERATOR COST EFFECTIVENESS
C  SPRINT REQUIRES THAT...
C     INPUT BE READ INTO COMMON OUTSIDE OF SUBROUTINE MODEL
C     OUTPUT BE PRINTED OUTSIDE OF SUBROUTINE MODEL
C     THE FIRST EXECUTABLE STATEMENT OF SUBROUTINE MODEL BE    CALL SCALE
```

CALL SCALE(2)

NERROR=0

C CR IS RECURRING COST

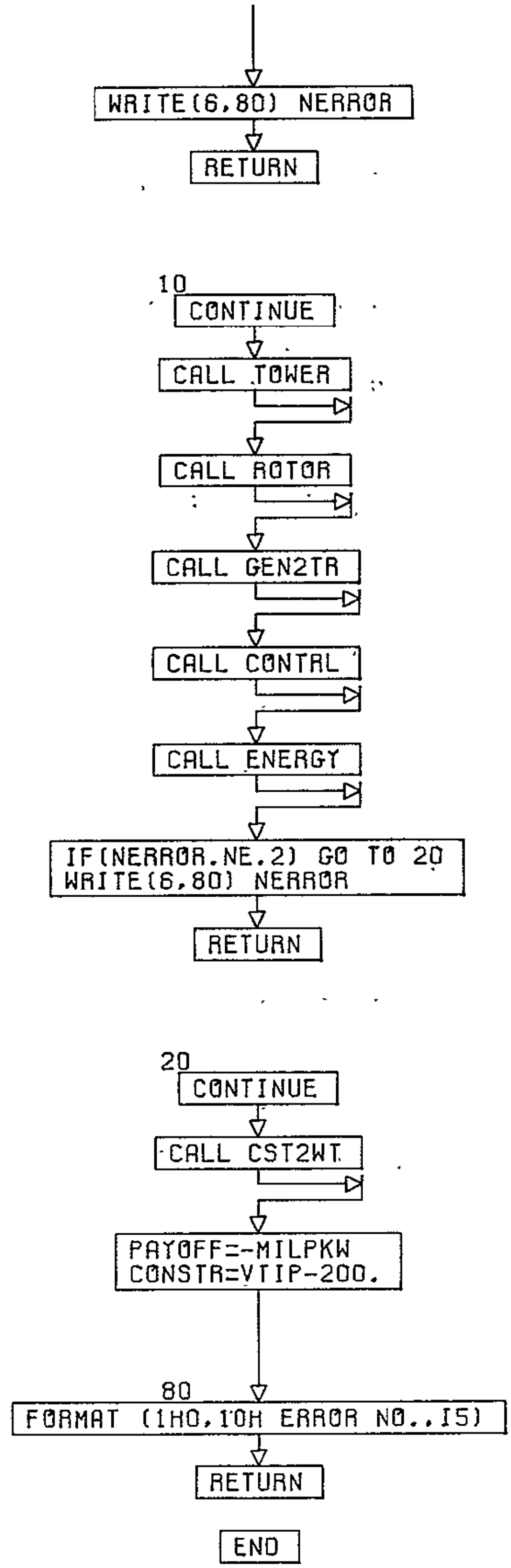
DO 5 I=1,LIFE

5 CR(I)=0.0

CALL DITER

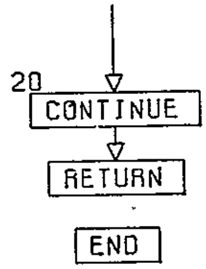
IF(NERROR.NE.1) GO TO 10

CONT. ON PG 2


WRITE(6,80) NERROR
RETURN
10
CONTINUE
CALL TOWER
CALL ROTOR
CALL GEN2TR
CALL CONTRL
CALL ENERGY
IF(NERROR.NE.2) GO TO 20
WRITE(6,80) NERROR
RETURN
20
CONTINUE
CALL CST2WT
PAYOFF=-MILPKW
CONSTR=VTIP-200.
80
FORMAT (1H0,10H ERROR NO.,I5)
RETURN
END

```
C

SUBROUTINE SCALE(J)

COMMON E(6,6),WT(6),MILPKW,ANNREV,NU,WTOTHR(10),COTHR(15),
       DEP(50),GS(50),TX(50),XNET(50),SF(50),CR(50),CAP,CINS,
       LIFE,DEBT,EQTY,DINT,EINT,TAX,TXCR,IDEPM,XFIX

COMMON NITER,D,VRDVM,CPPR,ETAT,NG,VR,QR,GR,QG,ETAGO,HR,CLRNCE,
       TERFAC,HO,RHO,VMEAN,VTIP,NERROR,PD,LSTITO

COMMON VW(90),LAMBDA(90),CP(90),ETAGEN(90),PLOAD(90),PE(90),
       CPMIN,PGMIN,
       VCI,VCO,VWDVO,TR,TCO,E,IVCO,IVCI,TCI,LSTITP,EW,ETASYS,
        PW(90),ETATR(90)

COMMON VO(90),CP1(30),CT1(30),LAMDA1(30),LVW,LCP1,LCT1,LLAM1

COMMON VAL(3),VALMAX(3),VALMIN(3),
       NOPT,KM,K,TOL,DELX,DELY,CHANGE(3),Y(4),F(4),G(4),IVCON

COMMON HT,CTPR,TRTR,IOPT
REAL MILPKW,NU
REAL NG
REAL LAMDA1
REAL LAMBDA
GO TO(100,200),J

100
DO 10 I=1,NOPT

10
CHANGE(I)=(VAL(I)-VALMIN(I))/(VALMAX(I)-VALMIN(I))   +1.

RETURN

200
DO 20 I=1,NOPT

IF (I.EQ.IVCON) GO TO 20
IF(CHANGE(I) .LT. 1.0)CHANGE(I) = 1.00
IF(CHANGE(I) .GT. 2.0)CHANGE(I) = 2.00
VAL(I)=(CHANGE(I)-1.)*(VALMAX(I)-VALMIN(I)) +VALMIN(I)
```

CONT. ON PG 2


20
CONTINUE
RETURN
END

```
C

SUBROUTINE IO(INOUT)
DIMENSION IQUIT(2)

COMMON C(6,6),WT(6),MILPKW,SALES,NU,WTOTHR(10),COTHR(15),
       DEP(50),GS(50),TX(50),XNET(50),SF(50),CR(50),CAP,CINS,
       LIFE,DEBT,EQTY,DINT,EINT,TAX,TXCR,IDEPM,XFIX

COMMON NITER,D,VRDVM,CPPR,ETAT,NG,VR,QR,GR,QG,ETAGO,HR,CLRNCE,
       TERFAC,HO,RHO,VMEAN,VTIP,NERROR,PD,LSTITD

COMMON VW(90),LAMBDA(90),CP(90),ETAGEN(90),PLOAD(90),PE(90),
       CPMIN,PMIN,
       VCI,VCO,VWDVO,TR,TCO,E,IVCO,IVCI,TCI,LSTITP,EW,ETASYS,
       PW(90),ETATR(90)

COMMON VO(90),CP1(30),CT1(30),LAMDA1(30),LVW,LCP1,LCT1,LLAM1

COMMON PR,LAMDAR,NR,PRMAX,LMDRMX,NRMAX,PRMIN,LMDRMN,NRMIN,
       N,KM,K,TOL,DELX,DELY,X(3),Y(4),F(4),G(4),IVCON

COMMON HT,CTPR,TRTR,IOPT
REAL MILPKW,NU
REAL NG
REAL LAMBDA
REAL LAMDA1
REAL LAMDAR,NR,LMDRMX,NRMAX,LMDRMN,NRMIN
INTEGER HEADER(40)
DATA IQUIT(1),IQUIT(2)/2HST,2HOP/

DATA KDVW,KDCP1,KDCT1,KDLAM1/90,30,30,30/
GO TO (1,800,100,05),INOUT

C  READ WIND VELOCITY DURATION CURVE

1  CALL READ (VO,KDVW,LVW)

C  READ POWER COEFFICIENT CURVE

CALL READ (CP1,KDCP1,LCP1)

C  READ  THRUST COEFFICIENT CURVE

CALL READ (CT1,KDCT1,LCT1)
```

CONT. ON PG 2

```
C  READ  VELOCITY RATIOS FOR VALUES ON CP AND CT CURVES
CALL READ (LAMDA1,KOLAM1,LLAM1)
RETURN
C  READ HEADER CARDS
5   DO 25 J=1,3
    READ (5,10) (HEADER(I),I=1,40)
10  FORMAT (40A2)
    WRITE (6,15) (HEADER(I),I=1,40)
15  FORMAT (1H ,40A2)
    IF (HEADER(1).EQ.IQUIT(1).AND.HEADER(2).EQ.IQUIT(2)) STOP
25  CONTINUE
C  READ BASE CASE VALUES OF VARIABLES
    READ (5,30) PR,LAMDAR,NR,NITER
30  FORMAT (3F10.4,I10)
    WRITE (6,35) PR,LAMDAR,NR,NITER
35  FORMAT (1H0,16H BASE CASE INPUT/3X,7H PRATED,3X,7H LAMDAR,7X,
            3H NR4X6H NITER/3F10.4,I10)
```

CONT. ON PG 3

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```
C  READ WIND CONSTANTS

READ (5,40) VMEAN,HO,CLRNCE,TERFAC,RHO

40
FORMAT (5F10.4)

WRITE (6,45) VMEAN,HO,CLRNCE,TERFAC,RHO

45
FORMAT (1H0,15H WIND CONSTANTS/4X,6H VMEAN,7X,3H HO,3X,
     7H CLRNCE,3X,7H TERFAC,6X,4H RHO/5F10.4)

C  READ MAXIMUM AND MINIMUM VALUES OF INDEPENDENT VARIABLES

READ (5,50) PRMAX,LMORMX,NRMAX,PRMIN,LMORMN,NRMIN

50
FORMAT (6F10.4)

WRITE (6,55) PRMAX,LMORMX,NRMAX,PRMIN,LMORMN,NRMIN

55
FORMAT(1H0,44H MAX AND MIN VALUES OF INDEPENDENT VARIABLES/
          4X,6H PRMAX,3X,7H LMORMX,4X,6H NRMAX,4X,6H PRMIN,
     3X,7H LMORMN,4X,6H NRMIN/6F10.4)

C  READ CONSTANTS FOR OPTIMIZATION

READ (5,60)N,KM,IVCON,TOL,DELX,DELY

60
FORMAT (3I10,3F10.6)

WRITE (6,65) N,KM,IVCON,TOL,DELX,DELY

65
FORMAT (1H0,23H OPTIMIZATION CONSTANTS/
        5X,5H NOPT,7X,3H KM,4X,6H IVCON,6X,4H TOL,
        5X,5H DELX,5X,5H DELY/3I10,3F10.6)
```

CONT. ON PG 4

```
C  READ TRANSMISSION AND GENERATOR CONSTANTS
      READ (5,70) NG
70    FORMAT (F10.4)
      WRITE (6,75) NG
75    FORMAT (1H0,37H GENERATOR AND TRANSMISSION CONSTANTS/
     1H 7X3H NG/1H F10.4)
C  READ ECONOMICS CONSTANTS
      READ (5,80) LIFE,IDEPM,NU,DEBT,EQTY,DINT,EINT,TAX,TXCR
80    FORMAT (2I10/7F10.4)
      WRITE(6,85) LIFE,IDEPM,NU,DEBT,EQTY,DINT,EINT,TAX,TXCR
85    FORMAT (1H0,20H ECONOMICS CONSTANTS/6X,5H LIFE,4X,6H IDEPM/2I10/
              5X,5H   NU,5X,5H DEBT,5X,5H EQTY,5X,5H DINT,
      5X,5H EINT,6X,4H TAX,5X,5H TXCR/1X,7F10.4)
C  READ OPTIONS
      READ (5,77) IOPT
77    FORMAT (I10)
      WRITE (6,78) IOPT
78    FORMAT (1H0,8H OPTIONS/5X,5H IOPT/I10)
```

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CONT. ON PG 5

```
RETURN
```

```
100
WRITE (6,110) Y(1),F(1),G(1)
```

```
110
FORMAT (1H0,50H FINAL OBJECTIVE, PAYOFF, AND CONSTRAINT FUNCTIONS/
        2X,10H OBJECTIVE,5X,7H PAYOFF,1X,11H CONSTRAINT/3F12.4)
```

```
WRITE (6,120) PR,LAMDAR,NR,D,VR,VCO,VCI,PMIN ,E,PD,VRDVM,VWDVO,
              LSTITD,TR,TCO,TCI,CPMIN,EW,ETASYS,CPPR,CTPR,RTR,
              QG,QR,LSTITP
```

```
120
FORMAT (1H0,5X,7H PRATED,5X,7H LAMDAR,5X,7H NROTOR,3X,9H DIAMETER,
        5X,7H VRATED,8X,4H VCO,8X,4H VCI,6X,6H  PMIN,9X,7H ENERGY/
        1H ,7X,5H (KW),12X,6X,6H (RPM),3X,9H (METERS),
        3(4X,8H (M/SEC)),7X,5H (KW),7X,9H (KW-HRS)/
        1H ,8F12.4,F16.4//
        1H ,3X,9H PDENSITY,6X,6H VR/VM,6X,6H VW/VO,5X,7H LSTITD,
        9X,3H TR,8X,4H TCO,8X,4H TCI,6X,6H CPMIN,10X,6H EWIND/
        1H 2X,10H (KW/M**2),12X,12X,12X,3(4X,8H (HOURS)),
```

```
        12X,7X,9H (KW-HRS)/1H ,3F12.4,I12,4F12.4,F16.4/
        /1H ,5X,7H ETASYS,7X,5H CPPR,7X,5H CTPR,1X,11H RTR THRUST,
        4X,8H QGENRTR,12X,5X,7H QROTOR,5X,7H LSTITP/
        1H ,12X,12X,12X,2X,10H (NEWTONS),5X,7H (NT-M),12X,5X,
        7H (NT-M)/1H ,5F12.4,8X,F16.4,I12)
```

```
WRITE (6,130) ((C(I,J),J=1,6),I=1,2), (C(I,6),I=3,5)
```

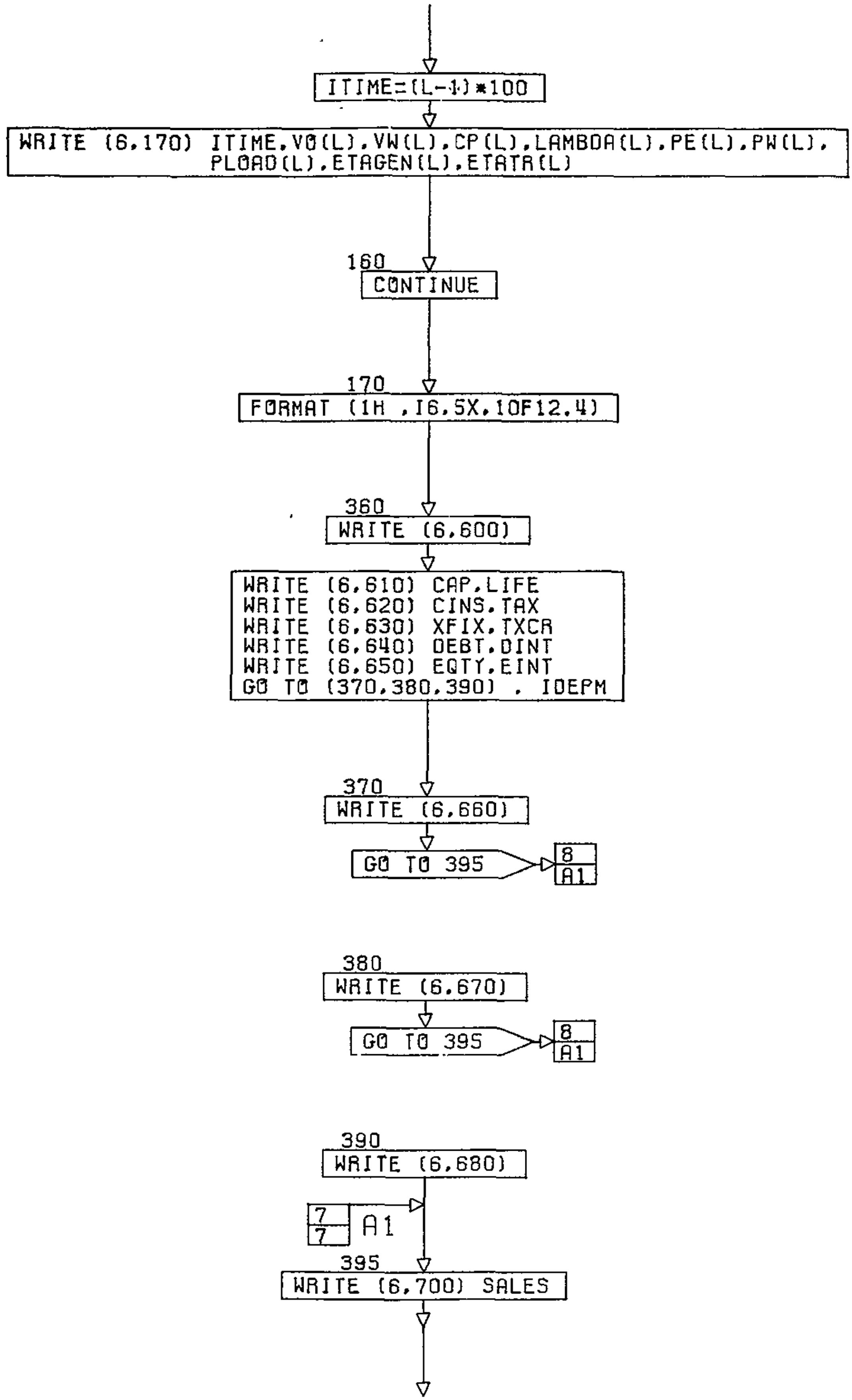
```
130
FORMAT (1H0,14H COST(DOLLARS),7X,6H TOWER,8X,6H ROTOR,4X,
         10H GENERATOR,6X,8H TRANSMN,5X,9H CONTROLS,8X,6H TOTAL/
         12H FABRICATION,2X,6F14.4/9H DELIVERY,5X,6F14.4/
         13H INSTALLATION,71X,F14.4/
         25H SYSTEM ENGRNG AND ADMIN,59X,F14.4/
         14H TOTAL CAPITAL,70X,F14.4)
```

```
WRITE (6,140) (WT(I),I=1,6)
```

CONT. ON PG 6

```
140
FORMAT (1H0,7H WEIGHT,14X,6H TOWER,8X,6H ROTOR,4X,10H GENERATOR,
        6X,8H TRANSMN,5X,9H CONTROLS,8X,6H TOTAL/
        12H (KILOGRAMS),2X,6F14.4)

WRITE (6,32) COTHR(1),COTHR(2),WTOTHR(1),WTOTHR(2)

32
FORMAT (1H0,33H TOWER FABRICATION COST BREAKDOWN/
        1H ,5X,7H PINTLE,2X,10H STRUCTURE/1H ,2F12.4/1H ,
        23H TOWER WEIGHT BREAKDOWN/1H ,5X,7H PINTLE,2X,
        10H STRUCTURE/1H ,2F12.4)

WRITE (6,34) COTHR(5),COTHR(6),WTOTHR(5),WTOTHR(6)

34
FORMAT (1H0,37H GENERATOR FABRICATION COST BREAKDOWN/
        1H ,2X,10H GENERATOR,2X,10H ACCESSORY/1H ,2F12.4/1H ,
        27H GENERATOR WEIGHT BREAKDOWN/1H ,2X,10H GENERATOR,
        2X,10H ACCESSORY/1H ,2F12.4)

WRITE (6,36) (COTHR(I),I=7,12),WTOTHR(8),WTOTHR(7)

36
FORMAT (1H0, 40H TRANSMISSION FABRICATION COST BREAKDOWN/
        1H ,2X,9H GEAR BOX,6X,6H BRAKE,5X,7H CLUTCH,6X,6H SHAFT,
        4X,8H BEARING,2X,10H COUPLINGS/1H ,6F12.4/1H ,
        30H TRANSMISSION WEIGHT BREAKDOWN/1H ,3X,9H GEAR BOX,
        5X,7H OTHERS/1H ,2F12.4)

WRITE (6,38) (C(6,J),J=1,6)

38
FORMAT (1H0,18H MAINTENANCE COSTS/6X,6H TOWER,6X,6H ROTOR,2X,
        10H GENERATOR,4X,8H TRANSMN,2X,10H  CONTROLS,6X,6H TOTAL/
        6F12.4)

WRITE (6,150)

150
FORMAT (1H0,1X,5H TIME,10X,7H WIND 0,5X,5H WIND,8X,3H CP,
        9X,7H LAMBDA,3X,3H PE,9X,3H PW,11X,6H PLOAD,6X,5H ETAG,
        7X5H ETAT)

DO 160 L=1,LVW
```

CONT. ON PG 7


ITIME=(L-1)*100
WRITE (6,170) ITIME,VO(L),VW(L),CP(L),LAMBDA(L),PE(L),PW(L),
PLOAD(L),ETAGEN(L),ETATR(L)
160
CONTINUE
170
FORMAT (1H ,I6,5X,10F12.4)
360
WRITE (6,600)
WRITE (6,610) CAP,LIFE
WRITE (6,620) CINS,TAX
WRITE (6,630) XFIX,TXCR
WRITE (6,640) DEBT,DINT
WRITE (6,650) EQTY,EINT
GO TO (370,380,390) , IDEPM
370
WRITE (6,660)
GO TO 395
8
A1
380
WRITE (6,670)
GO TO 395
8
A1
390
WRITE (6,680)
7
7
A1
395
WRITE (6,700) SALES

CONT. ON PG 8

```
      WRITE (6,710)
      DO 400 I=1,LIFE
      WRITE(6,720) I,CR(I),DEP(I),GS(I),TX(I),XNET(I),SF(I)
  400 CONTINUE
  600 FORMAT (1H0,//30H ENERGY INVESTMENT ANALYSIS   /)
  610 FORMAT(1H0,3H  $,F11.2,11H INVESTMENT,5X,I6,12H YEARS LIFE )
  620 FORMAT (1H ,F14.2,14H START-UP COST,F8.3,9H TAX RATE   )
  630 FORMAT (1H ,F14.2,13H ANNUAL COSTS,F9.3,11H TAX CREDIT  )
  640 FORMAT (1H ,F14.3,14H DEBT FRACTION,F8.3,14H DEBT INTEREST   )
  650 FORMAT (1H ,F14.3,13H EQUITY FRAC.,F9.3,14H EQUITY RETURN   )
  660 FORMAT (1H ,30X,24H STR. LINE DEPRECIATION        )
  670 FORMAT (1H ,30X,20H S.Y.D. DEPRECIATION      )
```

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CONT. ON PG 9

```
680
FORMAT (1H ,30X,20H D.D.B. DEPRECIATION         )

700
FORMAT(1H0,F14.2,28H ANNUAL REVENUE REQUIREMENT  /)

710
FORMAT (1H0,1X,3HYR.,2X,9HCASH REQ.,4X,5HDEPR.,6X,5HGROSS,7X,
        5HTAXES,7X,3HNET,6X,6HCAP F0)

720
FORMAT (1H ,I3,6F11.2)

RETURN

800
WRITE (6,810)K,F(1),G(1),PR,LAMDAR,NR

810
FORMAT (1H0,11H BASE CASE ,I3/8H PAYOFF=,F12.4,
        10X,12H CONSTRAINT=,F12.4,10X,4H PR=,F10.4,
        5X,8H LAMDAR=,F10.4,5X,4H NR=,F10.4)

RETURN

END
```

```
C
      SUBROUTINE READ (VEC,KDVEC,LVEC)
      DIMENSION VEC(KDVEC),TITLE(35)
      INTEGER TITLE
      READ (5,1) LVEC,IPRNT,(TITLE(I),I=1,35)
    1 FORMAT (2I5,35A2)
      READ (5,2) (VEC(I),I=1,LVEC)
    2 FORMAT (8F10.4)
      IF (IPRNT.EQ.0) GO TO 10
      WRITE (6,4) LVEC,IPRNT,(TITLE(I),I=1,35)
    4 FORMAT (1H0,5X,2I5,35A2)
      WRITE (6,5) (VEC(I),I=1,LVEC)
    5 FORMAT (1H ,8F10.4)
   10 CONTINUE
      RETURN
      END
```

```
C

SUBROUTINE  DITER

COMMON C(6,6),WT(6),MILPKW,ANNREV,NU,WTOTHR(10),COTHR(15),
       DEP(50),GS(50),TX(50),XNET(50),SF(50),CR(50),CAP,CINS,
       LIFE,DEBT,EQTY,DINT,EINT,TAX,TXCR,IDEPM,XFIX

COMMON NITER,D,VRDVM,CPPR,ETAT,NG,VR,QR,GR,QG,ETAGO,HR,CLRNCE,
       TERFAC,HO,RHO,VMEAN,VTIP,NERROR,PD,LSTITD

COMMON VW(90),LAMBDA(90),CP(90),ETAGEN(90),PLOAD(90),PE(90),
       CPMIN,PGMIN,
       VCI,VCO,VWDVO,TR,TCO,E,IVCO,IVCI,TCI,LSTITP,EW,ETASYS,
        PW(90),ETATR(90)

COMMON VO(90),CP1(30),CT1(30),LAMDA1(30),LVW,LCP1,LCT1,LLAM1

COMMON PR,LAMDAR,NR,PRMAX,LMDRMX,NRMAX,PRMIN,LMDRMN,NRMIN,
       N,KM,K,TOL,DELX,DELY,X(3),Y(4),F(4),G(4),IVCON

COMMON HT,CTPR,TRTR,IOPT
REAL MILPKW,NU
REAL NG
REAL LAMDA1
REAL LAMBDA
REAL LAMDAR,NR,LMDRMX,NRMAX,LMDRMN,NRMIN
DATA KDVW,KDCP1,KDCT1,KDLAM1/90,30,30,30/
PI=3.1415927

PRO=PR
VRAX=1.5*VMEAN
CPPR=VALUEY(LAMDA1,CP1,KDLAM1,KDCP1,LCP1,LAMDAR)
ETAGO=ETAG(PR,1.0)
ETAT=ETATRN(ETAGO,1.0)
DO 20 I=1,NITER
VTIP=LAMDAR*VRAX
D=VTIP*60./(PI*NR)

PR=PI*CPPR*ETAT*ETAGO*VRAX**3*D**2*RHO*.000125
ERR=PRO-PR
IF (ERR*ERR.LT.1.E-6) GO TO 60
VRAX=VRAX*(PRO/PR)**.2

20 CONTINUE

60 CONTINUE
```

CONT. ON PG   2

```
HR=D*.5+CLRNCE
VWOVO=(HR/HO)**TERFAC
VR=VRAX/VWOVO
IF (VR.GT.VO(1)) NERROR=1
VROVM=VR/VMEAN
PD=4.*PR/(PI*D*D)
LSTITD=I
```

```
RETURN
```

```
END
```

```
C

SUBROUTINE TOWER

COMMON C(6,6),WT(6),MILPKW,ANNREV,NUT,WTOTHR(10),COTHR(15),
       DEP(50),GS(50),TX(50),XNET(50),SF(50),CR(50),CAP,CINS,
       LIFE,DEBT,EQTY,DINT,EINT,TAX,TXCR,IDEPM,XFIX

COMMON NITER,D,VRDVM,CPPR,ETAT,NG,VR,QR,GR,QG,ETAGO,HR,CLRNCE,
       TERFAC,HO,RHO,VMEAN,VTIP,NERHOR,PD,LSTITD

COMMON VW(90),LAMBDA(90),CP(90),ETAGEN(90),PLOAD(90),PE(90),
       CPMIN,PGMIN,
       VCI,VCO,VWDVO,TR,TCO,E,IVCO,IVCI,TCI,LSTITP,EW,ETASYS,
        PW(90),ETATR(90)

COMMON VO(90),CP1(30),CT1(30),LAMDA1(30),LVW,LCP1,LCT1,LLAM1

COMMON PR,LAMDAR,NR,PRMAX,LMDRMX,NRMAX,PRMIN,LMDRMN,NRMIN,
       N,KM,K,TOL,DELX,DELY,X(3),Y(4),F(4),G(4),IVCON

COMMON HT,CTPR,TRTR,IOPT
REAL MILPKW,NUT
REAL NG
REAL LAMDA1
REAL LAMBDA
REAL LAMDAR,NR,LMDRMX,NRMAX,LMDRMN,NRMIN
DATA KDVW,KDCP1,KDCT1,KDLAM1/90,30,30,30/
PI=3.1415927

HT=HR
CTPR=VALUEY(LAMDA1,CT1,KDLAM1,KDCT1,LCT1,LAMDAR)
TRTR=CTPR*PI*RHO*VR*VR*D*D*.125*VWDVO*VWDVO
TEHT=1.5*TRTR*HT

C  WTOTHR(1) IS PINTLE WEIGHT

WTOTHR(1)=443.2*PR**.44

C  COTHR(1) IS PINTLE COST

COTHR(1)=1212.*PR**.427

C  COTHR(2) IS FABRICATION COST OF TOWER
C  COTHR(3) IS PINTLE MAINTENANCE COST

COTHR(3)=10.*PR**.5
GO TO (10,35),IOPT
```

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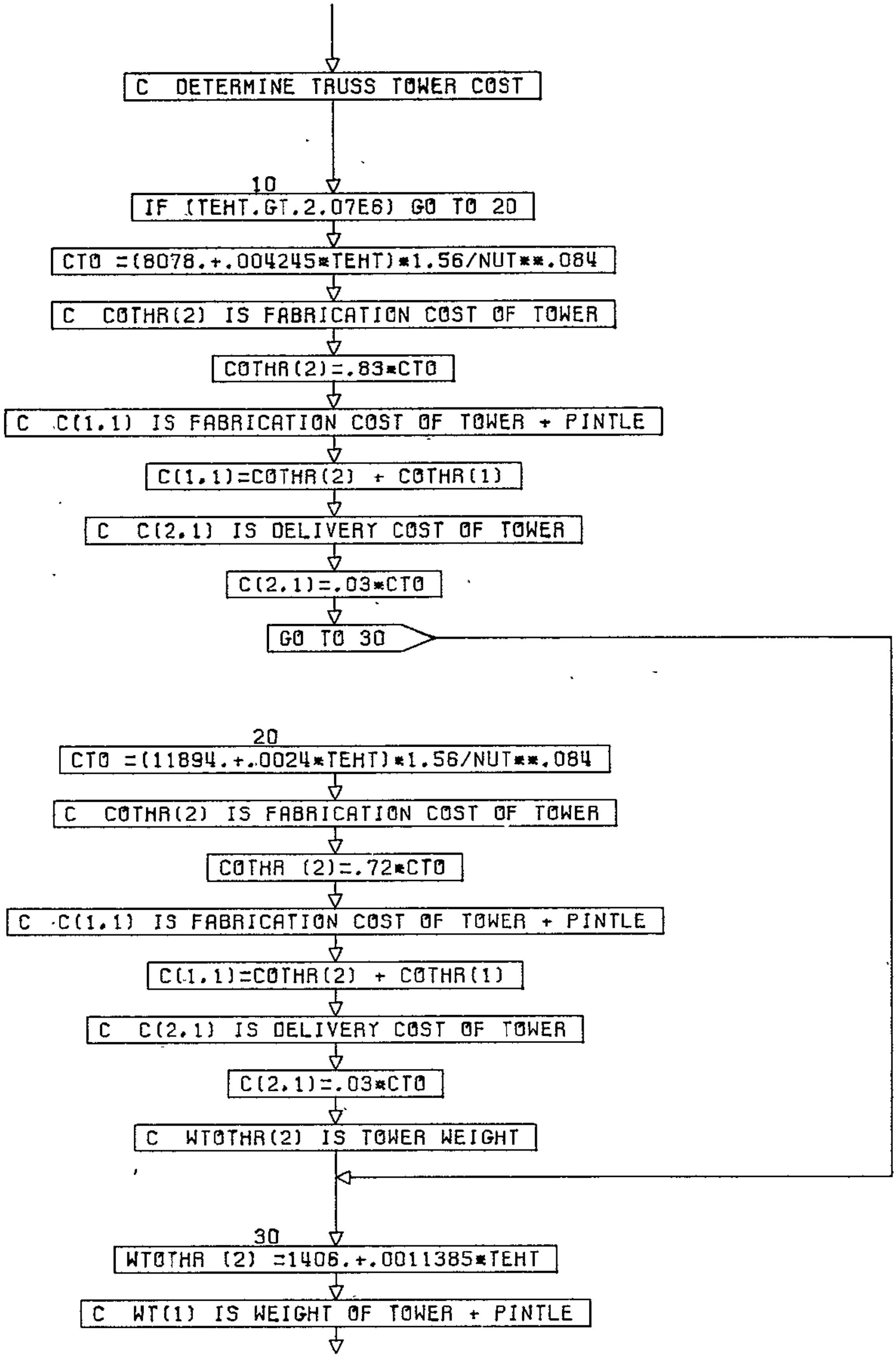
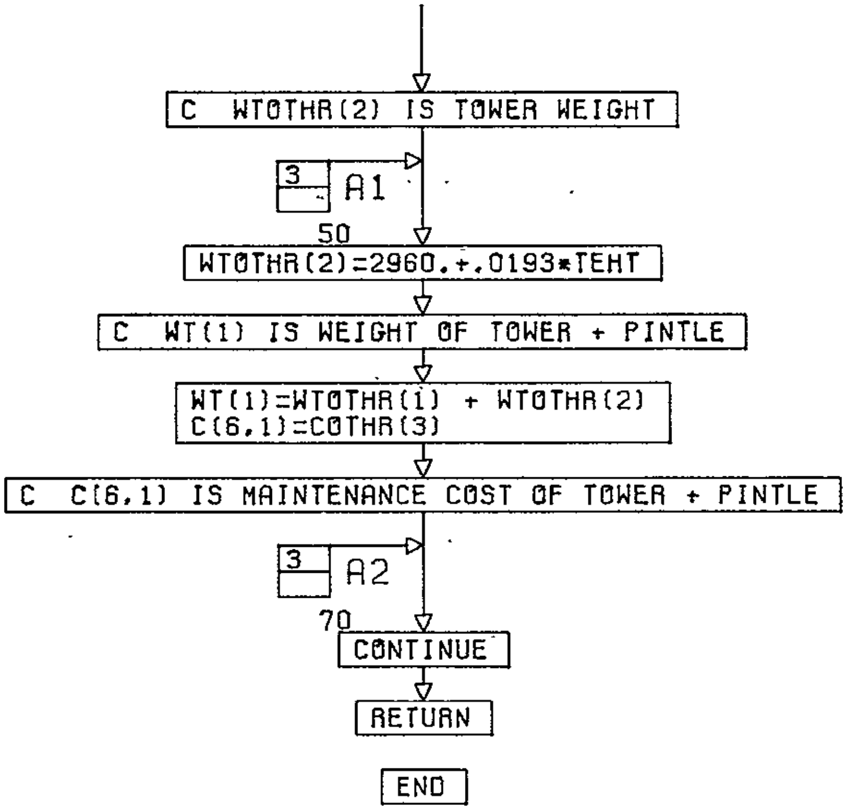
CONT. ON PG 2


C DETERMINE TRUSS TOWER COST
10
IF (TEHT.GT.2.07E6) GO TO 20
CTO =(8078.+.004245*TEHT)*1.56/NUT**.084
C COTHR(2) IS FABRICATION COST OF TOWER
COTHR(2)=.83*CTO
C C(1,1) IS FABRICATION COST OF TOWER + PINTLE
C(1,1)=COTHR(2) + COTHR(1)
C C(2,1) IS DELIVERY COST OF TOWER
C(2,1)=.03*CTO
GO TO 30
20
CTO =(11894.+.0024*TEHT)*1.56/NUT**.084
C COTHR(2) IS FABRICATION COST OF TOWER
COTHR (2)=.72*CTO
C C(1,1) IS FABRICATION COST OF TOWER + PINTLE
C(1,1)=COTHR(2) + COTHR(1)
C C(2,1) IS DELIVERY COST OF TOWER
C(2,1)=.03*CTO
C WTOTHR(2) IS TOWER WEIGHT
30
WTOTHR (2) =1406.+.0011385*TEHT
C WT(1) IS WEIGHT OF TOWER + PINTLE

CONT. ON PG 3

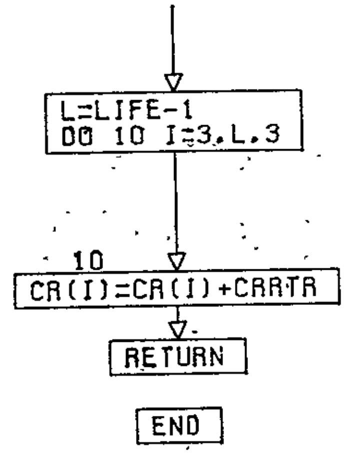
```
WT(1)=WTOTHR(1) + WTOTHR(2)
C  COTHR(4) IS TOWER MAINTENANCE COST
COTHR(4)=311.+.0256*WTOTHR(2)
C C(6,1) IS TOTAL TOWER + PINTLE MAINTENANCE COST
C(6,1) =COTHR(3)+COTHR(4)
GO TO 70          4 / A2

C  DETERMINE CONCRETE TOWER COST

35
EX=(FLOAT(NUT)-1.)/FLOAT(NUT)
IF (TEHT.GT.2.44E6) GO TO 40
CTO =(5053.+.00411*TEHT)/1.13**EX
C  COTHR(2) IS FABRICATION COST OF TOWER
COTHR(2)=.45*CTO
C(1,1)=COTHR(2) + COTHR(1)
C  C(2,1) IS DELIVERY COST OF TOWER
C(2,1)=.05*CTO
GO TO 50          4 / A1

40
CTO =(9397.+.00233*TEHT)/1.13**EX
C  COTHR(2) IS FABRICATION COST OF TOWER
COTHR(2)=.65*CTO
C  C(1,1) IS FABRICATION COST OF TOWER + PINTLE
C(1,1)=COTHR(2) + COTHR(1)
C  C(2,1) IS DELIVERY COST OF TOWER
C(2,1)=.05*CTO
```

CONT. ON PG 4


C WTOTHR(2) IS TOWER WEIGHT
3
A1
50
WTOTHR(2)=2960.+.0193*TEHT
C WT(1) IS WEIGHT OF TOWER + PINTLE
WT(1)=WTOTHR(1) + WTOTHR(2)
C(6,1)=COTHR(3)
C C(6,1) IS MAINTENANCE COST OF TOWER + PINTLE
3
A2
70
CONTINUE
RETURN
END

```
C

SUBROUTINE ROTOR

COMMON C(6,6),WT(6),MILPKW,ANNREV,NUR,WTOTHR(10),COTHR(15),
       DEP(50),GS(50),TX(50),XNET(50),SF(50),CR(50),CAP,CINS,
       LIFE,DEBT,EQTY,DINT,EINT,TAX,TXCR,IDEPM,XFIX

COMMON NITER,D,VRDVM,CPPR,ETAT,NG,VR,QR,GR,QG,ETAGO,HR,CLRNCE,
       TERFAC,HO,RHO,VMEAN,VTIP,NERROR,PD,LSTITO

COMMON VW(90),LAMBDA(90),CP(90),ETAGEN(90),PLOAD(90),PE(90),
       CPMIN,PGMIN,
       VCI,VCO,VWDVO,TR,TCO,E,IVCO,IVCI,TCI,LSTITP,EW,ETASYS,
        PW(90),ETATR(90)

COMMON VO(90),CP1(30),CT1(30),LAMDA1(30),LVW,LCP1,LCT1,LLAM1

COMMON PR,LAMDAR,NR,PRMAX,LMDRMX,NRMAX,PRMIN,LMDRMN,NRMIN,
       N,KM,K,TOL,DELX,DELY,X(3),Y(4),F(4),G(4),IVCON

COMMON HT,CTPR,TRTR,IOPT
REAL MILPKW,NUR
REAL NG
REAL LAMBDA
REAL LAMDA1
REAL LAMDAR,LMDRMX,LMDRMN,NR,NRMAX,NRMIN

C  WT(2)  IS ROTOR WEIGHT

WT(2)=.079*(D*3.281)**2.256

C  C(1,2) IS FABRICATION COST OF ROTOR

C(1,2)=33.77*D**2.22/NUR**.11 +20000.

C  C(2,2) IS DELIVERY COST OF ROTOR

C(2,2)=.0882*WT(2) + 32.8*D

C   C(6,2)IS MAINTENANCE COST

C(6,2)=.02*(C(1,2)+C(2,2))

C  CR IS RECURRING COST

CRRTR=.421*D**2.3
```

CONT. ON PG 2


L=LIFE-1
DO 10 I=3,L,3
10
CR(I)=CR(I)+CRRTR
RETURN
END

```
C

SUBROUTINE GEN2TR

COMMON C(6,6),WT(6),MILPKW,ANNREV,NUG,WTOTHR(10),COTHR(15),
       DEP(50),GS(50),TX(50),XNET(50),SF(50),CR(50),CAP,CINS,
       LIFE,DEBT,EQTY,DINT,EINT,TAX,TXCR,IDEPM,XFIX

COMMON NITER,D,VRDVM,CPPR,ETAT,NG,VR,QR,GR,QG,ETAGO,HR,CLRNCE,
       TERFAC,HO,RHO,VMEAN,VTIP,NERROR,PD,LSTITD

COMMON VW(90),LAMBDA(90),CP(90),ETAGEN(90),PLOAD(90),PE(90),
       CPMIN,PGMIN,
       VCI,VCO,VWDVO,TR,TCO,E,IVCO,IVCI,TCI,LSTITP,EW,ETASYS,
        PW(90),ETATR(90)

COMMON VO(90),CP1(30),CT1(30),LAMDA1(30),LVW,LCP1,LCT1,LLAM1

COMMON PR,LAMDAR,NR,PRMAX,LMDRMX,NRMAX,PRMIN,LMDRMN,NRMIN,
       N,KM,K,TOL,DELX,DELY,X(3),Y(4),F(4),G(4),IVCON

COMMON HT,CTPR,TRTR,IOPT
REAL MILPKW,NUG
REAL NG
REAL LAMBDA
REAL LAMDA1
REAL LAMDAR,NR,LMDRMX,NRMAX,LMDRMN,NRMIN
PI=3.1415927
QR=PI*RHO*CPPR*VR*VR*D*D*D/(16.*LAMDAR)*VWDVO*VWDVO

GR=NG/NR
QG=QR*ETAT/GR

C  WTOTHR(5) IS GENERATOR WEIGHT

WTOTHR(5)=12.2*PR**.825

C  WTOTHR (6) IS ACCESSORY WEIGHT

WTOTHR(6)=24.22*PR**.438

C  WT(3) IS GENERATOR + ACCESSORY WEIGHT

WT(3)=WTOTHR(5)+WTOTHR(6)

C  COTHR(5) IS GENERATOR FABRICATION COST

COTHR(5)=84.116*PR**.835/NUG**.017

C  COTHR(6) IS ACCESSORY COST
```

CONT. ON PG 2

```
COTHR(6)=10.713*PR**1.11/NUG**.0235
C  C(1,3) IS FABRICATION COST OF GENERATOR  + ACCESSORY
C(1,3)=COTHR(5)+COTHR(6)
C  C(2,3) IS DELIVERY COST OF GENERATOR
C(2,3)=.1324*WT(3)
C  C(6,3) IS MAINTENANCE COST
C(6,3)=60.*PR**.3
C  CR IS RECURRING COST
CRGEN=.6*(C(1,3)+C(2,3))
L=LIFE-1
IF (L.LT.30) GO TO 31
DO 10 I=30,L,30
10
CR(I)=CR(I)+CRGEN
31
CONTINUE
C  WTOTHR(7) IS WEIGHT OF BRAKES, CLUTCH, SHAFT, BEARING,COUPLING
WTOTHR(7)=.0278*QR**.998
C  WTOTHR(8) IS GEAR BOX WEIGHT
WTOTHR(8)=.4644*QR**.815
C  WT(4) IS TRANSMISSION WEIGHT
WT(4)=WTOTHR(7) + WTOTHR(8)
C  COTHR(7) IS GEAR BOX COST
COTHR(7)=2.987*QR**.795
C  COTHR(8) IS BRAKE COST
```

CONT. ON PG 3

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```
      COTHR(8)=145.7*PR**.3
C     COTHR(9) IS CLUTCH COST
      COTHR(9)=344.4*PR**.188
C     COTHR(10) IS SHAFT COST
      COTHR(10)=.01835*QR**1.0967
C     COTHR(11) IS BEARING COST
      COTHR(11)=.150*QR**.8645
C     COTHR(12) IS COUPLINGS COST
      COTHR(12)=.0324*QR**1.0297
C     C(1,4) IS FABRICATION COST OF TRANSMISSION
      C(1,4)=0.
      DO 40 I=7,12
   40 C(1,4)=C(1,4)+COTHR(I)
      C(1,4)=C(1,4)/NUG**.023
C     C(2,4) IS DELIVERY COST OF TRANSMISSION
      C(2,4)=.1782*WT(4)
C     C(6,4) IS MAINTENANCE COST
      C(6,4)=48.*PR**.5
C     CR IS RECURRING COST
      CRTRA=(C(1,4)+C(2,4))/6.
      L=LIFE-1
      DO 20 I=5,L,5
   20 CR(I)=CR(I)+CRTRA
```

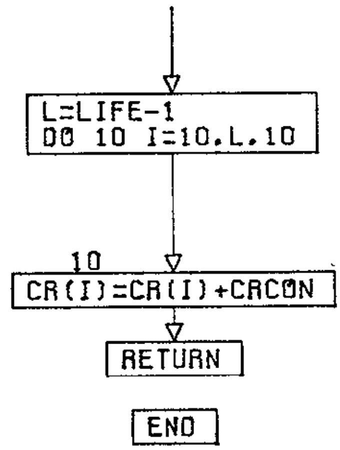
CONT. ON PG 4

RETURN

END

```
C
SUBROUTINE CONTRL
COMMON C(6,6),WT(6),MILPKW,ANNREV,NUC,WTOTHR(10),COTHR(15),
       DEP(50),GS(50),TX(50),XNET(50),SF(50),CR(50),CAP,CINS,
       LIFE,DEBT,EQTY,DINT,EINT,TAX,TXCR,IDEPM,XFIX
COMMON NITER,D,VRDVM,CPPR,ETAT,NG,VR,QR,GR,QG,ETAGO,HR,CLRNCE,
       TERFAC,HO,RHO,VMEAN,VTIP,NERROR,PD,LSTITD
COMMON VW(90),LAMBDA(90),CP(90),ETAGEN(90),PLOAD(90),PE(90),
       CPMIN,PGMIN,
       VCI,VCO,VWDVO,TR,TCO,E,IVCO,IVCI,TCI,LSTITP,EW,ETASYS,
       PW(90),ETATR(90)
COMMON VO(90),CP1(30),CT1(30),LAMDA1(30),LVW,LCP1,LCT1,LLAM1
COMMON PR,LAMDAR,NR,PRMAX,LMDRMX,NRMAX,PRMIN,LMDRMN,NRMIN,
       N,KM,K,TOL,DELX,DELY,X(3),Y(4),F(4),G(4),IVCON
COMMON HT,CTPR,TRTR,IOPT
REAL MILPKW,NUC
REAL NG
REAL LAMBDA
REAL LAMDA1
REAL LAMDAR,NR,LMDRMX,NRMAX,LMDRMN,NRMIN
C   WT(5)=CONTROL SYSTEM WEIGHT
WT(5)=0.0
C   C(1,5) IS FABRICATION COST OF CONTROL SYSTEM
C(1,5)=(3636.+9.2*PR)/NUC**.0235
C   C(2,5) IS DELIVERY COST OF CONTROL SYSTEM
C(2,5)=.004*C(1,5)
C   C(6,5) IS MAINTENANCE COST
C(6,5)=30.*PR**.5
C   CR IS RECURRING COST
CRCON=570.+1.43*PR
```


CONT. ON PG 2


L=LIFE-1
DO 10 I=10,L,10
10
CR(I)=CR(I)+CRCON
RETURN
END

```
C

SUBROUTINE ENERGY

COMMON C(6,6),WT(6),MILPKW,ANNREV,NU,WTOTHR(10),COTHR(15),
       DEP(50),GS(50),TX(50),XNET(50),SF(50),CR(50),CAP,CINS,
       LIFE,DEBT,EQTY,DINT,EINT,TAX,TXCR,IDEPM,XFIX

COMMON NITER,D,VRDVM,CPPR,ETAT,NG,VR,QR,GR,QG,ETAGO,HR,CLANCE,
       TERFAC,HO,RHO,VMEAN,VTIP,NERROR,PD,LSTITD

COMMON VW(90),LAMBDA(90),CP(90),ETAGEN(90),PLOAD(90),PE(90),
       CPMIN,PMIN,
       VCI,VCO,VWDVO,TR,TCO,E,IVCO,IVCI,TCI,LSTITP,EW,ETASYS,
        PW(90),ETATR(90)

COMMON VO(90),CP1(30),CT1(30),LAMDA1(30),LVW,LCP1,LCT1,LLAM1

COMMON PR,LAMDAR,NR,PRMAX,LMDRMX,NRMAX,PRMIN,LMDRMN,NRMIN,
       N,KM,K,TOL,DELX,DELY,X(3),Y(4),F(4),G(4),IVCON

COMMON HT,CTPR,TATR,IOPT
REAL MILPKW,NU
REAL NG
REAL LAMBDA
REAL LAMDA1
REAL LAMDAR,NR,LMDRMX,NRMAX,LMDRMN,NRMIN
DATA KDVW,KDCP1,KDCT1,KDLAM1/90,30,30,30/
PI=3.1415927

PGMIN=.001255*PR**1.255+1.5
PMIN=PGMIN + .04*PR
CPGS=.2

1
VAXMIN=(8000.*PMIN/(CPGS*        RHO*PI*D*D))**.3333

GSLMDA=PI*D*NR/(60.*VAXMIN)
IF (GSLMDA.LT.LAMDA1(LLAM1)) GO TO 3
CPGS=CPGS*.5

GO TO 1

3
CONTINUE

DO 2 I=1,NITER
CILMDA=PI*D*NR/(60.*VAXMIN)
```

CONT. ON PG 2

ORIGINAL PAGE IS
OF POOR QUALITY

```
      IF (CILMDA.LE.LAMDA1(LLAM1)) GO TO 6
      NERROR=2
      RETURN

    6 CONTINUE
      CPMIN=VALUEY(LAMDA1,CP1,KDLAM1,KDCP1,LCP1,CILMDA)
      PMINC=   RHO*CPMIN*PI*VAXMIN**3*D*D*.000125
      ERR=PMINC-PMIN
      IF (ERR*ERR.LT.1.E-6) GO TO 4
      VAXMIN=VAXMIN*(PMIN/PMINC)**.05

    2 CONTINUE

    4 VCI=VAXMIN/VWDV0
      LSTITP=I
      ETAGPR=ETAG0
      ETATPR=ETAT
      VC0=20.*(PR/(CPMIN*ETAGPR*ETATPR*D*D))**.3333/VWDV0
      EW=0.
      DO 10 I=1,LVW
      VW(I)=V0(I)*VWDV0
      PW(I)=PI*RHO*VW(I)**3*D*D*.000125
      EW=EW+PW(I)
      LAMBDA(I)=NR*D*PI/(60.*VW(I))
      IF (LAMBDA(I).GT.LAMDA1(LCP1)) GO TO 5
      VALUEX=LAMBDA(I)
      CP(I)=VALUEY(LAMDA1,CP1,KDLAM1,KDCP1,LCP1,VALUEX)
      GO TO 10

    5 CP(I)=0.0

   10 CONTINUE
```

CONT. ON PG 3

ORIGINAL PAGE IS
OF POOR QUALITY

```
EW=EW*100.
IVCO=1
DO 20 I=2,LVW
IF (VO(I).LE.VCO.AND.VO(I-1).GT.VCO) IVCO=I
IF (VO(I).LE.VR.AND.VO(I-1).GT.VR) IVR=I
IF (VO(I).LE.VCI.AND.VO(I-1).GT.VCI) IVCI=I-1
```

```
20 CONTINUE
```

```
IVRM1=IVR-1
TR=FLOAT(IVRM1)*100.
IVCOM1=IVCO-1
TCO=FLOAT(IVCOM1)*100.
IVCIM1=IVCI-1
TCI=FLOAT(IVCIM1)*100.
IVCIP1=IVCI+1
DO 30 I=1,IVRM1
```

```
PLOAD(I)=1.
ETAGEN(I)=ETAGPR
ETATR(I)=ETATPR
PE(I)=PR
```

```
30 CONTINUE
```

```
SUMPE=0.
PLOAD(IVR)=CP(IVR)/CPPR*(VO(IVR)/VR)**3
DO 50 I=IVR,IVCI
PLD=PLOAD(I)
ETAGEN(I)=ETAG(PR,PLD)
ETG=ETAGEN(I)
ETATR(I)=ETATRN(ETG,PLD)
PE(I)=.000125*CP(I)*ETATR(I)*ETAGEN(I)*RHO*PI*VW(I)**3*D*D
```

```
SUMPE=SUMPE+PE(I)
PLOAD(I+1)=PE(I)*CP(I+1)/(PR*CP(I)) * (VW(I+1)/VW(I))**3
```

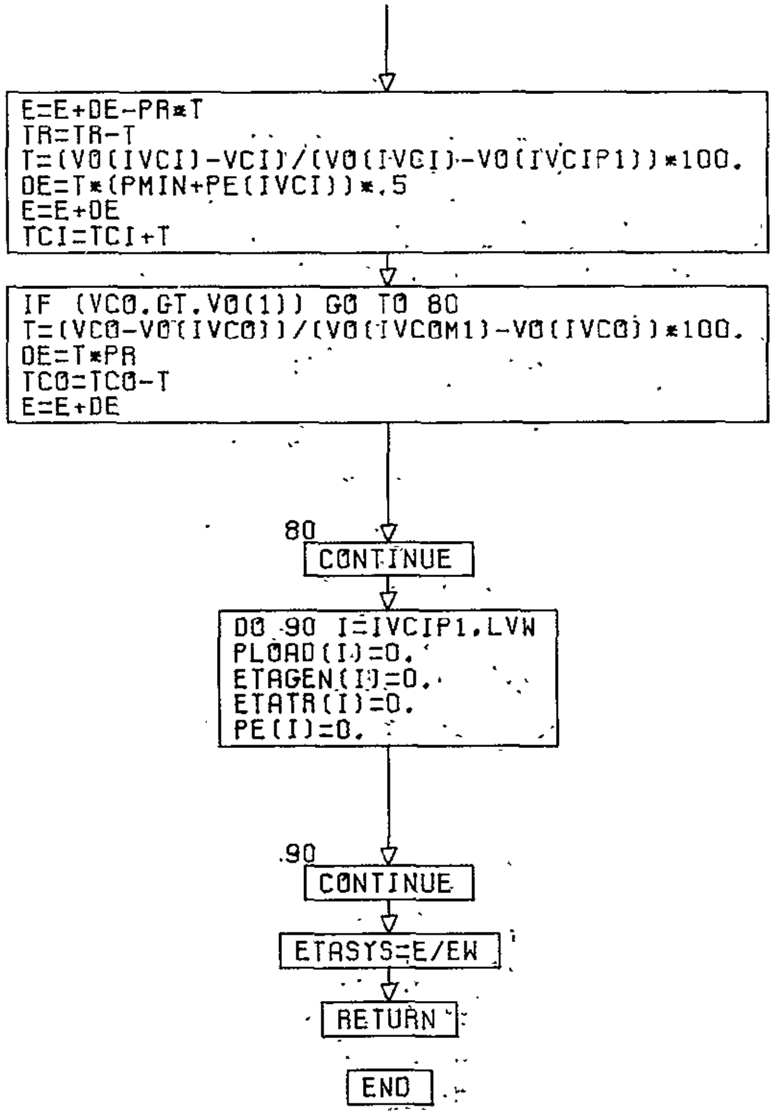
```
50 CONTINUE
```

```
SUMPE=SUMPE-.5*(PE(IVR)+PE(IVCI))
E=SUMPE*100. + PR*(TR-TCO)
```

```
C END POINT CORRECTION FACTORS
```

```
T=(VR-VO(IVR))/(VO(IVRM1)-VO(IVR))*100.
DE=T*(PR+PE(IVR))*.5
```

CONT. ON PG 4



E=E+DE-PR*T
TR=TR-T
T=(V0(IVCI)-VCI)/(V0(IVCI)-V0(IVCIP1))*100.
DE=T*(PMIN+PE(IVCI))*.5
E=E+DE
TCI=TCI+T
IF (VCO.GT.V0(1)) GO TO 80
T=(VCO-V0(IVCO))/(V0(IVCOM1)-V0(IVCO))*100.
DE=T*PR
TCO=TCO-T
E=E+DE
80
CONTINUE
DO 90 I=IVCIP1,LVW
PLOAD(I)=0.
ETAGEN(I)=0.
ETATR(I)=0.
PE(I)=0.
90
CONTINUE
ETASYS=E/EW
RETURN
END

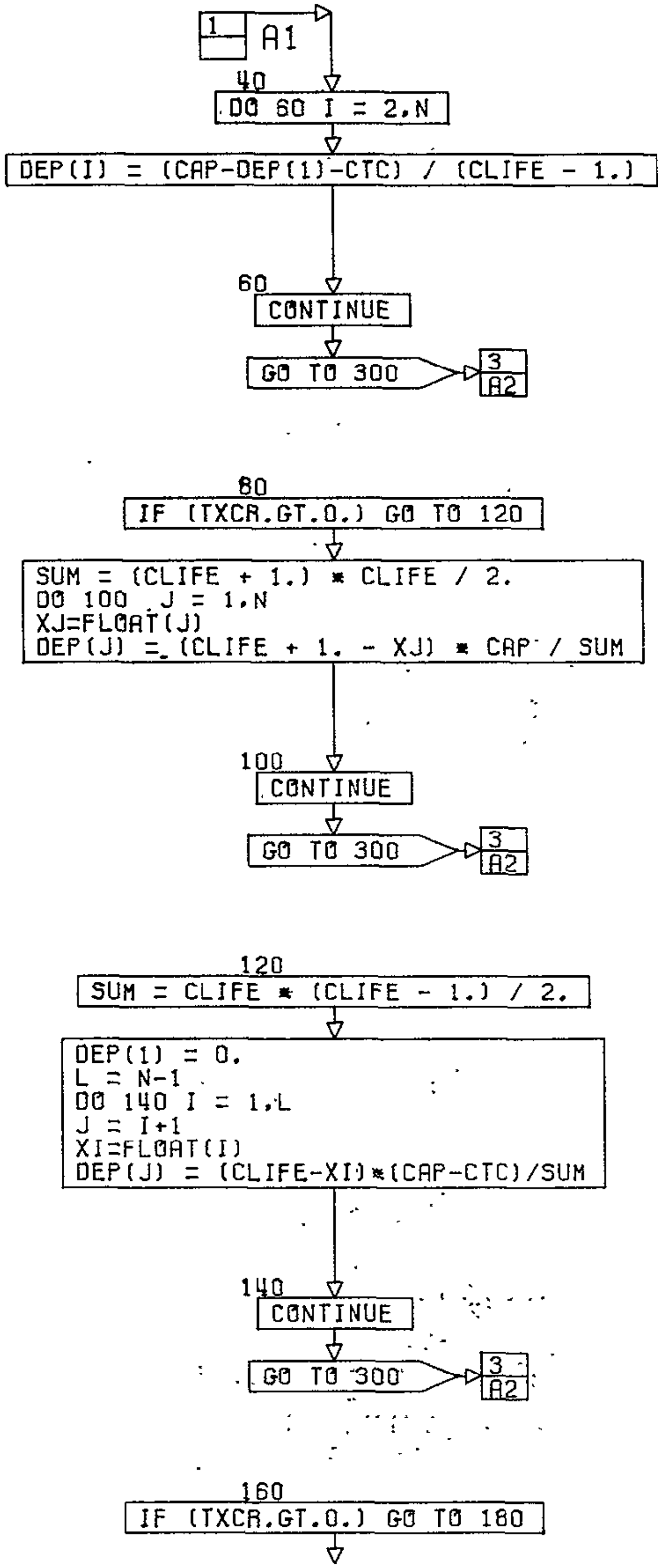
```
C

SUBROUTINE CST2WT

COMMON C(6,6),WT(6),MILPKW,ANNREV,NU,WTOTHR(10),COTHR(15),
       DEP(50),GS(50),TX(50),XNET(50),SF(50),CR(50),CAP,CINS,
       LIFE,DEBT,EQTY,DINT,EINT,TAX,TXCR,IDEPM,XFIX

COMMON NITER,D,VRDVM,CPPR,ETAT,NG,VR,QR,GR,QG,ETAGO,HR,CLRNCE,
       TERFAC,HO,RHO,VMEAN,VTIP,NERROR,PD,LSTITO

COMMON VW(90),LAMBDA(90),CP(90),ETAGEN(90),PLOAD(90),PE(90),
       CPMIN,PGMIN,
       VCI,VCO,VWDVO,TR,TCO,E,IVCO,IVCI,TCI,LSTITP,EW,ETASYS,
       PW(90),ETATR(90)

COMMON VO(90),CP1(30),CT1(30),LAMDA1(30),LVW,LCP1,LCT1,LLAM1

COMMON PR,LAMDAR,NR,PRMAX,LMDRMX,NRMAX,PRMIN,LMDRMN,NRMIN,
       N,KM,K,TOL,DELX,DELY,X(3),Y(4),F(4),G(4),IVCON

COMMON HT,CTPR,TRTR,IOPT
REAL MILPKW,NU
REAL NG
REAL LAMDA1
REAL LAMBDA
REAL LAMDAR,NR,LMDRMX,NRMAX,LMDRMN,NRMIN

C  C(1,6) IS TOTAL FABRICATION COST
C  C(2,6) IS TOTAL DELIVERY COST
C  C(3,6) IS TOTAL INSTALLATION COST
C  C(4,6) IS TOTAL SYSTEM ENGINEERING AND ADMINISTRATION COST
C  C(5,6) IS TOTAL WGS CAPITAL COST
C  C(6,6) IS TOTAL WGS MAINTENANCE COST
C  WT(6) IS TOTAL WGS WEIGHT

WT(6)=0.
C(1,6)=0.
C(2,6)=0.
C(6,6)=0.
DO 10 J=1,5
WT(6)=WT(6)+WT(J)
C(1,6)=C(1,6)+C(1,J)
C(2,6)=C(2,6)+C(2,J)

C(6,6)=C(6,6)+C(6,J)

10 CONTINUE

C(3,6)=12000./NU**.069  + 44.*PR/NU**.29
```


CONT. ON PG 2

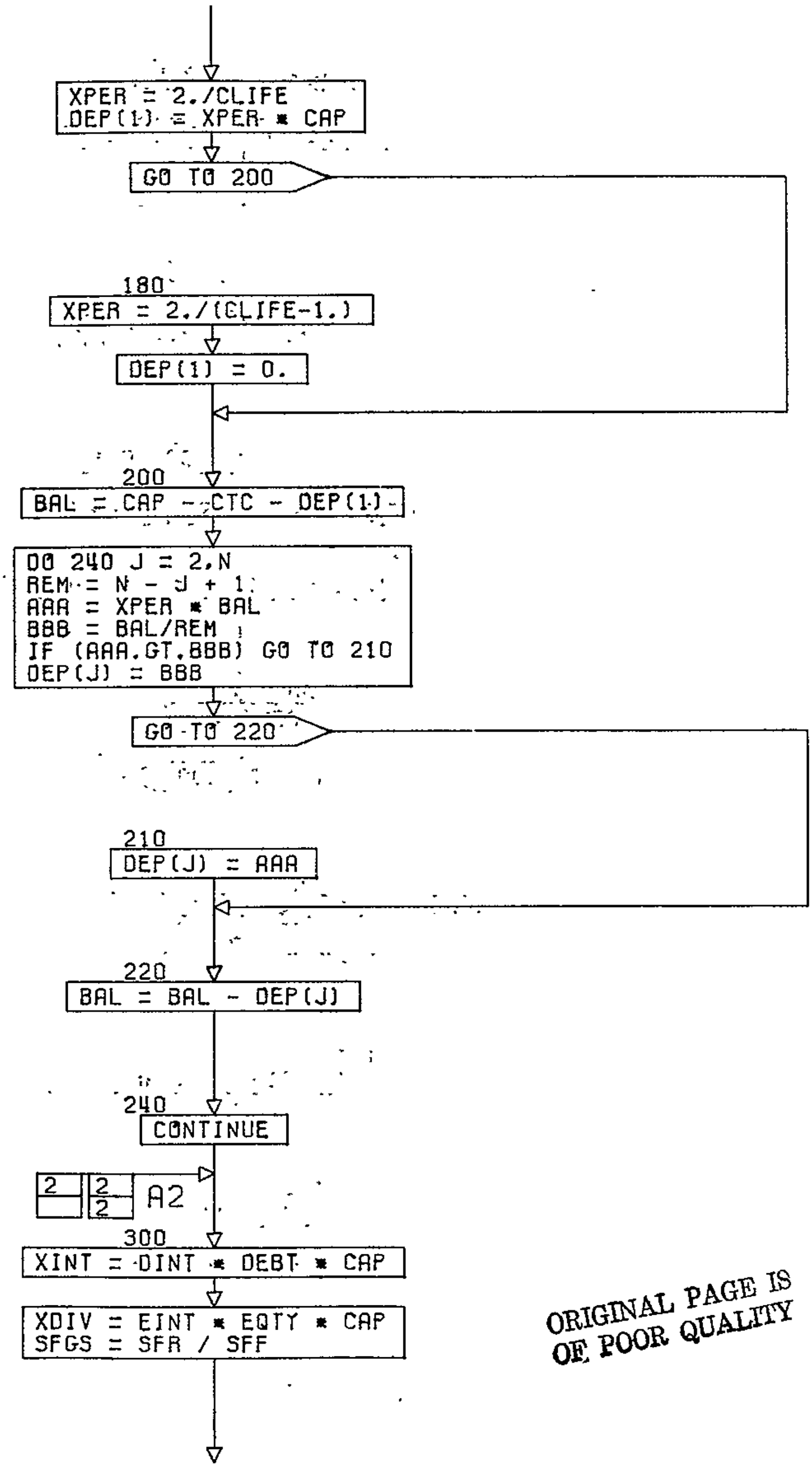
```
CPMC46=C(1,6)+C(2,6)+C(3,6)
C(4,6)=(55000.*NU**.5 + 150000.)*1.07/NU + .07*CPMC46
C(5,6)=CPMC46+C(4,6)
XFIX=C(6,6)
CAP=C(5,6)
CINS=20.*PR+6000.

CALL ECON

MILPKW=ANNREV/E*100.

RETURN

END
```

```
SUBROUTINE ECON

COMMON C(6,6),WT(6),MILPKW,SALES,NU,WTOTHR(10),COTHR(15),
       DEP(50),GS(50),TX(50),XNET(50),SF(50),CR(50),CAP,CINS,
       LIFE,DEBT,EQTY,DINT,EINT,TAX,TXCR,IDEPM,XFIX

COMMON NITER,D,VRDVM,CPPR,ETAT,NG,VR,QR,GR,QG,ETAGO,HR,CLRNCE,
       TERFAC,HO,RHO,VMEAN,VTIP,NERROR,PD,LSTITD

COMMON VW(90),LAMBDA(90),CP(90),ETAGEN(90),FLOAD(90),PE(90),
       CPMIN,PGMIN,
       VCI,VCO,VWDVO,TR,TCO,E,IVCO,IVCI,TCI,LSTITP,EW,ETASYS,
        PW(90),ETATR(90)

COMMON VO(90),CP1(30),CT1(30),LAMDA1(30),LVW,LCP1,LCT1,LLAM1

COMMON VAL(3),VALMAX(3),VALMIN(3),
       NOPT,KM,K,TLN,DELX,DELY,X(3),Y(4),F(4),G(4),IVCON

COMMON HT,CTPR,TATR,IOPT
REAL MILPKW,NU
REAL NG
REAL LAMDA1
REAL LAMBDA
N=LIFE
CLIFE=FLOAT(LIFE)

8
R = EQTY * EINT + (DEBT * DINT)*(1.-TAX)

SFF = ((1.+R)**N - 1.) / R
SFR = DEBT * CAP
CTC = TXCR * CAP
GO TO (10,80,160) , IDEPM

10
IF (TXCR.GT.0.) GO TO 20

DEP(1) = CAP/CLIFE

GO TO 40  -->  2
               A1

20
DEP(1) = 0.
```

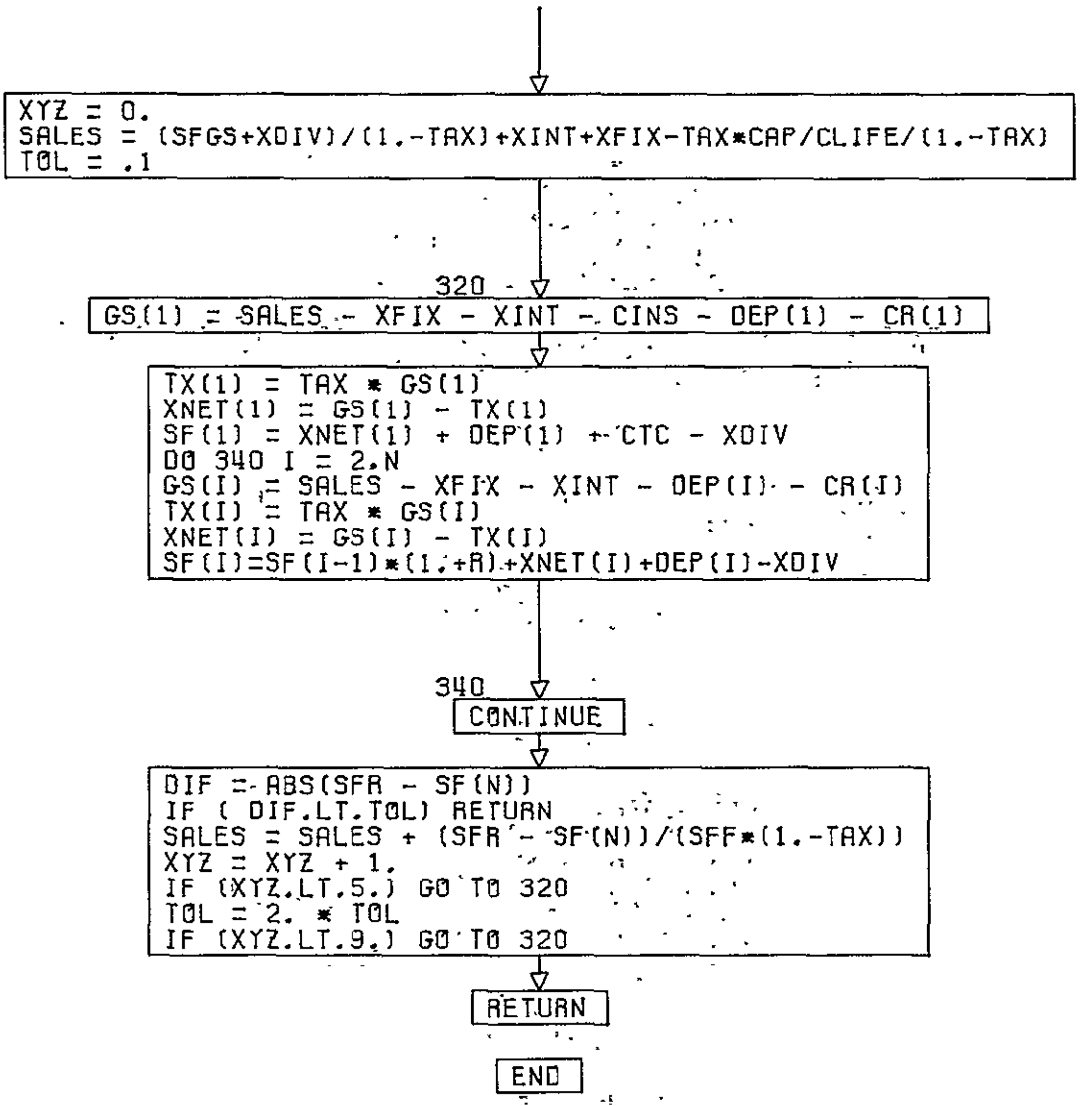
CONT. ON PG 2


1
A1
40
DO 60 I = 2,N
DEP(I) = (CAP-DEP(1)-CTC) / (CLIFE - 1.)
60
CONTINUE
GO TO 300
3
A2
80
IF (TXCR.GT.0.) GO TO 120
SUM = (CLIFE + 1.) * CLIFE / 2.
DO 100 J = 1,N
XJ=FLOAT(J)
DEP(J) = (CLIFE + 1. - XJ) * CAP / SUM
100
CONTINUE
GO TO 300
3
A2
120
SUM = CLIFE * (CLIFE - 1.) / 2.
DEP(1) = 0.
L = N-1
DO 140 I = 1,L
J = I+1
XI=FLOAT(I)
DEP(J) = (CLIFE-XI)*(CAP-CTC)/SUM
140
CONTINUE
GO TO 300
3
A2
160
IF (TXCR.GT.0.) GO TO 180

CONT. ON PG 3

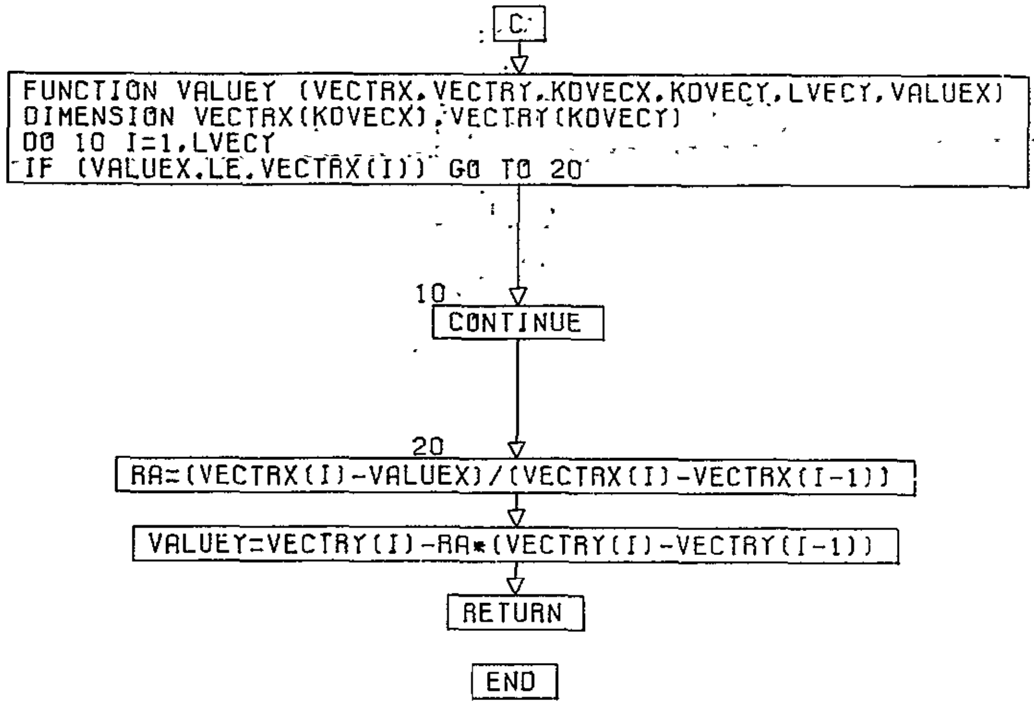

XPER = 2./CLIFE
DEP(1) = XPER * CAP
GO TO 200
180
XPER = 2./(CLIFE-1.)
DEP(1) = 0.
200
BAL = CAP - CTC - DEP(1)
DO 240 J = 2,N
REM = N - J + 1
AAA = XPER * BAL
BBB = BAL/REM
IF (AAA.GT.BBB) GO TO 210
DEP(J) = BBB
GO TO 220
210
DEP(J) = AAA
220
BAL = BAL - DEP(J)
240
CONTINUE
2
2
2
A2
300
XINT = DINT * DEBT * CAP
XDIV = EINT * EQTY * CAP
SFGS = SFR / SFF

CONT. ON PG 4


XYZ = 0.
SALES = (SFGS+XDIV)/(1.-TAX)+XINT+XFIX-TAX*CAP/CLIFE/(1.-TAX)
TOL = .1
320
GS(1) = SALES - XFIX - XINT - CINS - DEP(1) - CR(1)
TX(1) = TAX * GS(1)
XNET(1) = GS(1) - TX(1)
SF(1) = XNET(1) + DEP(1) + CTC - XDIV
DO 340 I = 2,N
GS(I) = SALES - XFIX - XINT - DEP(I) - CR(I)
TX(I) = TAX * GS(I)
XNET(I) = GS(I) - TX(I)
SF(I)=SF(I-1)*(1.+R)+XNET(I)+DEP(I)-XDIV
340
CONTINUE
DIF = ABS(SFR - SF(N))
IF (DIF.LT.TOL) RETURN
SALES = SALES + (SFR - SF(N))/(SFF*(1.-TAX))
XYZ = XYZ + 1.
IF (XYZ.LT.5.) GO TO 320
TOL = 2. * TOL
IF (XYZ.LT.9.) GO TO 320
RETURN
END

```
C

DIMENSION PLOAD(10),SLP(10)
FUNCTION ETAG(PR,PLD)
REAL LOSSES,MECH
DATA KOPLD,KOSLP/10,10/
DATA LSLP/8/

DATA OUTPUT,SLOPE,SCR,STATOR,ROTOR,STRAY,MECH,CORE/
     125.,.4142,.55,2.4,1.25,1.25,1.5,1.25/

DATA COSTH/1./

DATA PLOAD(1),PLOAD(2),PLOAD(3),PLOAD(4),PLOAD(5),PLOAD(6),
     PLOAD(7),PLOAD(8)
         /0.00,.1,.2,.3,.4,.5,1.0,1.5/

DATA SLP(1),SLP(2),SLP(3),SLP(4),SLP(5),SLP(6),
     SLP(7),SLP(8)
         /100.E-6,4.3E-5,2.5E-5,1.7E-5,1.3E-5,1.2E-5,1.E-5,1.E-5/

C CALCULATE EFFICIENCY FOR 100KW GENERATOR INPUT

PIF=SLOPE*PLD+SCR
STTR=STATOR*PLD*PLD
RTR=ROTOR*PIF*PIF
STRY=STRAY*PLD*PLD
LOSSES=STTR+RTR+CORE+STRY+MECH
OPUT=OUTPUT*PLD*COSTH
ETAG=1.-LOSSES/(OPUT+LOSSES)

C INTERPOLATE SLOPE OF EFFICIENCY VS. POWER LINE (AS FUNCTION OF PLOAD)

SLPG=VALUEY(PLOAD,SLP,KOPLD,KOSLP,LSLP,PLD)
ETAG=ETAG+SLPG*(PR-125.) -.01

RETURN

END
```

```
C

FUNCTION ETATRN(ETG,PLD)
ETATRN=(PLD/ETG)/(PLD/ETG+.04433)

RETURN

END
```



*USGPO: 1977 — 757-139/6309 Region 5-II
PG 1 FINAL