NASA-CR-134937

DOE/NASA/9404-76/2
NASA CR-134937
R-1382

DESIGN STUDY OF WIND TURBINES 50 kW TO 3000 kW FOR ELECTRIC UTILITY APPLICATIONS

Analysis and Design

See also:
NASA-CR-134936
for Executive Summary

Kaman Aerospace Corporation

February 1976

Prepared for
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
Under Contract NAS 3-19404

for
U.S. DEPARTMENT OF ENERGY
Office of Energy Technology
Division of Solar Energy

DOE/NASA/9404-76/2
NASA CR-134937
R-1382

DESIGN STUDY OF WIND TURBINES 50 kW TO 3000 kW FOR ELECTRIC UTILITY APPLICATIONS

Analysis and Design

Kaman Aerospace Corporation
Bloomfield, Connecticut 06002

February 1976

Prepared for
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135
Under Contract NAS 3-10094

for
U. S. DEPARTMENT OF ENERGY
Office of Energy Technology
Division of Solar Energy
Washington, D. C. 20545
Under Interagency Agreement E(49-26)-1010

Page intentionally left blank

Page intentionally left blank

FOREWORD

This report summarizes the results of a study of wind energy conversion systems for electric utility applications. The 9 month program was conducted under contract NAS3-19404, awarded by NASA Lewis Research Center in November 1974, as a part of the Energy Research and Development Administration's Federal Wind Energy Program. Mr. John Sholes was the NASA Project Manager and provided valuable review and guidance throughout the program.

Special acknowledgement is made to Northeast Utilities and their personnel who assisted in the project. Mr. Michael Lotker served as the project coordinator for Northeast Utilities. Valuable assistance was also rendered by the Colorado Springs Public Utilities and by the Connecticut Department of Environmental Projection.

The Mueller Engineering Corporation and the Lightning and Transient Research Institute contributed critical specialized technical assistance. In addition, numerous equipment and structure suppliers furnished technical and cost data for the study.

Many personnel at Kaman Aerospace Corporation contributed to the study. Those who led the effort and wrote the final report were Messrs. Donald Brierley, Robert Collins, Thomas Cook, Herbert Gewehr, George McCoubrey, Richard Meier, Robert Paterson, Arved Plaks, John Schauble and Charles Wirth.

Page intentionally left blank

Page intentionally left blank

TABLE OF CONTENTS

TABLE OF CONTENTS (continued)

TABLE OF CONTENTS (continued)

TABLE OF CONTENTS (continued)

TABLE OF CONTENTS (continued)

TABLE OF CONTENTS (continued)

TABLE OF CONTENTS (continued)

TABLE OF CONTENTS (continued)

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (continued)

LIST OF ILLUSTRATIONS (continued)

LIST OF TABLES

LIST OF TABLES (continued)

1.0 SUMMARY

The objective of the contract was to develop optimized preliminary designs of low power (50 - 500 KW) and high power (500 - 3000 KW) Wind Generator Systems (WGS) for electric utility applications. The low power system is designed for a utility site with a yearly median wind speed of 5.4 m/sec (12 mph) and the high power WGS is designed for a utility site with an 8 m/sec (18 mph) median wind speed. The preliminary designs prepared in the study are intended to provide the bases for follow-on programs which will involve the detail design, fabrication and experimental demonstration testing of these units at selected utility sites.

The median wind speed and wind distribution at each site were specified by NASA. Also, NASA specified the environmental conditions at the sites, including the model to be used for wind gusts and the maximum wind speed of 54 m/s (120 mph). The NASA specification also defined significant features which the systems must have to assure their usefulness for the intended application, and the ranges of the parameters to be investigated during the design optimization task. NASA limited the study to horizontal axis, propeller type systems, directly connected to a power network (no storage), utilizing current state-of-the-art technology. WGS life goals were specified at 50 years for the static components, with 30 year dynamic component lives.

The program included four tasks: conceptual design; optimization; preliminary design; and a utility requirements evaluation. In the conceptual design task, several feasible WGS configurations were evaluated, and the concept offering the lowest energy cost potential and minimum technical risk for utility applications was selected. In the optimization task, the selected concept was optimized using a parametric computer program prepared for this purpose. In the preliminary design task, the optimized selected concept was designed and analyzed in detail. The utility requirements evaluation task examined the economic, operational and institutional factors affecting the WGS when connected to an electric utility network.

Results of the conceptual design task indicated that a rotor operating at constant speed, driving an AC synchronous or induction generator through a gear transmission is the most cost effective WGS configuration. The optimization task led to the selection of a 500 KW rating for the low power WGS and a 1500 KW rating for the high power WGS. It was also determined that these two machine designs could be installed at utility sites with yearly median wind speeds from 4 m/sec to 11 m/sec (8 to 24 mph), and provide energy at costs which approach those of machines optimized for each specific site. The preliminary design task produced a detailed refinement of the optimized selected concept, which utilizes a rotor with two variable pitch, filament wound composite blades mounted on a rigid hub, driving a standard AC synchronous or induction generator through a commercial gearbox. The system designs were prepared for both a conventional steel truss tower and a pre-cast, post-tensioned concrete shell tower. Conventional electromechanical primary controls are employed in the designs, with startup, shutdown, operational monitoring, failure reporting, data transmission and recording, and other sequencing and supervisory functions controlled by a microprocessor. The utility requirements analyses indicate that conventional electric utilities can operate and maintain WGS units with no substantial change in normal operating and maintenance procedures.

In summary, the work of this program concluded that wind generator systems can be designed in the dictated size ranges, using presently existing technology, to meet the operating environment and life requirements set forth by NASA. Moreover, when built in quantity, the WGS capital and energy costs can be economically competitive with alternative energy systems in certain applications. The low power, 500 KW system has a capital cost of $900 per kilowatt and an energy cost of 7¢ per kilowatt-hour. The 1500 KW high power WGS has a capital cost of $480 per kilowatt and an energy cost of 2.7¢ per kilowatt-hour. Note that this lower energy cost results primarily from the higher median wind speed for the 1500 KW system, rather than from scale factor alone. These costs include all equipment and operations necessary to install the WGS and connect the system to a utility distribution line based on a production lot of 1000 units. Complete details are presented in the remainder of this volume.

2.0 INTRODUCTION

The objective of the program reported in this document was to develop optimized preliminary designs of low power and high power Wind Generator Systems (WGS) for electric utility applications. The work encompassed several tasks covering the conceptual study, optimization and preliminary design of these systems and the economic, operational and institutional issues involved in their application. The preliminary designs prepared in the study are intended to provide the bases for follow-on programs which will include detail design, fabrication and experimental demonstration testing of these units at selected utility sites. This introduction acquaints the reader with the background leading to the program, outlines the basic guidelines and approach, and describes the relationship of the various program tasks to the results obtained.

2.1 Background

Although man has harnessed wind energy for a variety of uses in the past, including transportation, industrial and consumer applications, recent decades have seen a decline in utilization of this basic energy source due to competition from alternative energy sources, particularly petroleum. As energy demand increased and potential shortages of fuel became apparent, interest in energy sources other than fossil fuels and nuclear energy began to rise, particularly with the advent of the oil embargo of 1973 and the four-fold price increase of imported oil.

These recent trends and events have indicated the advisability of re-examining wind energy as a potential part of the world energy supply. This examination currently includes a major development effort by the Energy Research and Development Administration, delegating to the National Aeronautics and Space Administration, the development of large utility-compatible wind generator systems. The study reported here, which is part of the overall NASA program, is designed to provide the basis for the first industry-built, utility-operated experimental demonstration wind generator systems.

This facet of the NASA program will draw upon two other elements of NASA's effort: the experience gained from the fabrication and operation of a 100 KW experimental wind generator system which NASA has recently put into operation; and the supporting research and technology (S R & T) programs which will examine technology pertinent to wind energy conversion systems and related components. In concert, these elements of the NASA program are designed to demonstrate cost-competitive wind energy conversion system units before 1980.

2.2 Guidelines and Approach

In order to be adopted by electric utilities in the United States, wind generator systems must be economically competitive and electrically compatible with other forms of power generation. As mentioned above, efforts in the recent past to develop wind generator systems, such as the famous Smith-Putnam wind turbine developed in the early years of World War II, failed because their high capital costs could not be amortized against fuel savings at the then current prices for fossil fuels.

Although fuel costs have increased dramatically in the recent past, the economic benefits of wind energy conversion systems have not changed in proportion due to the parallel increase in the cost of capital. The problem facing the development of such systems today is the same as in the past; i.e., trading off the higher capital costs of wind generator systems with the potential savings in fuel costs. Dominating the approach to this study, therefore, was the recognition that all decisions, beyond meeting the basic technical requirements of the wind generator system, would have to be made on the basis of minimizing the capital cost and the energy cost of the system. This was the guiding precept of the study.

In addition to this principal guideline, several ground rules were established by NASA for the study. These include:

1. Consideration of only the propeller-type horizontal axis rotor

2. Maximum use of current state-of-the-art technology

3. Direct connection to a standard electric utility network (no storage)

4. Fully automatic operation, with remote dispatch capability.

Kaman adopted several additional guidelines to assure that the systems, as designed, would be both economical and practical. These included:

1. Off-the-shelf components and associated fabrication technology, wherever possible

2. Application of current low cost rotor fabrication technologies being developed by Kaman and other helicopter contractors

3. Rotor and control system design utilizing the experience residing in the rotary wing aircraft industry

4. Close coordination with an operating utility to insure that the evolving design would meet the requirements, both technical and economic, of a typical electric utility

5. Use of subcontractors, consultants and advisors with expertise in critical technical and institutional aspects of wind energy conversion systems to assure that these issues would be adequately addressed in the formulation of the system designs.

The above Kaman approach proved beneficial in guiding the designers and analysts evolving the WGS designs. Use of the latest rotor fabrication technology insured that the lowest cost fabrication approach was adopted for the wind turbine blade. A major effort in the helicopter industry today is the pursuit of cost reductions in the fabrication and life-cycle maintenance of helicopter components, particularly through the widespread application of composite materials and automated fabrication processes.

Application of the experience base in rotor design and control systems developed for rotary wing aircraft is also a logical approach, since helicopter rotors and related control systems are the closest technological analogies to wind turbines available. For example, incorporating in the WGS blade the common helicopter practice of balanced aerodynamic center/center-of-gravity, to insure adequate blade stability, is a logical extension of helicopter experience.

Close coordination with an operating utility was provided by the Northeast Utilities Company of Newington, Connecticut, the major southern New England electric utility company. Northeast Utilities also provided direct technical assistance in the selection of appropriate protective and interface equipment, performing on-line operating stability analyses of the WGS and conducting economic studies of wind energy conversion systems in utility applications.

Use of expert subcontractors, consultants and advisors ensured that expertise in the particular technical or institutional area involved was applied to the WGS system designs. The Mueller Engineering Corporation of West Hartford, Connecticut, furnished basic consulting services for tower design studies and assistance in defining electrical equipment requirements. The Lightning and Transient Research Institute of St. Paul, Minnesota, provided design guidance and review for the lightning protection provisions so important to the WGS unit. As advisors, the Colorado Springs Public Utilities Company of Colorado Springs, Colorado, provided an additional operating utility evaluation of the required design features, thus adding the valuable perspective of another utility. The Connecticut Department of Environmental Protection furnished a preliminary assessment of the potential environmental impact of wind generator systems and the associated requirements for site approval, licensing requirements and other institutional issues that must be faced by this type of power generation facility. All of these organizations contributed substantially to the final system designs.

2.3 Program Description

Because of the broad scope of the study, it is useful to describe the program plan to orient the reader toward the material presented. NASA directed that the work be carried out in three phases, shown in Figure 2-1 and described below.

Phase I of the program consisted of two major tasks:

- Conceptual design
- Development of a parametric computer program

The objective of the conceptual design task was to select the best concept for the WGS, starting with a review of feasible WGS system design alternatives within the NASA and Kaman guidelines. As described in Section 3 of the report, various combinations of fixed pitch and variable pitch rotors, AC and DC generators, tower configurations, transmission system components, etc., were examined to determine their suitability for the intended application. Based on estimates of sizes, weights, efficiencies and costs of approximately eight different

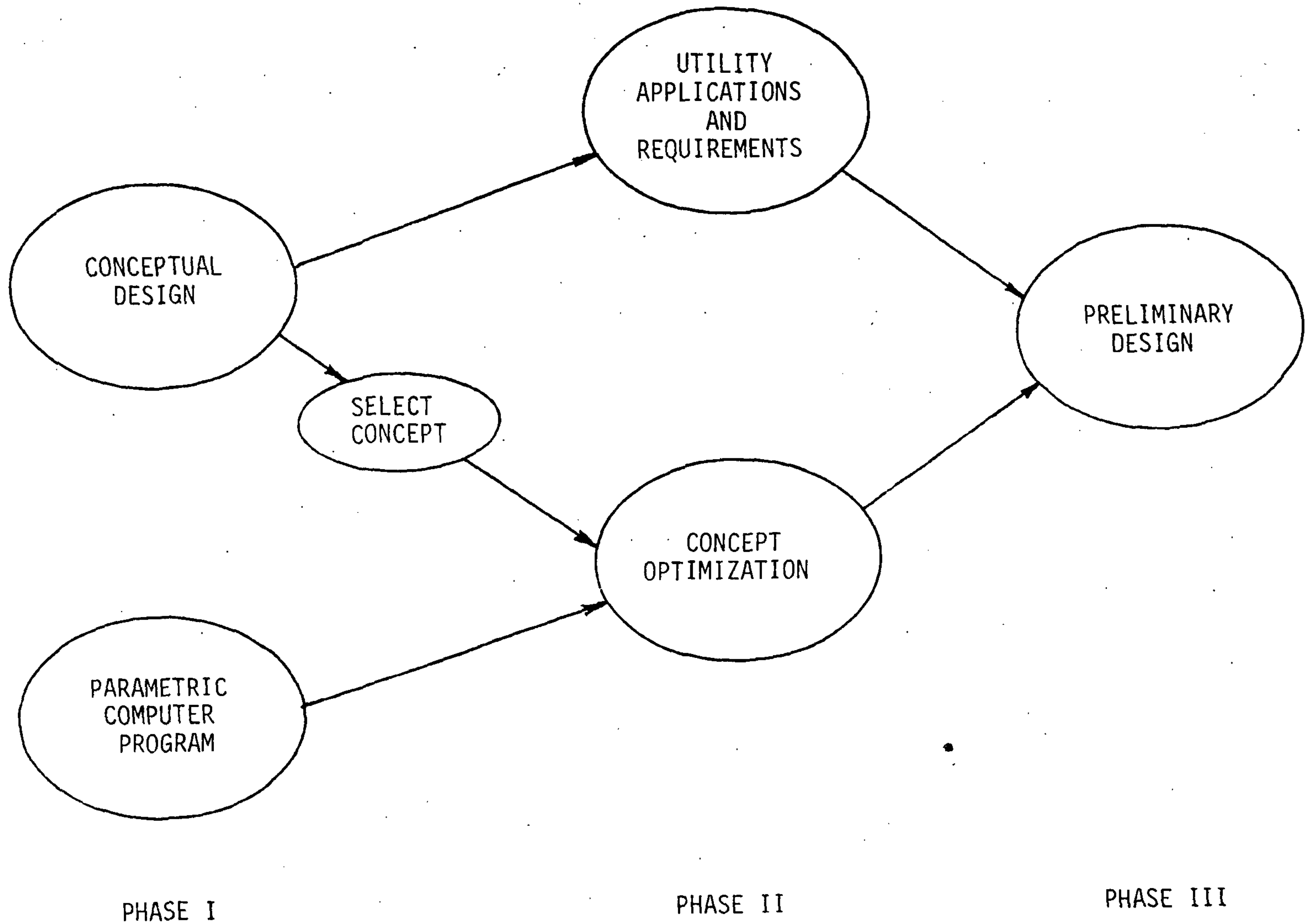


Figure 2-1. Study Program Summary

feasible configurations, three final candidates were selected for conceptual design. After these three systems were evaluated in more depth, the final system configuration was selected.

In parallel with the conceptual design task, a parametric computer program was prepared for use during the next phase in optimizing the parameters of the selected concept.

Phase II of the program consisted of two tasks, conducted in parallel:

- Concept optimization
- Study of utility applications and requirements

Concept optimization utilized the parametric computer program developed in Phase I to systematically examine numerous parameters involved with the rotor, drive system and electrical system, to arrive at an optimum configuration for the selected concept. The primary optimization parameter was energy cost (in ¢/kilowatt-hour), with capital costs, technical risk and other operational factors being important secondary considerations. At the conclusion of this task, the specific component sizes (500 KW and 1500 KW) and characteristics of the WGS were established for the preliminary design phase.

A parallel study of utility applications and requirements examined both the technical interface requirements of the WGS with a utility system and the economic requirements imposed on such a system in preparation for the preliminary design phase. This study also explored many of the institutional issues that may be faced by the WGS prior to its widespread adoption by electric utilities.

The results of the optimization study in Phase II indicated that wind turbine rotor solidity, or blade area, was a critical economic parameter. In order to minimize energy costs while meeting the basic technical requirements of the system, a separate study of the rotor diameter/solidity effects on WGS design and economics was conducted prior to initiating the preliminary design effort. Results of this analysis were used to modify the parametric computer program and, subsequently, reoptimize the selected concept.

Phase III of the program consisted of the preliminary design of the selected and optimized WGS configuration.

The preliminary design effort produced drawings and specifications, with supporting analyses, for a 500 KW, 150 foot diameter rotor, low power wind generator system, and a 1500 KW, 180 foot diameter rotor, high power wind generator system. The low power system is designed for a utility site with a yearly median wind speed of 5.4 m/sec (12 mph) and the high power WGS is designed for a utility site with an 8 m/sec (18 mph) median wind speed, in accordance with NASA specifications.

This preliminary design information is intended by NASA to be used as the basis for a subsequent contract involving the detail design, fabrication and experimental testing of the first industry-built, utility-operated, full-scale wind generator system. Hence, considerable detail information concerning the preliminary design results is included in this report.

2.4 Report Organization

The above outline of the program was described in chronological order. However, for the purposes of this report, it was deemed desirable to describe the study by subject, and to cover each subject in general chronological order through the several tasks and phases. This report, then, is organized to present the following subjects:

- Systems Analyses
- Subsystems
  - Rotor
  - Controls
  - Structure
  - Drive
  - Electrical
- Utility Applications and Operational Requirements

Each of the subsystem sections is divided into two major parts, the design and analysis effort leading up to the selection of the final subsystem configuration, and the preliminary design and supporting analyses of the specific selected systems and subsystems. The material has been organized into an evaluation section and a design description section, effectively conveying the results of two major objectives of the study:

1. To select and optimize the best WGS configurations for electric utility applications
2. To provide preliminary designs of the optimized concepts for subsequent fabrication and testing.

3.0 SYSTEM ANALYSES

The system analyses conducted in the WGS study integrated the results of the design and supporting analysis tasks conducted for each of the WGS subsystems. This was accomplished by providing initial guidance for the subsystem designs, evaluating the results of the subsystem efforts with respect to the overall WGS and by identifying where refinements in the subsystem design should be directed. Thus, the system analyses provided both guidance and evaluation functions for the subsystem design tasks, insuring that the results of the study met the prime contract goal: preliminary designs of optimized, cost-competitive wind generator systems for electric utility use.

3.1 Approach

The WGS system analyses consisted of four steps:

1. A concept formulation step where a limited number (8 - 10) of feasible, competitive WGS concepts were defined from among many possible system component combinations

2. A concept evaluation step where these defined concepts were reduced to a limited number (3) of concepts for which designs and analyses were conducted and a final concept selected

3. An optimization step where the selected WGS concept was optimized

4. A design evaluation step where details of the evolving preliminary design were evaluated in terms of their impact on overall system cost effectiveness.

The concept formulation step was accomplished by examining many past and proposed WGS concepts and components in view of the requirements and restrictions of the current study. This step started with the definition provided by NASA Lewis Research Center Wind Power Office in their contract Statement of Work. The Statement of Work section describing the essential requirements of the WGS is presented in Appendix A.

Capsuling the description of the WGS in Appendix A is the general system schematic shown in Figure 3-1. The WGS meeting the requirements given in Appendix A consists of an energy conversion section (rotor, transmission, generator, tower, etc.), a control and protective equipment section (lightning protection, controls, suppressors, etc.), and a utility interface section (transformers, switchgear, etc.). The schematic of Figure 3-1 shows that not all components of each section are physically grouped together. However, they all are necessary to meet the system requirements and must be evaluated as a whole.

Many potential candidate components (for example, vertical shaft turbines) and concepts (DC generators connected directly to the power output line) were not considered in the concept formulation step because they were either outside the scope of the study (vertical axis turbines) or did not meet the defined NASA requirements (DC generator output is not compatible with standard utility power lines).

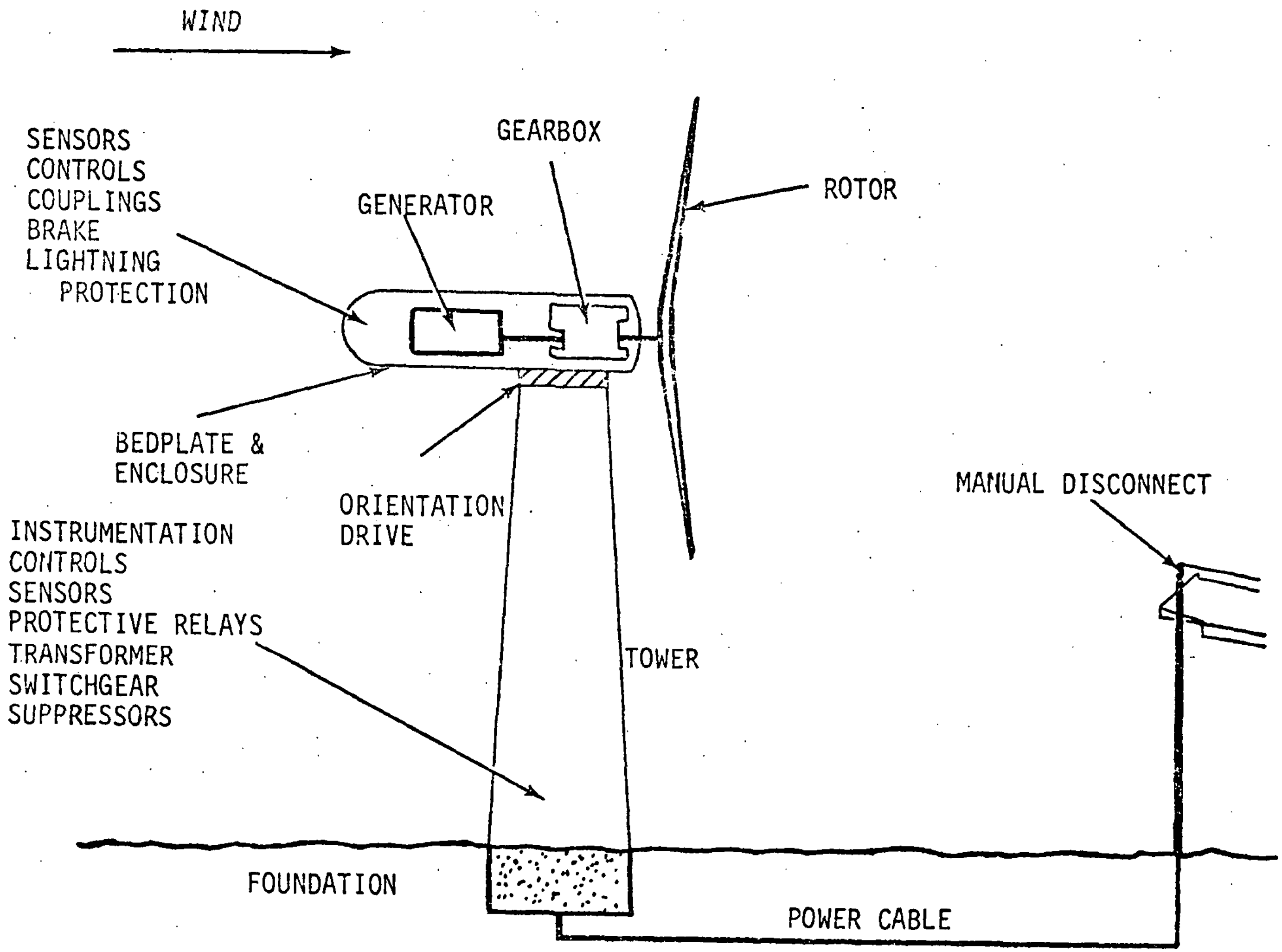


Figure 3-1. Typical Wind Generator System Elements

Other component and system concepts were eliminated because they did not appear to offer any advantages. For example, some weight on the system tower can be saved by having conversion machinery on the ground rather than at the top of the tower. Also, the slipring which transfers generator output power from the rotating nacelle to the stationary tower could be eliminated. However, this configuration requires a high torque 100° - 105° angle gearbox to change direction from the rotor shaft axis to the tower axis. If such gearboxes were presently available, they would be very heavy, negating the weight reduction effect sought. Using a smaller speed-increasing, angle gearbox, the generator could be mounted inside the top of the tower, below the turntable, thereby eliminating the long shafting in the tower. Speed-increasing gearboxes of the required configuration are not presently available. Therefore, it was concluded that the most direct approach, offering the least development risk, was to place all components atop the tower inside the nacelle.

The foregoing is typical of the number and complexity of factors considered in many of the decisions relating to the concept selection process, some of which could be treated analytically, while others were resolved by engineering judgement.

The information presented in the remainder of this section primarily covers the last three steps listed above. The concept evaluation step, which was part of the conceptual design task of Phase I, reduced the prime candidate concepts from 8 to 3 to the final configuration. The optimization step, conducted in Phase II of the study, utilized the parametric model developed in parallel with the conceptual design task of Phase I. Finally, the design evaluation step, an adjunct to the Phase III preliminary design task, examined the characteristics and flexibility of the selected preliminary designs.

As is the case with all extensive analyses of complex systems with multiple evaluation criteria, these steps did not stay neatly confined to each phase of the study. The evaluations and conclusions concerning concepts, components and utilization often were done on different bases for system rated power level, rated wind speed and other parameters, depending on what point in the study was reached at that time. However, it is believed that all of the conclusions reached are valid, and that repeating some of the analyses with revised, updated bases for system parameters would not alter the conclusions reached.

3.2 Concept Evaluation

As stated above, an evaluation of the WGS concepts was conducted after the conceptual formulation task evolved 8 prime system candidates from the many other approaches considered. The purpose of the concept evaluations was to evaluate these candidates in a consistent manner, so that the design effort could be directed at the preferred configuration.

Different rotor, transmission and electric components were arranged into 8 configurations, from which all the major component options could be evaluated. In this analysis, tower configurations were not evaluated. It was determined that the tower configurations could be selected almost independently of the Rotor, Drive and Electrical Subsystems. The tower selection analysis was more effectively accomplished by the independent design analysis covered in Section 6.

3.2.1 Candidate Components

The component concepts considered for the conceptual evaluation analyses are described below. These components were considered in various combinations for the WGS application. Since constant frequency power must be delivered to the utility, those systems using a variable speed drive rotor must provide some mechanical or electrical means of replacing the standard, constant speed, constant frequency AC generator usually used in utility generating plants.

Rotor - The rotor concepts considered are characterized by their operating mode and method of torque regulation. The operating mode may be either constant or variable speed. Torque regulation requires some device by which the limit torque is maintained.

In the variable speed mode, the rotor is operated at its peak aerodynamic efficiency or peak fraction of wind energy captured, up to the limit of the drive and electrical components of the WGS. For wind powered rotors, the aerodynamic efficiency varies uniquely with velocity ratio, V_{TIP}/V_{EFF} in a bell-shaped curve (see Figure 3-9). For a given geometry, the velocity ratio is equivalent to RPM/V_{WND} ratio, and thus a rotor can be operated at peak efficiency by simply varying RPM to give the RPM/V_{WND} ratio corresponding to the peak efficiency. A variable speed rotor is usually of the fixed pitch type which simplifies the rotor design.

A constant speed mode is desirable to simplify the Drive and Electrical Subsystems by delivering power at a constant rpm. Once the rpm is selected, the rotor will operate at maximum efficiency at only one wind speed and thus, the power output at other speeds will suffer a penalty. A constant speed rotor can drive directly either an AC synchronous or induction generator connected with the utility network. The utility network provides a synchronizing torque through the generator, which forces the rotor to operate at constant speed.

For either a constant speed or variable speed rotor system, some means of controlling rotor torque is necessary. The WGS is usually designed to some rated wind speed, where it generates its rated power output. Above this rated wind speed, the rotor is capable of extracting more power than the Drive and Electrical Subsystems can handle. Torque control above rated wind speed, for example, can be provided by drag devices, blade pitch variation or variable diameter.

Blade pitch control is used to reduce aerodynamic efficiency at some wind speeds to regulate the power output. The complexity of a pitch change mechanism is introduced in this concept. This mechanism may consist of direct mechanical control of the blade root end or a mechanism by which pitch control is effected by movement of an aerodynamic flap located near the blade tip. The latter approach was selected in the conceptual design task for the variable pitch systems.

Drag flaps or aerodynamic spoilers may also be used to control rotor net torque above rated wind speed. If this approach is used with a fixed pitch rotor, then the benefit of removing the pitch changing mechanism is replaced partly by the drag flap system. The variable diameter concept controls rotor size, and thus, energy absorbed, to regulate power. The benefit due to the absence of a pitch changing mechanism is replaced by a rotor blade retraction system.

Variable speed rotors must also be provided with some device to limit torque or power above rated wind speed. The same devices used for the constant speed rotor can be used, but since one advantage of a variable speed system is the elimination of a pitch control mechanism, the only torque control device that appears worthwhile would be some form of drag device or lift spoiler.

Complete descriptions of these rotor options may be found in Section 4, Rotor Subsystem.

Drive System - For wind turbine applications, some means is required to increase the relatively low speed of the rotor (less than 50 rpm) to the relatively high speed of the generator (greater than 1000 rpm). The simplest, least costly way to step up speed is through a fixed ratio gearbox. Other types of drive systems which have belts or chains have capacity limitations for the WGS application or are too costly and inefficient.

A variable speed step up ratio for use with a variable speed rotor can be obtained by coupling a hydrostatic drive to a fixed ratio gearbox. This drive system, though inefficient, allows simplification of the electrical system when the rotor is operated at variable speeds. Here the drive system delivers power to the generator at constant rpm, permitting the use of an AC synchronous generator.

A two step ratio gearbox is costlier than a fixed gear ratio gearbox, and has size limitations. Nevertheless, it was considered for use with an AC synchronous generator. The two step gearbox allows operation of the rotor at two speeds, thus reducing the performance penalty due to off-optimum rotor speed operation of the constant speed rotor.

A complete description of these concepts is presented in Section 7, Drive Subsystem.

Electrical Components - The WGS electrical machinery requirement is met best by the AC synchronous or induction generators. Early investigation indicated an industry preference for AC synchronous generators and thus it was selected for this study. However, it was noted that should induction generators be found acceptable, they could be substituted with some cost benefit.

For concepts with rotors operating at variable speed (and fixed ratio transmissions), the conversion of power to the desired constant frequency output can be obtained in several ways. Those considered are summarized in Table 3-1.

All of these approaches use the variable speed rotor to produce DC power. The DC is then converted into 60 Hz AC for delivery to the utility network. Each approach suffers from efficiency, weight and cost penalties over a straight AC generator approach, due to extra components. The benefit to be gained is the additional energy generated by the rotor operating at peak efficiency and the (usually) simpler rotor design, which must be traded off against the more elaborate Electrical Subsystem. A complete discussion of these options and their characteristics is given in Section 8, Electrical Subsystem.

TABLE 3-1. ELECTRICAL SYSTEMS FOR VARIABLE SPEED POWER INPUT

DC POWER SOURCE	DC TO AC CONVERSION
DC Generator	DC Motor Driving an AC Synchronous Generator
AC Synchronous Generator with Transformer and Rectifier	DC Motor Driving an AC Synchronous Generator
AC Synchronous Generator with Transformer and Rectifier	Solid State Inverter

3.2.2 Candidate System Concepts

Table 3-2 lists the eight candidate WGS concepts considered and evaluated. These are combinations of the component approaches described in the previous section, and represent the leading contenders derived from the conceptual formulation screening process.

Concept 1 served as a baseline configuration. It has a variable pitch rotor which drives the AC generator through a fixed ratio gearbox. Concept 2 differs from Concept 1 only in that rotor power output is increased at low wind speeds by operating the rotor at another, lower speed. This is accomplished with a more expensive two speed gearbox. Concept 3 differs from Concept 1 in that the pitch control mechanism is removed and the rotor is operated at fixed pitch. However, drag flaps are added for torque regulation. Concept 8 is similar to Concept 3, except that torque regulation is achieved by varying rotor diameter. All of these concepts have constant speed rotors.

The remaining Concepts have rotors operated at variable speeds and thus allow pitch to be fixed. In each case, drag flaps are used for torque control. For Concepts 4, 5 and 6, a fixed step up gear transmission is used with different electrical methods of obtaining constant frequency electric power output. Concept 7 uses a variable speed transmission to obtain constant speed to drive an AC synchronous generator.

3.2.3 Method of Evaluation and Sources of Data

For the purposes of obtaining a consistent evaluation of the candidate concepts, it was necessary to model the concepts to show correct relative differences. The method of evaluation of the candidate concepts parallels the final method explained in Section 3.3. The difference lies only in that, due to the fact that the mathematical model was not completed, assumptions for many of the power weight and cost relationships were of preliminary nature. However, the basic rotor aerodynamic efficiency map is the final version.

TABLE 3-2. CANDIDATE CONFIGURATIONS

CONCEPT	ROTOR	GEARBOX	GENERATORS
1.	VARIABLE PITCH CONSTANT SPEED	FIXED RATIO GEAR	AC SYNCHRONOUS
2.	VARIABLE PITCH TWO SPEED	TWO SPEED RATIO GEAR	AC SYNCHRONOUS
3.	FIXED PITCH CONSTANT SPEED	FIXED RATIO GEAR	AC SYNCHRONOUS
4.	FIXED PITCH VARIABLE SPEED	FIXED RATIO GEAR	DC GENERATOR DC MOTOR/GENERATOR
5.	FIXED PITCH VARIABLE SPEED	FIXED RATIO GEAR	AC SYNCHRONOUS TRANSFORMER AND RECTIFIER DC MOTOR/GENERATOR
6.	FIXED PITCH VARIABLE SPEED	FIXED RATIO GEAR	AC SYNCHRONOUS TRANSFORMER AND RECTIFIER SOLID STATE INVERTER
7.	FIXED PITCH VARIABLE SPEED	VARIABLE RATIO (HYDROSTATIC)	AC SYNCHRONOUS
8.	FIXED PITCH CONSTANT SPEED VARIABLE DIAMETER	FIXED RATIO GEAR	AC SYNCHRONOUS

The evaluation concentrated on the prime measure defined by NASA as the cost effectiveness yardstick; energy cost. However, it was recognized that capital cost of the WGS (in $/KW) and technical risk and development cost were also important factors, and were used to supplement the energy cost for concept selection. The method of calculating the energy cost and capital cost of each concept, as well as other technical parameters, is summarized below.

Rotor diameters were sized for a fixed rated wind speed or, in other words, the rotor diameter was calculated so that the power output rating of the system matched the specified system rating at the rated wind speed. The rated wind speed was selected to correspond to the speed at which yearly wind energy (wind speed^3 x hours per year of occurrence) is highest. This was, subsequently, proven to be a good approximation for the optimum rated wind speeds for the WGS sizes studied. Figure 3-2 shows a typical wind energy occurrence curve illustrating this value.

In the sizing process, particular attention was paid to those aerodynamic efficiencies which affect differences between rotor concepts. For example, variable speed rotors always operate at maximum aerodynamic efficiency when winds are below rated wind speed. On the other hand, constant speed rotors achieve maximum aerodynamic efficiency at only the design wind speed. In the WGS conceptual study phase, the rated wind speed was determined by maximizing wind energy output for a given yearly wind spectrum and median wind speed, as shown in Figure 3-2. Then maximizing <u>total</u> yearly energy output, the design wind speed for a given rotor diameter and solidity was determined. This analysis showed that design wind speed should be 7/8 of rated wind speed. Aerodynamic efficiency was maximized at the design wind speed.

For rotors thus sized, it was possible to determine the power output vs wind speed by accounting for differences in part power efficiencies between concepts. The energy output at any given wind speed is the product of wind occurrence frequency (in hours per year) and power output. The total yearly energy is simply the integral of energy over the operating envelope of the system.

Complete weight and cost estimates were determined for the baseline system. For all other candidate concepts, differences in component weights and costs were determined to maintain proper relationships. For the rotor, only the removal of pitch bearings was accounted for; the servo flap and drag flap being considered a tradeoff. However, a major cost increase was estimated for the retractable rotor. In the transmission and electrical system components, manufacturers' weight and cost estimates were used to assess differences. The tower cost changed only due to weight differences of components it had to support. This procedure led to estimates of total direct capital costs and energy cost for each system.

3.2.4 <u>Candidate Concept Evaluations</u>

The evaluation of the WGS concepts was conducted for systems sized for two conditions: a system designed to operate at a site with a 5.4 m/s (12 mph) median wind speed and having a power rating of 100 KW and another designed for an

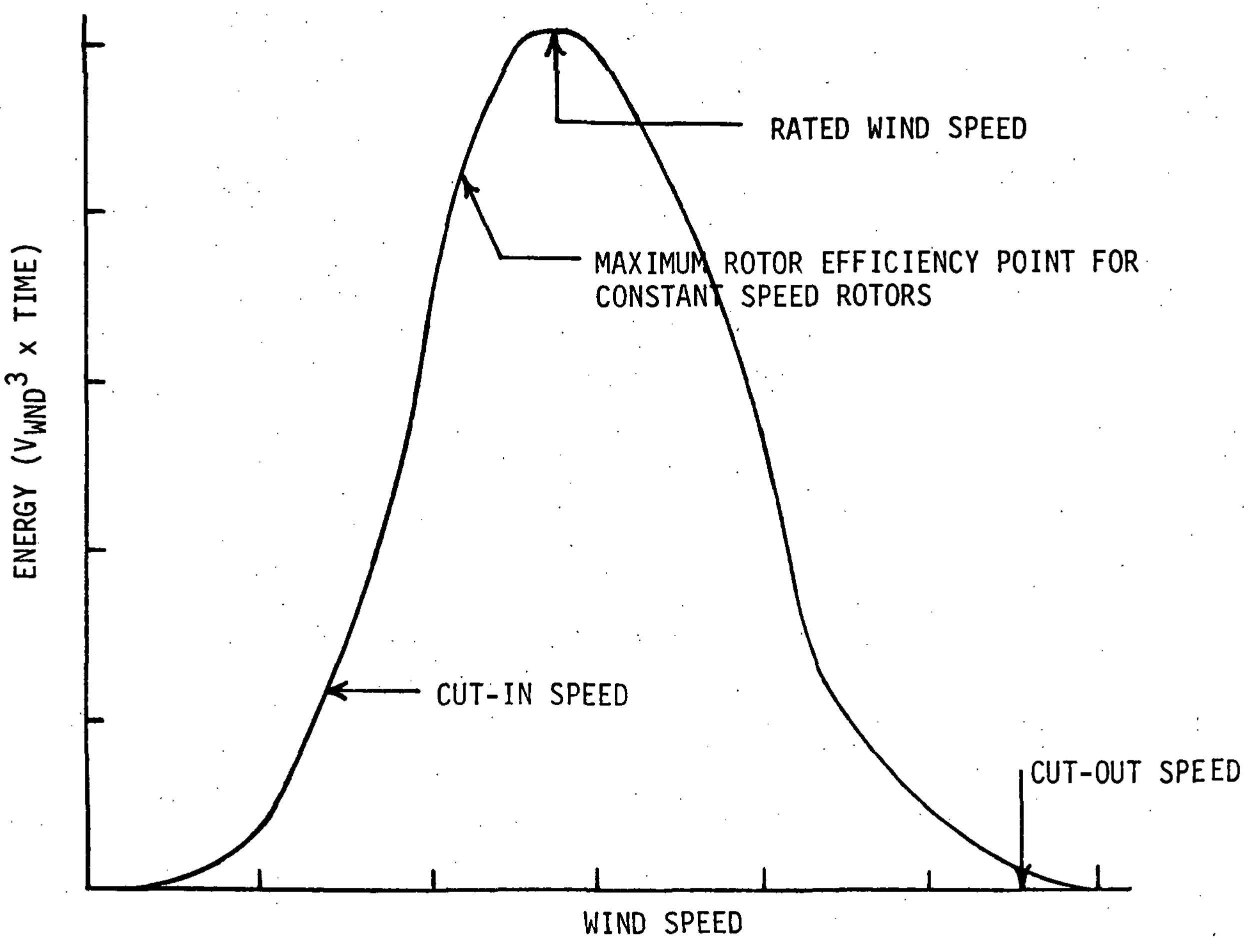


Figure 3-2. Wind Energy Frequency Distribution

8 m/s (18 mph) median wind speed site with a 1000 KW power rating. These were the baseline specifications for the low power and high power wind generator systems.

Figure 3-3 shows output power variation with wind speed for the various concepts studied. The slope of the power curve reflects the relative drop off of efficiency as the wind speed is reduced below rated. Variable speed rotor systems are seen to fare better below rated power because their rotors can operate at peak efficiency independent of wind speed.

Table 3-3 shows yearly energy output, rotor diameter and cut-in wind speed for the eight candidate systems. The energy output does not vary much from concept to concept because of their common rated wind speed and rated power. The rotor diameter reflects the efficiency of the system at rated power. The cut-in wind speed gives an indication of ease of starting.

In Table 3-4, the energy output, power output, direct capital cost and energy cost are normalized using the first concept as a baseline. Data shown are for the 1000 KW rated systems. However, the results with the 100 KW rated system are essentially the same. The tabular values are also shown graphically in Figure 3-4 for visual comparison of the systems. In Table 3-4, relative power rating is the rating that results when the rotors are scaled to the baseline system diameter. This normalization was used in recognition of the fact that rotors are the major contributing factor to WGS costs, but most difficult to scale. With all configurations having the same diameter, the basic rotor cost is now identical for them all. By holding the diameter constant, tower height also is constant and only needs cost and weight adjustments for the weight differences it must support.

3.2.5 Selected Configurations for Conceptual Design

Based on the results of the evaluation shown in Tables 3-3 and 3-4 and in Figure 3-4, three of the candidate WGS concepts were selected for further study. These were Concepts 1, 3 and 5, described in Table 3-2.

Figure 3-4 best summarizes the essential quantitative data used to make the selections for conceptual design. As can be seen in the figure, the constant rotor speed systems offer the lowest energy and capital cost candidates. This is primarily due to their simplicity, each using only a fixed gear transmission and synchronous generator to convert the rotor power into 60 Hz AC electrical power.

For the constant speed concepts (1, 2, 3 and 8), Concepts 1 and 3 were selected because they were competitive economically and inherently simpler than Concepts 2 and 8. Concept 2 also suffers from the requirement to operate the rotor at two different speeds, which is a very demanding technical requirement having potentially major impact on cost because of the technical risk involved. This facet of rotor design is discussed in Section 4, Rotor Subsystem. It involves the problem of tuning the rotor to avoid sustained operation at a resonant frequency which can lead to excessive fatigue damage or structural failure. Concept 8 was dropped because of high potential technical risk. Although variable diameter rotors have been built and operated successfully, the technical

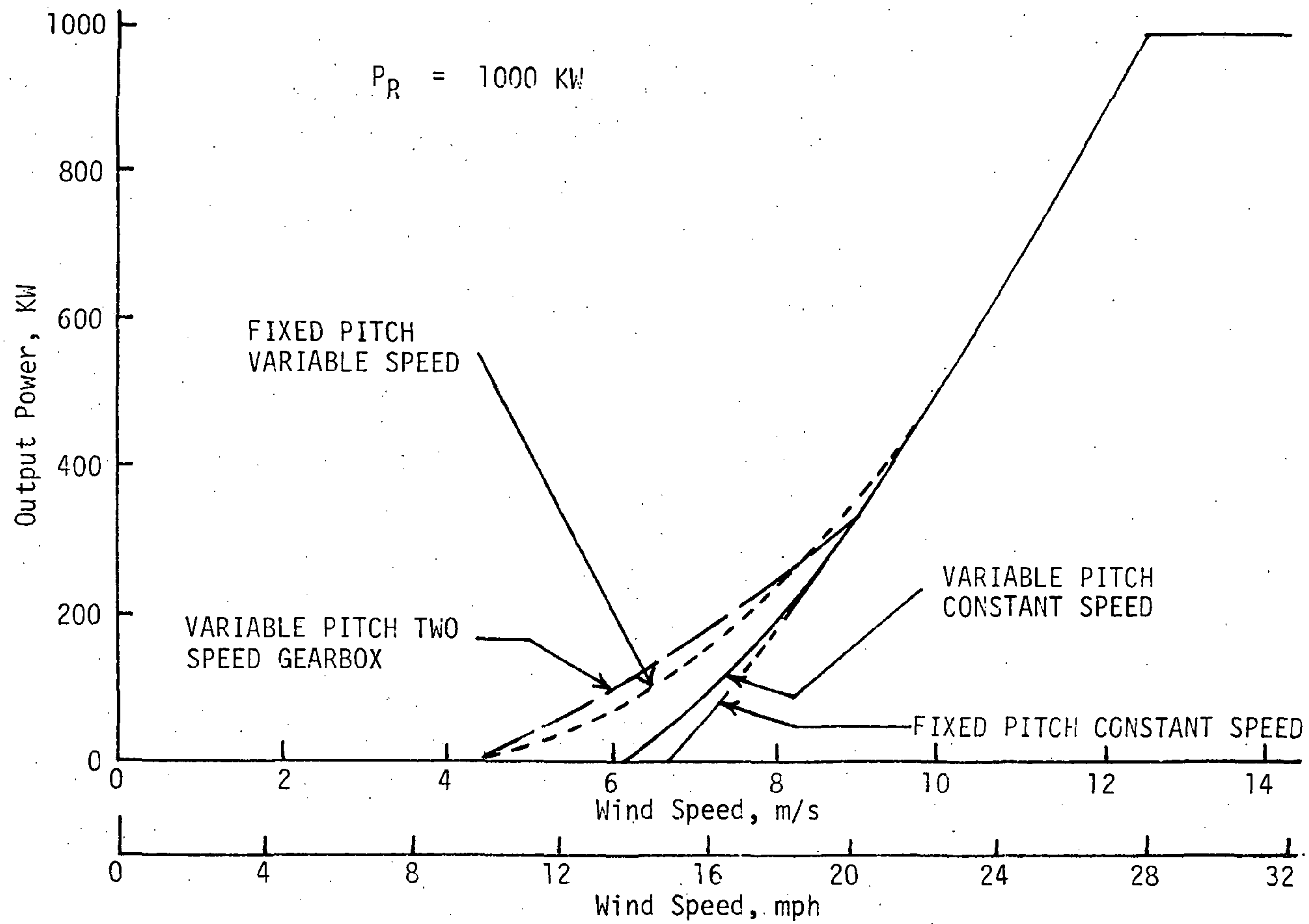


Figure 3-3. Power Output of Candidate Configurations

TABLE 3-3. CANDIDATE CONFIGURATION CHARACTERISTICS

CONCEPT	ROTOR GEARBOX GENERATOR	P_R = 100 KW 5.4 m/s (12 MPH) MEDIAN WIND SPEED 8 m/s (18 MPH) RATED WIND SPEED			P_R = 1000 KW 8 m/s (18 MPH) MEDIAN WIND SPEED 12.5 m/s (28 MPH) RATED WIND SPEED		
		YEARLY ENERGY KW-hrs	ROTOR DIAMETER m (ft)	CUT-IN V_{WND} m/s (mph)	YEARLY ENERGY KW-hrs	ROTOR DIAMETER m (ft)	CUT-IN V_{WND} m/s (mph)
1	VARIABLE PITCH/CONSTANT SPEED FIXED RATIO AC SYNCHRONOUS	325×10^3	29.3 (96)	4 (9)	3.04×10^6	43.3 (142)	6 (14)
2	VARIABLE PITCH/TWO SPEED TWO SPEED RATIO AC SYNCHRONOUS	346×10^3	29.3 (96)	3 (7)	3.29×10^6	43.3 (142)	4.5 (10)
3	FIXED PITCH/CONSTANT SPEED FIXED RATIO AC SYNCHRONOUS	325×10^3	29 (95)	4.5 (10)	3.04×10^6	43 (141)	7 (15)
4	FIXED PITCH/VARIABLE SPEED FIXED RATIO DC GEN/MOTOR/AC SYN	333×10^3	31.1 (102)	3 (7)	3.19×10^6	44.2 (145)	4.5 (10)
5	FIXED PITCH/VARIABLE SPEED FIXED RATIO AC/TR/MOTOR/AC SYN	333×10^3	31.4 (103)	3 (7)	3.19×10^6	44.8 (147)	4.5 (10)
6	FIXED PITCH/VARIABLE SPEED FIXED RATIO AC/TR/SOLID INVERTER	333×10^3	31.4 (103)	3 (7)	3.19×10^6	46.3 (152)	4.5 (10)
7	FIXED PITCH/VARIABLE SPEED VARIABLE RATIO AC SYNCHRONOUS	333×10^3	34.7 (114)	3 (7)	3.19×10^6	51.5 (169)	4.5 (10)
8	FIXED PITCH/VARIABLE DIAMETER FIXED RATIO AC SYNCHRONOUS	325×10^3	29 (95)	4.5 (10)	3.04×10^6	43 (141)	7 (15)

TABLE 3-4A. CANDIDATE CONFIGURATION CHARACTERISTICS NORMALIZED ON DIAMETER

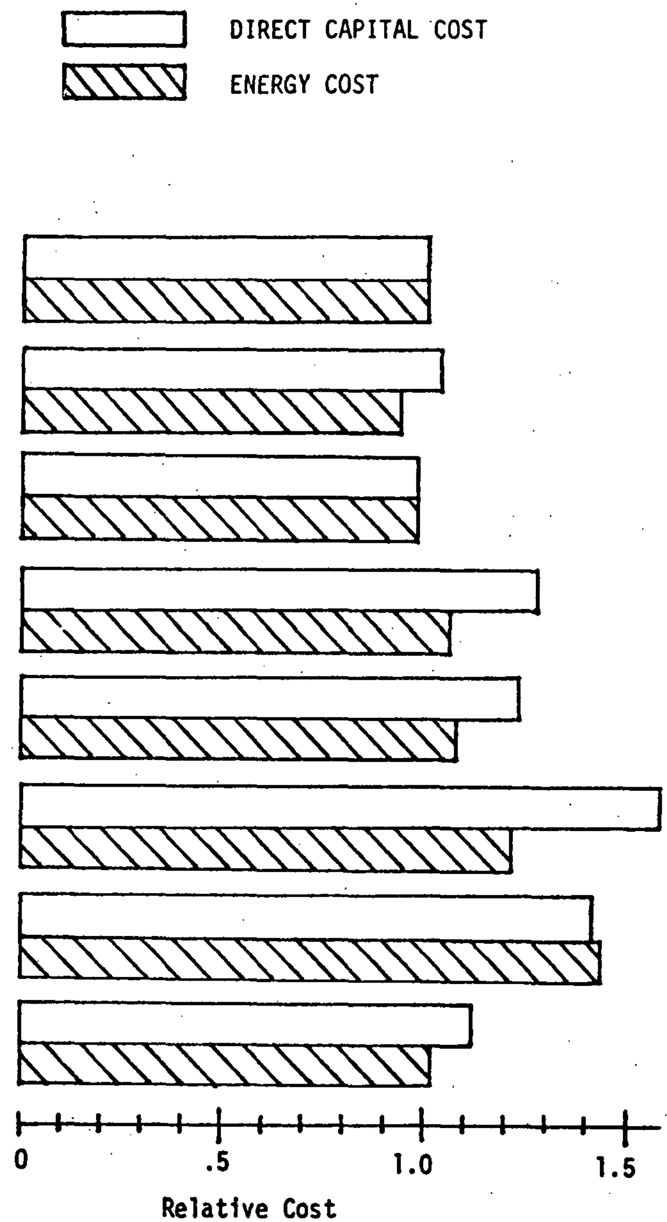
Diameter = 43.3m (142 ft)
8m/s (18 mph) median wind speed
12.5m/s (28 mph) rated wind speed

CONCEPT	ROTOR / GEARBOX / GENERATOR	Rated Power Kw	Yearly Energy output, kw-hrs	Capital Cost $	Yearly Cost $
1	Variable Pitch/Constant Speed Fixed Ratio AC Synchronous	1000	3.04×10^6	558×10^3	128×10^3
2	Variable Pitch/Two Speed Two Speed Ratio AC Synchronous	1000	3.29×10^6	574×10^3	130×10^3
3	Fixed Pitch/Constant Speed Fixed Ratio AC Synchronous	1011	3.07×10^6	552×10^3	127×10^3
4	Fixed Pitch/Variable Speed Fixed Ratio DC Gen/Motor/AC Syn	954	3.04×10^6	673×10^3	140×10^3
5	Fixed Pitch/Variable Speed Fixed Ratio AC/TR/Motor/AC Syn	928	2.96×10^6	638×10^3	126×10^3
6	Fixed Pitch/Variable Speed Fixed Ratio AC/TR/Solid Inverter	875	2.77×10^6	767×10^3	142×10^3
7	Fixed Pitch/Variable Speed Variable Ratio AC Synchronous	703	2.25×10^6	555×10^3	137×10^3
8	Fixed Pitch/Variable Diameter Fixed Ratio AC Synchronous	1010	3.07×10^6	631×10^3	132×10^3

TABLE 3-4B. CANDIDATE CONFIGURATION COMPARISON
(NORMALIZED ON ROTOR DIAMETER)

CONCEPT	ROTOR GEARBOX GENERATOR	RELATIVE ENERGY OUTPUT (KW-hrs/yr)	RELATIVE POWER RATING (KW)	RELATIVE DIRECT CAPITAL COST	RELATIVE ENERGY COST
1	VARIABLE PITCH/CONSTANT SPEED FIXED RATIO AC SYNCHRONOUS	1.0	1.0	1.00	1.00
2	VARIABLE PITCH/TWO SPEED TWO SPEED RATIO AC SYNCHRONOUS	1.08	1.00	1.03	.93
3	FIXED PITCH/CONSTANT SPEED FIXED RATIO AC SYNCHRONOUS	1.01	1.01	.98	.98
4	FIXED PITCH/VARIABLE SPEED FIXED RATIO DC GEN/MOTOR/AC SYN	1.0	.95	1.27	1.06
5	FIXED PITCH/VARIABLE SPEED FIXED RATIO AC/TR/MOTOR/AC SYN	.97	.93	1.23	1.08
6	FIXED PITCH/VARIABLE SPEED FIXED RATIO AC/TR/SOLID INVERTER	.91	.87	1.58	1.21
7	FIXED PITCH/VARIABLE SPEED VARIABLE RATIO AC SYNCHRONOUS	.74	.70	1.42	1.44
8	FIXED PITCH/VARIABLE DIAMETER FIXED RATIO AC SYNCHRONOUS	1.01	1.01	1.12	1.02

CONCEPT	ROTOR / GEARBOX / GENERATOR
1	VARIABLE PITCH/CONST. SPEED FIXED RATIO AC SYNCHRONOUS
2	VARIABLE PITCH/TWO SPEED TWO SPEED RATIO AC SYNCHRONOUS
3	FIXED PITCH/CONST. SPEED FIXED RATIO AC SYNCHRONOUS
4	FIXED PITCH/VARIABLE SPEED FIXED RATIO DC GEN/MOTOR/AC SYN.
5	FIXED PITCH/VARIABLE SPEED FIXED RATIO AC SYN/T&R/DC MOTOR/AC SYN.
6	FIXED PITCH/VARIABLE SPEED FIXED RATIO AC SYN/T&R/SOLID STATE INVERTER
7	FIXED PITCH/VARIABLE SPEED VARIABLE RATIO (HYDROSTATIC) AC SYNCHRONOUS
8	FIXED PITCH/CONST. SPEED/VARIABLE DIA. FIXED RATIO AC SYNCHRONOUS



* T&R = TRANSFORMER & RECTIFIER

Figure 3-4. Candidate Configuration Cost Comparison.

unknowns facing the design of a rotor typical of the size of the WGS were judged to exceed the risk guidelines of the study. The variable ratio gearbox concept was eliminated because of typically poor efficiency of hydrostatic drives, the type of drive offering the highest torque capacity for this kind of application, and the unavailability of these units in the required torque ranges. (The largest commercial units available have capacities only 1/50 of the WGS rotor torque.)

All of the variable speed rotor WGS concepts evaluated had high energy costs, and faced the same technical rotor design problem of the two-speed gearbox system (Concept 2). Section 8, Electrical Subsystem, discusses the relative advantages and disadvantages of the variable speed systems, key points of which are reviewed here.

A variable speed electrical system requires a means of converting variable shaft rpm into the constant frequency required by the utility network. Although there are several combinations of equipment that could be used to perform this conversion, cost and weight considerations would favor the use of a variable rpm AC generator driving a transformer-rectifier. Conversion of the DC power would be accomplished via a DC motor driving an AC generator or by a solid state inverter.

A variable speed electrical system such as this suffers several major disadvantages. DC machinery is substantially heavier and more costly than AC equipment of equivalent capacity. It is also less efficient, especially at partial power, and requires more maintenance. The solid state inverter offers the potential of better reliability and lower maintenance, but still suffers substantial weight and cost penalties. Lastly, when equipments are cascaded to perform the conversion process necessary for a variable speed system, net efficiency of the system is considerably reduced and the cost of controls and protective equipment is increased.

It appeared prudent, however, to carry one of the variable speed systems through conceptual design. The goal was to derive more thorough and accurate cost and weight estimates for one of these systems, and to evaluate their operating characteristics in more detail. Concept 5 was selected, therefore, as the variable speed system for conceptual design. It offered the lowest energy and capital costs and the lowest weight on the tower (the transformer-rectifier, and motor-generator set being mounted on the ground) of the variable speed rotor systems evaluated.

3.2.6 Selected Concept for Optimization

Conceptual designs for the candidate concepts described above were then prepared, and estimates made of component weights and costs from the designs. From these data, the baseline configuration (Concept 1), with a truss type tower, was recommended for optimization and preliminary design (Phases II and III of the study).

The conceptual designs for the three selected configurations were similar in overall layout to the preliminary designs shown in Figures 7-1 and 7-2, but with less detail developed. Descriptions of the components are given in Sections 4, 5, 6, 7 and 8 for the Rotor, Control, Structure, Drive and Electrical Subsystems, respectively.

The only essential difference between the conceptual and preliminary designs, aside from scale, is in the rotor. The 100 KW (5.4 m/s median wind speed site) low power system and the 1000 KW (8 m/s median wind speed site) high power system conceptual designs used servo flap pitch controls for the variable pitch rotor candidate (Concept 1) and were designed for either constant chord metal blades or tapered filament wound spar, bonded afterbody/skin composite blades. The 500 KW (5.4 m/s) low power system and the 1500 KW (8 m/s) high power system preliminary designs use direct root actuated pitch control and all filament wound composite blades. It was noted at the time that the conceptual designs were prepared that if the rotor diameter was greater than 150 feet, then metal blades would be less attractive than composite blades and, at times infeasible.

Weight and cost summaries for the three conceptual designs are shown in Tables 3-5 and 3-6, respectively. Of the three concepts, the variable speed rotor (Concept 5) has the greatest technical risk because some operating rotor speeds are near blade bending natural frequencies, resulting in dynamic amplifications of vibratory bending moments. As shown, the variable speed rotor system, irrespective of its technical problems, is a poor cost competitor and did not receive further consideration.

The final WGS concept choice was thus reduced to two systems which used AC synchronous (or induction) generators driven at constant speed directly by the rotor through a fixed ratio gearbox. The difference in the two concepts was solely in the mode of torque control; by variable blade pitch for Concept 1 or by a drag flap for Concept 3.

The choice of Concept 1, the baseline system, hinged on the questions of blade life and control. Using the fixed pitch approach eliminates only two sets of pitch bearings, a relatively minor cost item, and one that can be designed to have a very high reliability.

Using the fixed pitch system requires that the rotor operate in the so-called positive mode over its entire operating range. As explained in Section 4, Rotor Subsystem, this results in substantially higher rotor loads at high wind speeds than for the variable pitch system, and potentially higher fatigue damage to the blades. In addition, the rotor must often operate off-design in the so-called "vortex ring" state, an aerodynamically unstable regime with possible wide fluctuations in rotor thrust, also discussed in Section 4. This characteristic, extremely difficult to predict, could raise serious control problems and would result in a higher risk development program.

Thus, because the fixed pitch concept did not offer an economic advantage over the variable pitch system, but did threaten to increase development risks, the baseline configuration of a variable pitch rotor was selected as the preferred concept. The choice of a truss tower was based on the results of a parallel study of both a steel shell and a truss tower, indicating the truss tower to be substantially less costly. This is discussed in Section 6, Structure Subsystem.

3.3 Parametric Model

In parallel with the conceptual design task, a parametric computer program was prepared to optimize the preferred concept after it was selected. This section

TABLE 3-5. WEIGHT SUMMARY, kg (lb)

(WEIGHT ABOVE FOUNDATION)

	CONCEPT #1 VARIABLE PITCH CONSTANT SPEED		CONCEPT #3 FIXED PITCH CONSTANT SPEED		CONCEPT #5 FIXED PITCH VARIABLE SPEED	
	100 KW	1000 KW	100 KW	1000 KW	100 KW	1000 KW
ROTOR	3858 (8505)	14630 (32252)	3343 (7370)	12689 (27975)	4414 (9732)	16155 (35615)
DRIVE SYSTEM	2451 (5404)	14605 (32198)	2451 (5404)	14605 (32198)	3038 (6697)	17429 (38424)
ELECTRICAL SYSTEM	1334 (2941)	3459 (7626)	1334 (2941)	3459 (7626)	1479 (3261)	3808 (8396)
CONTROL SYSTEM	347 (765)	372 (820)	347 (765)	372 (820)	347 (765)	372 (820)
TOTAL, DYNAMIC SYSTEM	7990 (17615)	33065 (72896)	7475 (16480)	31125 (68619)	9278 (20455)	37764 (83255)
PINTLE AND DRIVE	5645 (12445)	19465 (42912)	5645 (12445)	19465 (42912)	5645 (12445)	19465 (42912)
WEIGHT ON TOWER	13635 (30060)	52531 (115808)	13120 (28925)	50590 (111531)	14923 (32900)	57229 (126167)
TOWER*	6841 (15082)	29166 (64300)	6841 (15082)	29167 (64300)	7847 (17300)	31026 (68400)
TOTAL SYSTEM	20476 (45142)	81697 (180108)	19961 (44007)	79757 (175831)	22770 (50200)	88255 (194567)

*EXCLUDING FOUNDATION

TABLE 3-6. CONCEPTUAL DESIGN WGS COSTS - ($)

	CONCEPT #1 VARIABLE PITCH CONSTANT SPEED		CONCEPT #3 FIXED PITCH CONSTANT SPEED		CONCEPT #5 FIXED PITCH VARIABLE SPEED	
	100 KW	1000 KW	100 KW	1000 KW	100 KW	1000 KW
ROTOR						
BLADES AND FLAPS	30,105	68,945	29,920	66,440	38,125	83,335
HUB	20,025	88,865	19,645	86,165	24,900	109,560
	50,130	157,810	49,565	152,605	64,025	192,895
TOWER						
STRUCTURE	7,950	35,440	7,950	35,440	9,745	38,530
EQUIPMENT	13,615	23,040	13,615	23,040	14,110	23,540
FOUNDATION	25,300	41,200	24,300	41,200	29,000	43,700
	46,865	99,680	46,865	99,680	52,855	105,770
PINTLE AND DRIVE	12,960	52,090	12,960	52,090	12,960	52,090
DRIVE SYSTEM	19,475	110,150	19,470	110,115	23,980	130,675
ELECTRICAL SYSTEM	28,695	79,740	28,695	79,740	53,960	219,685
CONTROLS SYSTEM	11,160	11,630	11,160	11,630	11,160	11,630
TOTAL WGS	169,285	511,100	168,715	505,860	218,940	712,745
ERECTION AND INSTALLATION	16,160	24,625	16,160	24,625	17,525	33,725
TOTAL INSTALLED COST	185,445	535,725	184,875	530,485	236,465	746,470

describes the model used to optimize the selected concept and the computer program structure used to automate the calculation procedure.

3.3.1 Model Description

The model consists of equations and tables which describe the WGS environment and constituent component performance, size, weight and cost. The inputs to the model are dimensions, ratings or other performance parameters, such as rated wind speed. The model yields derivative dimensions, complete weight and cost breakdowns and other parameters required to evaluate the system, such as unit energy cost, unit direct capital cost and plant factor.

The system is broken down into components which represent recognizable units for which dimensions, weight or cost can be described. The selected independent variables make the mathematical model responsive to changes that would result from sizing, tradeoff and sensitivity studies. Thus, the requirement was established at the outset that the equations be flexible enough to be easily changed and updated, and that the basic computer program structure be adaptable to different WGS concepts and components.

The equations and tables used evolved in several phases. The equations described here and shown in Appendix B are those which were used for the final parametric analysis of the WGS presented in paragraph 3.4. These were initially developed for the conceptual design evaluation task, and modified as results from that task and subsequent supplemental studies evolved. During the preliminary design phase, changes and design innovations were introduced in the design so that the final preliminary design no longer is precisely described by the model. However, sensitivity studies have been run, from which the impact of these changes may be evaluated. A comparison of the actual preliminary design results with the predicted model results is given in paragraph 3.6.

3.3.2 Calculation Procedure

The procedure used in sizing a WGS to specified requirements and evaluating it is summarized in Figure 3-5. System design parameters are specified as input for which related component sizes and ratings are determined for consistent ground rules. With the system and component dimensions known, it is possible to calculate yearly utilization and energy output. Also, weight and cost are determined to meet the required design criteria. Component costs are summed to yield system cost and, after adding costs for site and other installation costs, the overall acquisition or purchase cost of the WGS is determined.

WGS purchase cost is amortized as a yearly carrying charge, and operating and maintenance costs are added to determine total yearly operating cost. Finally, unit energy cost is calculated by dividing yearly operating cost by yearly energy output.

The design parameters that were used as input to the model are listed in Table 3-7. However, due to the flexibility of the computer program which is used to implement the calculation of the data, all model components can be thought of as input data and may be varied. Table 3-8 lists some typical parameters determined by the model.

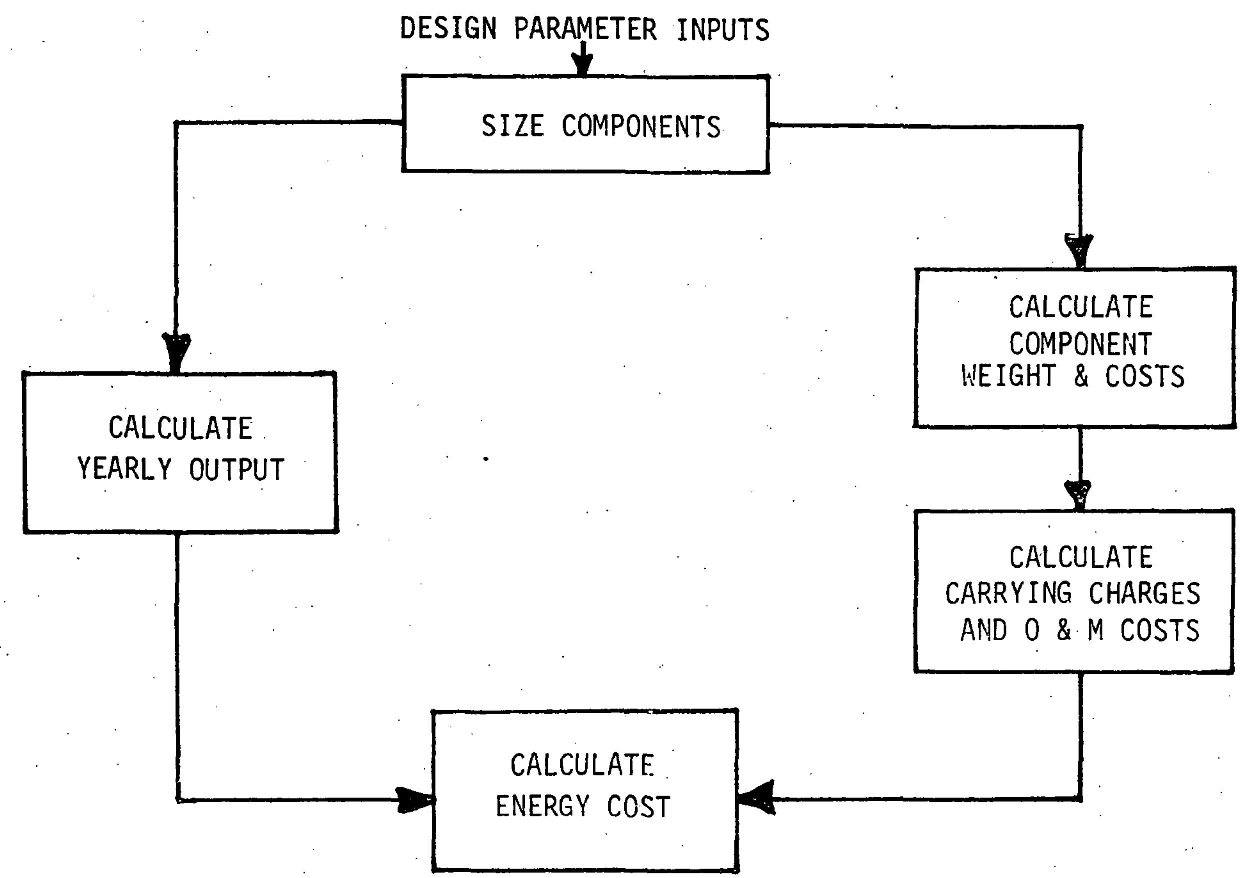


Figure 3-5. Model Procedure

TABLE 3-7. TYPICAL INPUT PARAMETERS

Rated Power	
Rated Wind Speed, or Rated Wind to Median Wind Speed Ratio	
Environment	Median Wind Speed
	Annual Wind Speed Frequency Distribution
	Density Altitude
	Terrain (Wind Shear Gradient)
Rotor	Diameter
	Solidity (Blade Area/Disc Area)
	Tip Speed or Rotational Speed
	Number of Blades
	Aerodynamic Efficiency (represents blade geometry)
	Inclination to Wind (Shaft Tilt)
Drive	Efficiency
Generator	Speed
Cable	Size (Area)
Tower	Rotor Ground Clearance
	Aspect Ratio (Height/Base)

TABLE 3-8. TYPICAL OUTPUT PARAMETERS AND DIMENSIONS (DETERMINED BY THE MODEL)

Site	Effective Wind Speed at Rotor
Rotor	Rotor Power
	Chord
	Velocity Ratio for Maximum Aerodynamic Efficiency
	Thrust Coefficient, Thrust
Drive	Transmission Gear Ratio
	Rated Torque
Electrical	Component Ratings
	Component Losses at Rated Condition
Tower	Height
	Base Width
	Projected Area of Tower and Enclosure
System	Output Power
	Component Weights
	Components Costs
	Plant Factor
	Energy Cost

A complete description of the calculation procedure is given in paragraph 3.3.4 and Appendix B. The summary description above is the overall procedural framework for the specific model relationships described in the next section.

3.3.3 Analytical Relationships

To properly and completely optimize the selected WGS concept, it was necessary to develop detailed relationships describing the overall system and its environment and each of the WGS subsystems and components. These relationships are described below.

Each of the equations, curves and tables used to calculate component size, rating, weight, cost, etc., were developed from information generated in the conceptual design task, modified by additional or parallel studies done in subsequent phases of the program. Where more than one relationship was used in the study, all are shown to give the reader a broader choice of model relationships.

All relationships are based on information covered in other sections of this report for the WGS subsystems. However, the model development and utilization was largely completed while the preliminary design effort was underway. Hence, as mentioned before, the relationships presented here do not always represent the final preliminary design component configurations, since refinements in some components continued through preliminary design. These differences and their probable impact on the results obtained with the computer model are discussed in paragraph 3.4, Results of Parametric Analysis.

3.3.3.1 Environment

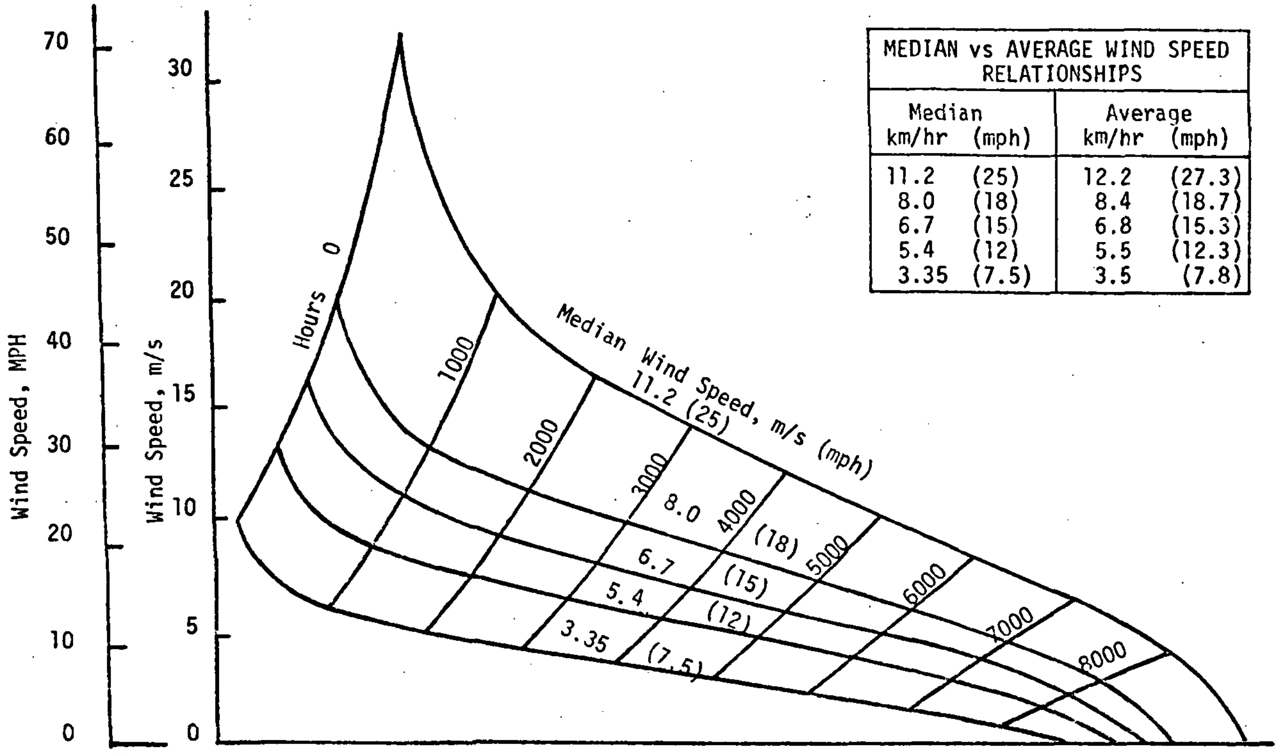
Several facets of the WGS site characteristics are accounted for by the model. These are grouped under "environment" to denote that they are site-specific and should be altered if the model is used for WGS sites other than that specified for this study. (However, the effect of site parameters on the results of the study were evaluated and are summarized in paragraph 3.5, Site Adaptability Studies.)

Wind Occurrence - Yearly wind occurrence statistics, i.e., wind frequency curves, are identified by their median wind speed value. NASA defined three wind occurrence profiles with median wind speeds of 3.35 m/s (7.5 mph), 6.7 m/s (15 mph) and 11.2 m/s (25 mph). These, together with interpolated curves for 5.4 m/s (12 mph) and 8.0 m/s (18 mph) are shown in Figure 3-6. These data simply describe the number of hours per year that the wind blows at a given velocity.

Wind Velocity Variation With Height - Wind velocity increases with height above ground at a rate which depends on terrain features (forests, ground contours, etc.). The wind occurrence statistics define wind velocity (V_{REF}) at a height (h_{REF}) of 9.1 m (30 feet) above ground. The variation of wind velocity (V) at any height (h) is given by:

$$V/V_{REF} = (h/h_{REF})^{.17}$$

where .17 is representative of typical conditions. Figure 3-7 illustrates this relationship.



MEDIAN vs AVERAGE WIND SPEED RELATIONSHIPS			
Median		Average	
km/hr	(mph)	km/hr	(mph)
11.2	(25)	12.2	(27.3)
8.0	(18)	8.4	(18.7)
6.7	(15)	6.8	(15.3)
5.4	(12)	5.5	(12.3)
3.35	(7.5)	3.5	(7.8)

Figure 3-6. Yearly Velocity Duration for Various Median Wind Speeds

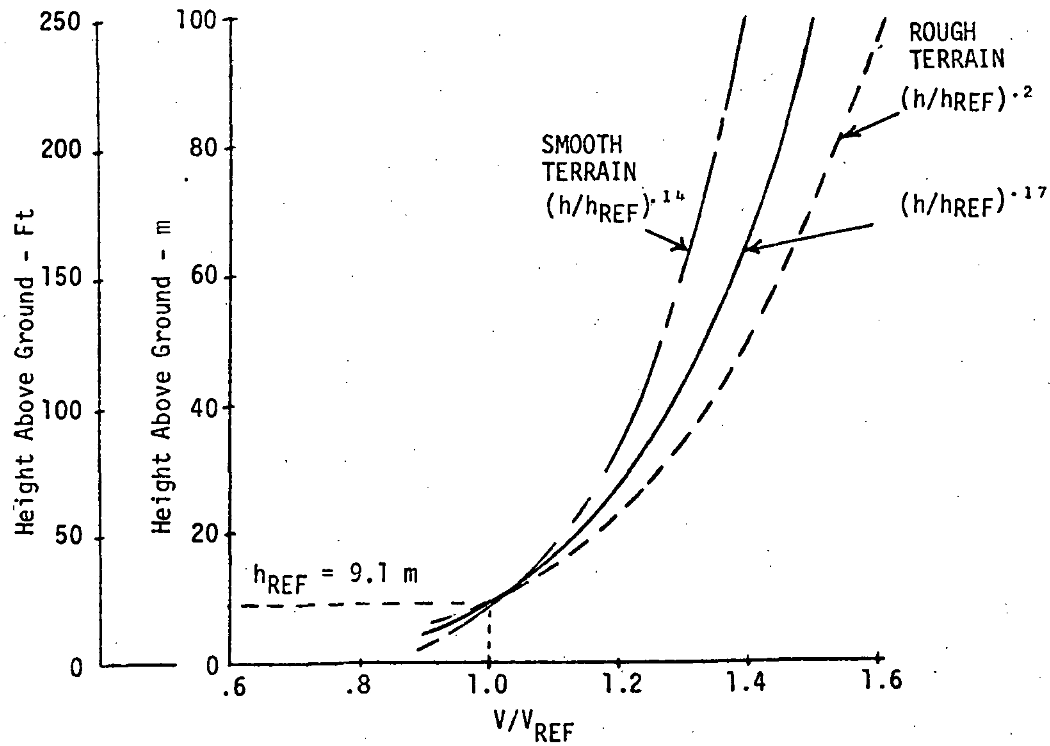


Figure 3-7. Wind Speed Variation With Height Above Ground

Due to the vertical variation of wind speed across the rotor disc, the rotor "sees" a value related to its area distribution with respect to height. This effective wind velocity, V_{EFF}, is obtained by integrating the wind velocity over the rotor disc (excluding the root cutout) as follows:

$$V_{EFF}/V_{REF} = \frac{(R/h_{REF})^{.17}}{1 - (r_c/R)^2} \int_{(r_c/R)}^{1.0} \frac{r/R}{\pi} \int_{0}^{2\pi} \left[\left(\frac{h_c}{R} + 1\right) - (r/R)\cos\psi\right]^{.17} d\psi \, d(r/R)$$

where:

h_c is the rotor minimum clearance from the ground

r/R is the normalized rotor spanwise station

R is the rotor radius

ψ is the blade azimuth position

r_c is the blade spanwise station where the airfoil begins (root cut-out).

From the above equation, it is seen that the effective wind velocity is a function of rotor clearance, radius (and thus, tower height) and blade root cutout. The equation is seen to account for the fact that reference winds are given a fixed reference height. Effective velocity variation with rotor radius and ground clearance is shown in Figure 3-8.

In the model, the above equation is represented by a polynomial curve fit of the results of the indicated integration. For ground clearances greater than 10 m, the effect of ignoring root cutout is small. The results of this integration are, therefore, used in the model without this effect.

It is important to note that, throughout this report, site wind speeds are quoted for the standard 9.1 m (30 ft) height, but that the actual velocity seen by the rotor is often much higher. Therefore, care should be exercised in direct scaling with wind speed to insure the correct speed is actually representative of the calculation.

Altitude - Temperature and ground elevation both affect the density of air in which the rotor operates. The model accounts for this using the standard density altitude approach. The density ratio, σ, is used for this purpose. σ = 1.0 at the baseline condition of sea level, 15°C.

3.3.3.2 Rotor Performance

The rotor extracts only a portion of the available wind energy. The power contained in wind flowing through a unit area is 1/2 ρV^3, where ρ is the air density and V is the wind velocity. The aerodynamic efficiency, E_{AER}, is defined

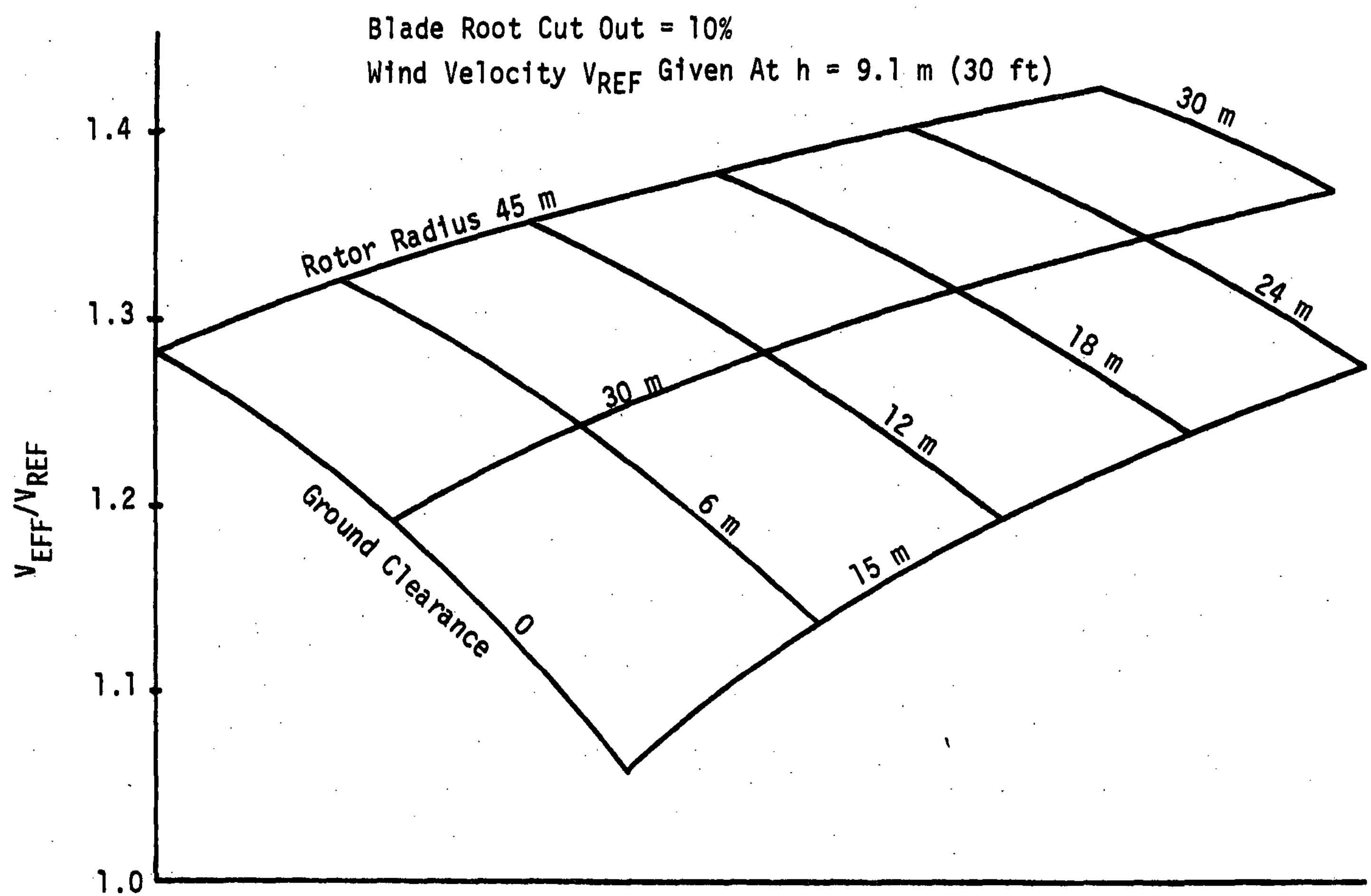


Figure 3-8. Ratio of Effective Velocity to Reference Velocity

as that portion of the energy captured by the rotor and converted into torque at the rotor shaft. Therefore, for a given rotor with a disc area of A_{ROT}, the rotor output power is:

$$P = E_{AER} \cdot 1/2 \, \rho V^3 \, A_{ROT}$$

Theoretically, the highest possible efficiency is 59%; however, for practical rotors, it is about 40%, depending on airfoil smoothness, geometry and other factors.

The baseline aerodynamic efficiency map used in the parametric model is shown in Figure 3-9. The graph shows efficiency at optimum blade pitch angles as a function of velocity ratio (V_{TIP}/V_{EFF}) for several rotor solidities. The baseline rotor performance model represents the following configuration:

Airfoil	230 XX, standard roughness
Thickness Taper	Varies from 40% inboard to 12% at the tip
Solidity	Equivalent solidity, integrated for equivalent torque
Twist	Optimum
Chord Taper	Accounted for in the equivalent solidity
Root Cutout	10%
Rotor Inclination	10° to wind axis

Variation of any of these parameters will change the aerodynamic efficiency. No attempt was made to model each variant since it was known that small variations in most parameters cause negligible changes. This is shown in paragraph 3.4.

Rotor performance data were calculated by a strip analysis method using propeller vortex theory. The calculations were performed using a high speed computer program which, in its basic form, has been in use for helicopter performance determination. Modifications were made to accommodate the wind turbine mode of operation, such as reversing the airfoil camber.

The method used for rotor performance data generation was compared to another program using blade element methods which were modified for other wind turbine requirements, such as gravity effects and tower wake. However, the results of the vortex theory program are more conservative and were used to represent the lower limit of expected performance.

For conditions where the wind exceeds rated wind speed, blade pitch is advanced to reduce rotor efficiency and keep system power output constant. Figure 3-10 shows how this is achieved. However, there is no need to define this pitch schedule in the model, once it is demonstrated that it is always possible to set the rotor controls to obtain rated power above rated wind speed.

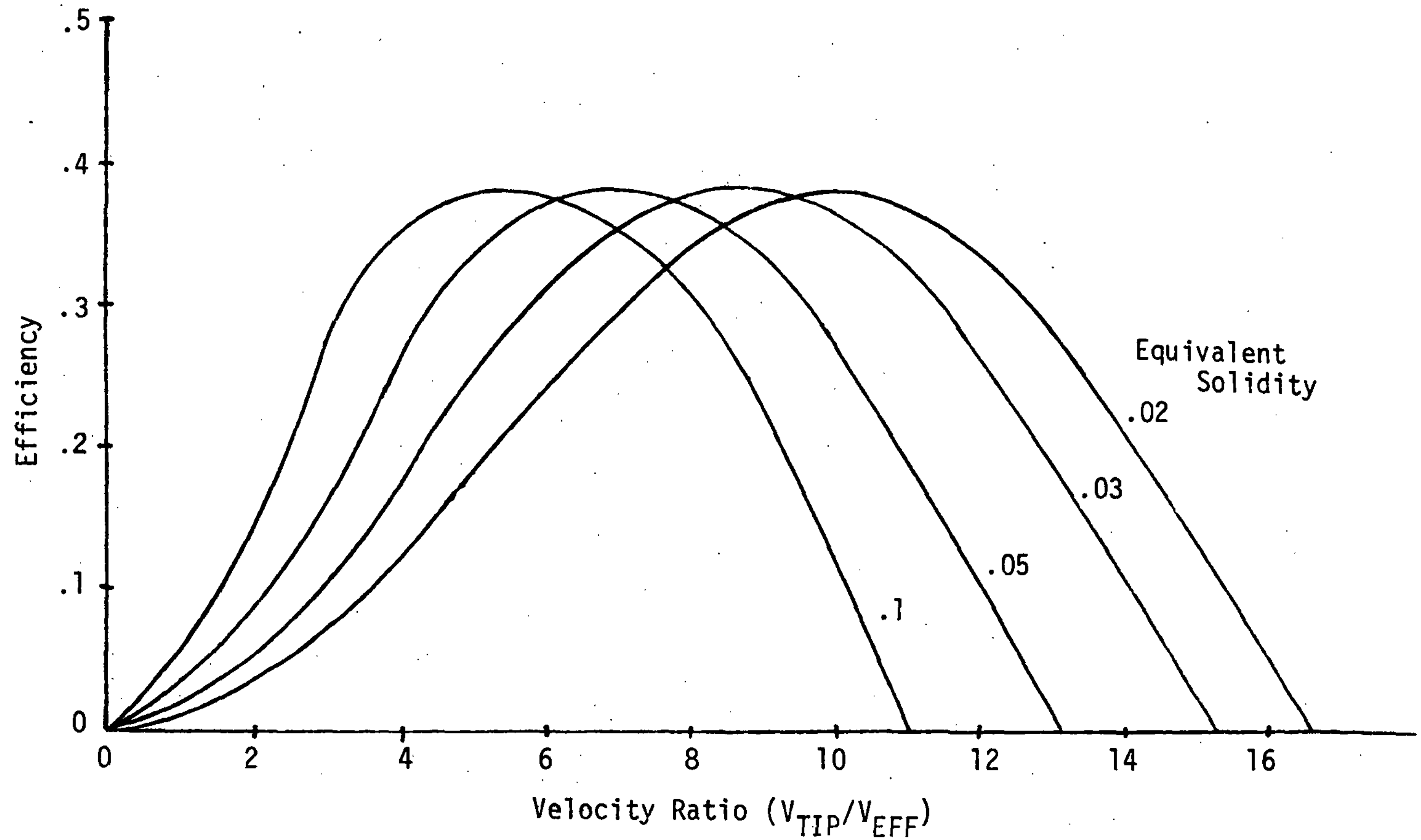


Figure 3-9. Rotor Aerodynamic Efficiency for Optimum Blade Angles

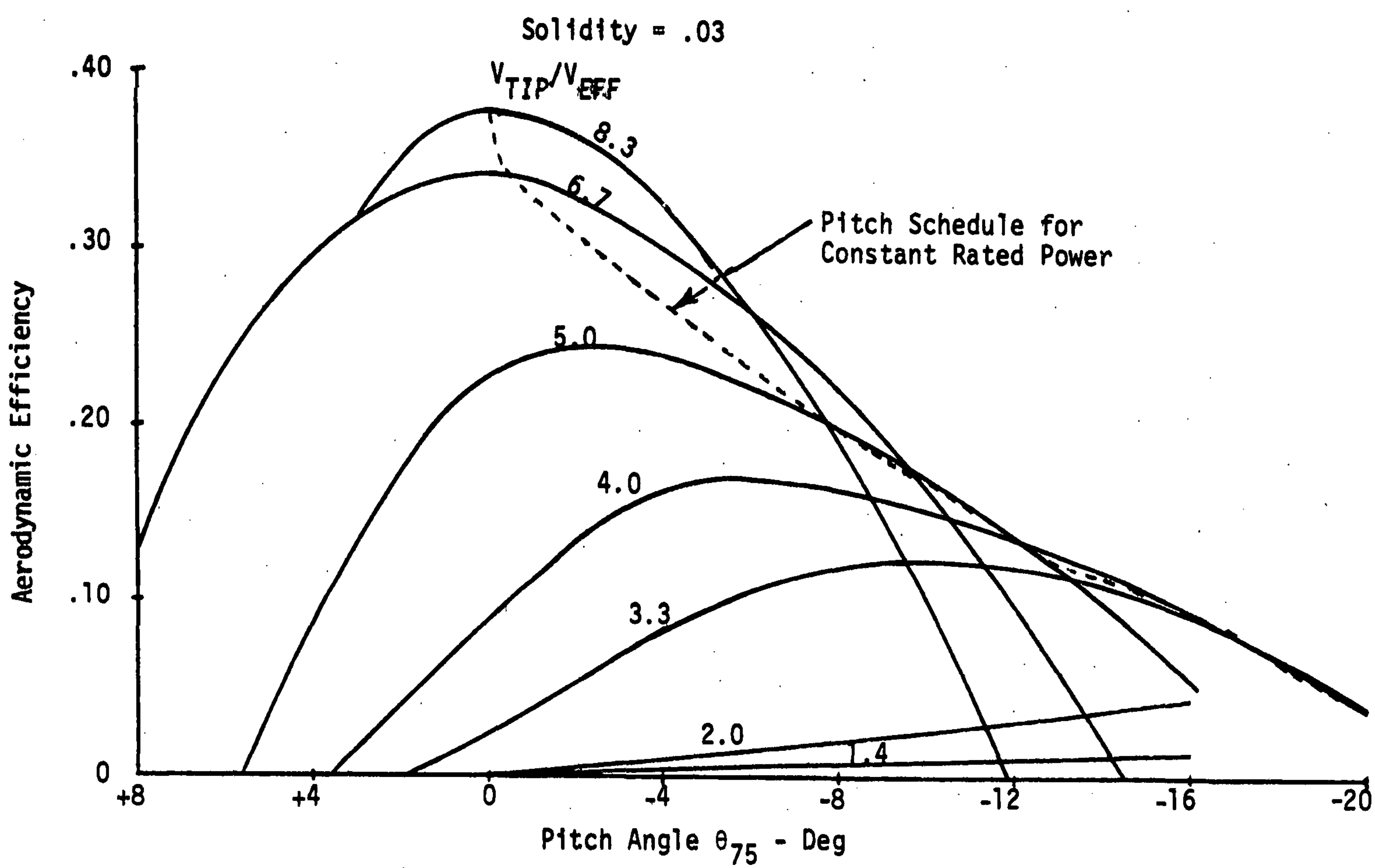


Figure 3-10. Aerodynamic Efficiency Variation With Pitch and Velocity Ratio

3.3.3.3 Power Generation

The rotor shaft drives a transmission by which the desired generator input rpm is achieved. The generator transforms the mechanical power to electrical power which is conducted through a cable to a transformer on the ground. The WGS system output is defined at the transformer terminal.

For each component, the efficiency (P_{out}/P_{in}) and losses used in the parametric model are a composite of data supplied by component manufacturers.

Gearbox - The following efficiencies were obtained from manufacturers for fixed ratio gearboxes:

	2 MESH GEARBOX	3 MESH GEARBOX
Normal Quality	.97	.96
Best Commercial Quality	.98	.97
Aircraft Quality	.985	.98

For the selected system concept, the best commercial quality was chosen as representative of the wind turbine requirements. Three meshes are required when gear ratios exceed 30. Sizing studies showed this is the case for the WGS, thus the efficiency used is .97.

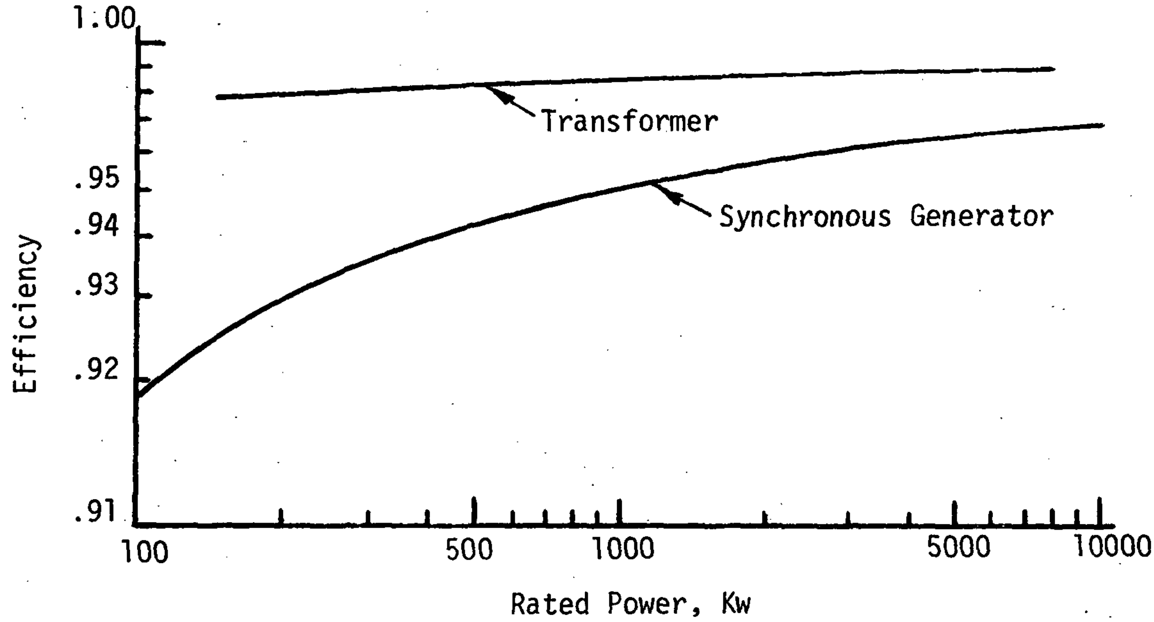
In addition, power is extracted at the gearbox input to drive the oil pump and control actuator hydraulic pump. The power absorption of these units was estimated at 3.5 KW for a 1500 KW WGS. These losses are expressed in the model as a function of rated power.

Generator - The generator losses depend on generator size, i.e., generator rating, and actual power output. The losses of a synchronous generator at rated power, P_R, are expressed as a fraction of input power as follows:

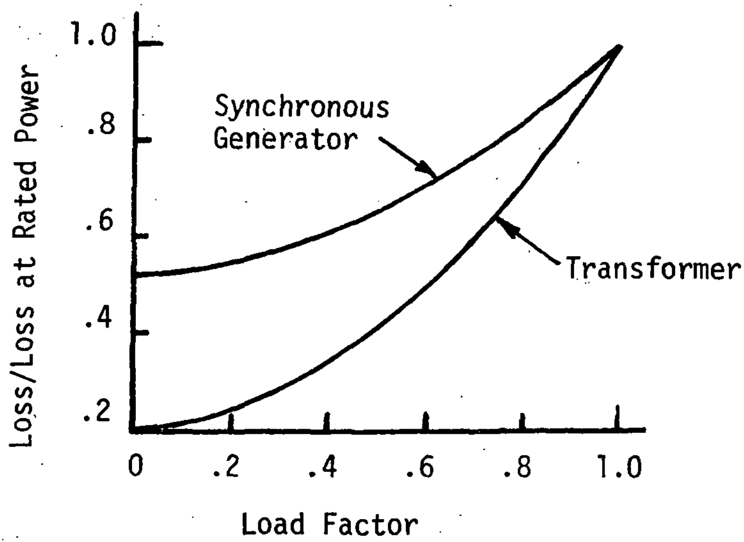
$$\text{LOSS} = .05\,(1000/P_R)^{.215}$$

The form of losses shown in Appendix B for the generator results from rearranging the above so that the loss is a fraction of output (and thus rated) power. The losses apply to a high speed generator with an rpm range of 900 to 3600 and include mechanical power input to drive the exciter, but do not include power consumed by the regulator. The losses at part power are defined as a polynomial function of load factor, i.e., fraction of rated power. Figure 3-11 shows the losses at part power and rated power efficiency of the synchronous generator and transformer.

Cable - The cable losses depend on cable cross-sectional area and cable length. The cable cross-sectional area is expressed in standard units, in which one unit is 350 Mcm. The cable length is the total of cable running the length of



(a) Efficiency At Rated Power



(b) Losses At Part Power

Figure 3-11. Generator and Transformer Losses and Efficiency the tower plus the length from the tower base to the transformer, the latter is here taken as 61 m (200 ft). For the baseline case, the cable area is one unit.

The cable power loss is proportional to the power level squared. The equations in the model reflect this. Losses for the baseline cable are shown in Figure 3-12.

Transformer - The transformer is the last component in the power conversion line. Therefore, its rating coincides with the WGS rating. Similar to the generator, the transformer losses are a function of size and actual power output. At rated power, the losses may be expressed as a fraction of input power as:

$$\text{LOSS} = .0153\ (1000/P_R)^{.21}$$

The expression for losses in Appendix B results from rearranging the above so that the loss is a fraction of output power, which in this case is synonymous with rated power. The transformer losses at part power are defined as a polynomial function of load factor, i.e., fraction of rated power. Data for the transformer losses at part power and the rated efficiency at rated power were shown in Figure 3-11.

Energy Output Integration - The energy output at any given wind speed is the product of the hours the wind occurs in a year and the output power of the system at that wind speed. Energy output starts at the lowest wind speed at which the system delivers net power to the utility, i.e., the cut-in wind speed. Power increases with wind until rated wind speed is reached. Above this, the energy output is at the rated power until the cutout, or shutdown wind speed is reached. Total yearly energy output is simply the integral of the energy output from minimum velocity for generating power to the cut-out wind velocity which, in this study, was selected to be the maximum statistically significant wind velocity. Increasing the cut-out wind velocity above this value will not increase the energy output and, therefore, will not improve the energy cost as represented in the model. Increasing cut-out wind velocity, however, would have several disadvantages. Blade fatigue loading would be increased, resulting in heavier blades or reduced fatigue life of the existing blades. Since the expected wind gust amplitudes are higher at the higher wind speeds, the probability of dangerous overspeeds due to loss of electrical load during a wind gust is increased. The pitch control system would have to be designed with faster response to prevent such an overspeed, adding to the control system costs. Also, it is not desirable to make the control system any more responsive than necessary, since the risks associated with hardover failures in the control system are increased as the system response rate is increased.

3.3.3.4 Weight

Weight equations were developed for WGS components above the foundation, since only these weights affect WGS costs. Weight items are listed in Table 3-9, showing their build-up hierarchy. Although the weight of components supported by the tower is used for tower sizing, the primary purpose for calculating weights is to provide component parameters for cost determination.

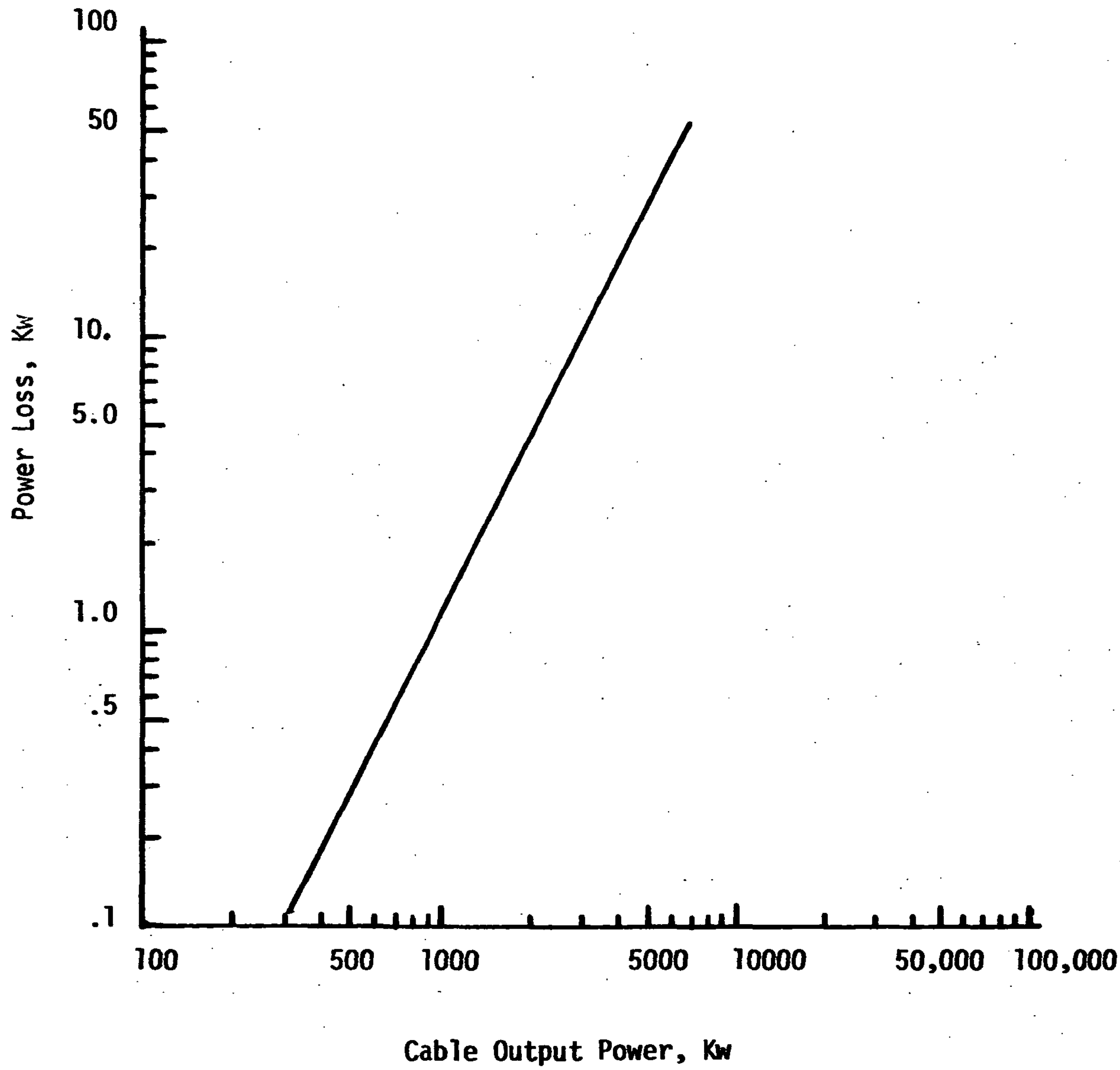


Figure 3-12. Cable Losses

TABLE 3-9. WGS COMPONENT WEIGHTS BUILD-UP

COMPONENTS ABOVE FOUNDATION

WGS Above Foundation	WWGS		
ROTOR	WROT	CONTROLS	WCTR
HUB	WHUB	PINTLE ASSEMBLY	WPTL
Housing	WHSG	Yaw Mechanism	WYAW
Hub Mounted Control	WRCL	Bed Plate	WBPT
Pitch Mechanism	WPCH	Enclosure	WENC
TOTAL COMPLETE BLADE	WBLS	TOTAL TOWER	WTTW
Flap (one)	WFLP	Total Structural Steel	WSTL
Blade (one)	WBLD	Base Support	WBAS
		Miscellaneous (Attach)	MMSC
DRIVE SYSTEM	WDRS	Gussets	WGUS
Slip Device	WSLP	Horizontal Braces	WBRC
Clutch	WCLH	Diagonals	WDIA
High Speed Coupling	WCP2	Chords	WCHD
Low Speed Coupling	WCP1	Pintle Support	WPTS
Gearbox	WTRN	Ladders, Platform	WLAD
Shaft	WSFT	Electric Lights, etc.	WEL2
		Cable (on Tower)	WCAB
ELECTRIC SYSTEMS ON TOP	WELE	Ring Gear	WRGR
Electric Equip. on top	WEL1		
Generator	WGEN		

All equations used appear in the model computer program listing, Appendix B. In this section, the bases of the equations will be given, with references to Appendix B. All components on top of the tower will be found in the MODULE TOP WEIGHT, while the components associated with the tower are found in the MODULE TOWER WEIGHT. It should be noted that the equations shown in Appendix B have variables in English units.

Blades - The blade weight (WBLD) equation is based on an analytical design analysis conducted in the study, using composite blades and a flexbeam hub. (See Section 4.) In this study, a variety of rotor solidities and diameters were chosen. Analyses were carried to a point where the rotor had reasonable weight distributions for required flatwise and chordwise stiffnesses. Section properties at selected stations were determined, from which the blade weights were calculated. The weights of these blade designs were matched by an equation on a point-to-point basis, with no attempt to force any preconceived trends through the values. Figure 3-13 illustrates the "goodness" of the equation by comparing it with the analytical study data points.

The blade weight model shown in Figure 3-13 evolved from the detailed study of rotor weight and technical limits initiated after a preliminary model for both metal and composite blades had been developed using statistical methods. The system model based on the original blade model indicated a very high cost incentive to minimize rotor solidity. Therefore, the more detailed rotor study was performed to develop an improved rotor model. Figure 3-14 illustrates both rotor models, where the original statistical blade weights are compared with the analytical or revised weights. The analytical study results are seen to yield a steeper weight versus diameter relationship for the .03 solidity shown. Figure 3-15 illustrates weight trends with solidity. The analytical study resulted in a shallower trend with solidity than with the original statistical equation.

Hub - The hub equation form reflects that the hub weight (WHUB) is related to the retention arm length and blade weight, the latter being indicative of the blade moment distribution. The coefficients are based on helicopter statistics. For the purpose of cost estimating, the hub weight was broken down into its components; pitch mechanism (WPCH), hub mounted controls (WRCL) and housing (WHSG). The equations are presented in Appendix B.

Total rotor weight (WROT) is then the sum of total blade weight and hub weight.

Drive Components - The Drive Subsystem consists of several components, of which the shaft and gearbox are most significant.

The shaft weight (WSFT) includes bearings and bearing support. The weight is a function of rotor torque and is based on an analytical study.

The gearbox weight (WTRN) includes its accessories and is a function of input (rotor) torque. The equation is based on manufacturers' statistics for normal commercial quality gearboxes with a factor applied for improved quality gears, as is anticipated for this application. The equation reflects three-mesh systems as would be required by the step-up gear ratios for the low solidity

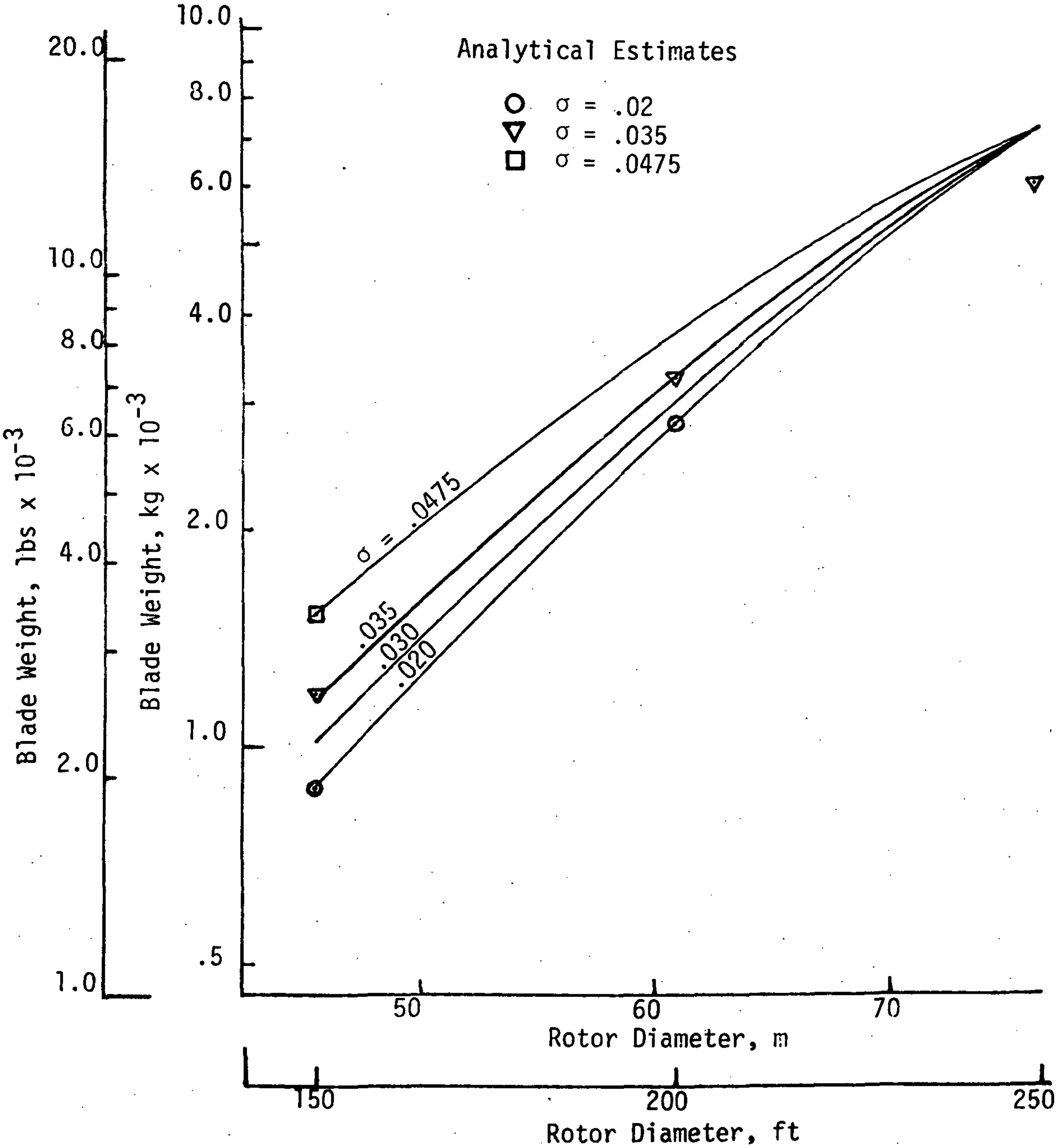


Figure 3-13. Rotor Blade Weight Model

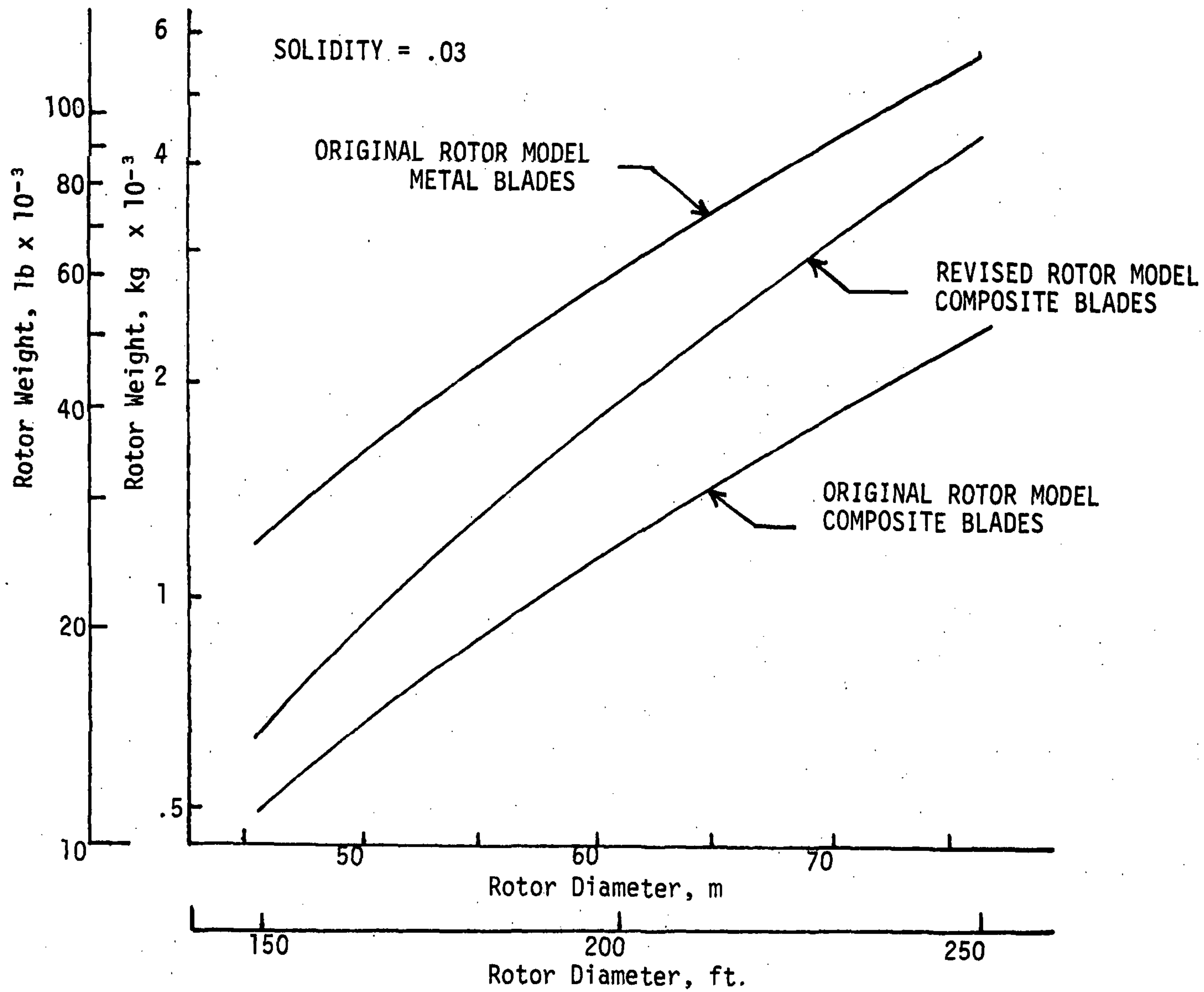


Figure 3-14. Rotor Weight Versus Diameter

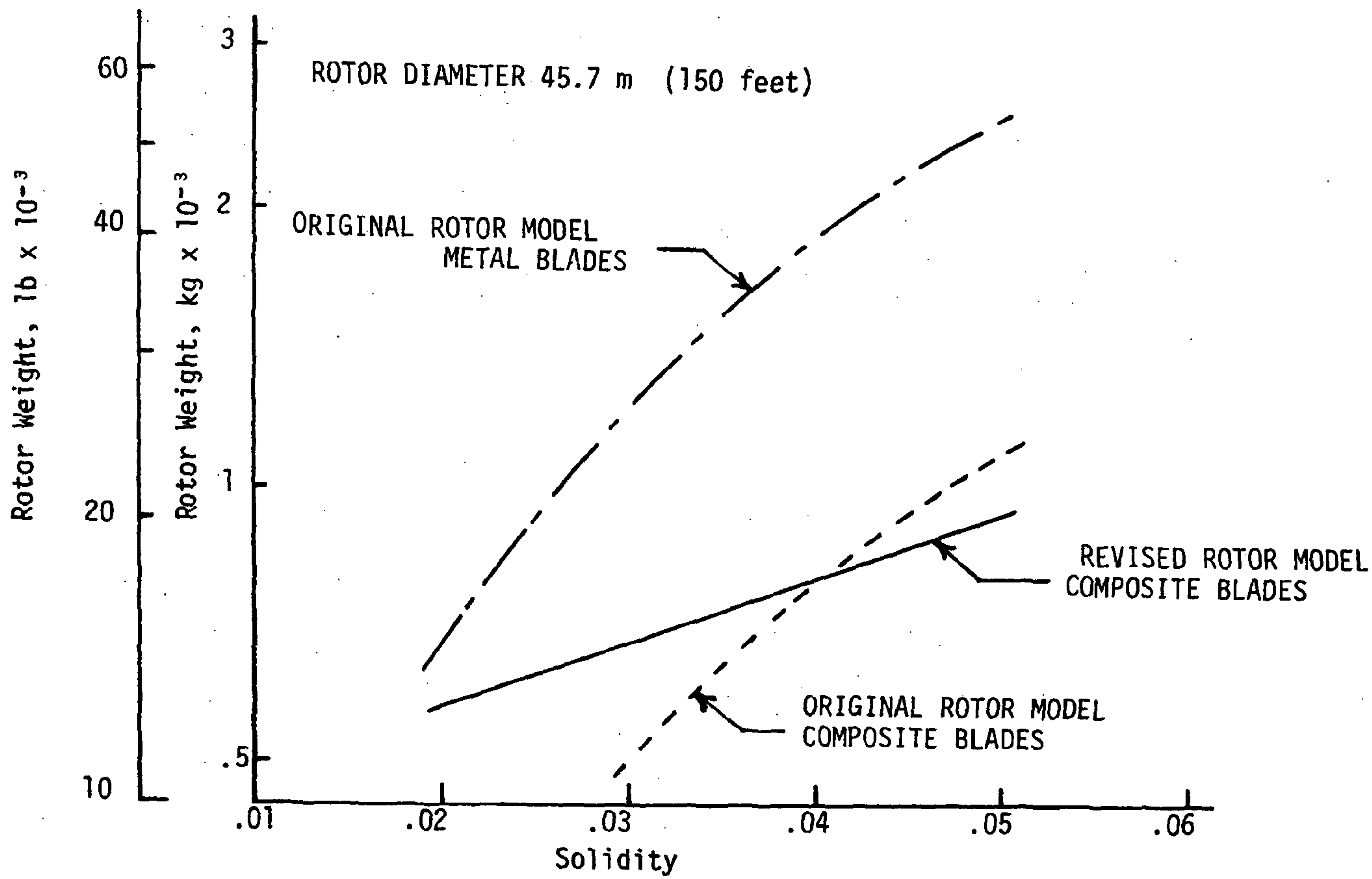


Figure 3-15. Rotor Weight Versus Solidity

rotor systems. The weight-torque relationships are shown graphically in Figure 3-16. In using this model, no margin is included for cyclic peak torques, which were included in the preliminary design.

Other components in the drive system are the low and high speed couplings (WCP1 and WCP2) and clutch (WCLH). The weights of these are based on manufacturers' statistics.

The total Drive Subsystem weight (WRDS) is the total of the preceding.

Electrical Equipment on the Tower - The electrical equipment on the tower includes the AC synchronous generator and miscellaneous equipment, such as sensors, wiring, sliprings, etc. All weights are based on manufacturers' data. The generator weight equation (WGEN) includes mounting provisions and represents data for 1800 rpm machines with adjustments for drive speeds less than 1800 rpm. Figure 3-17 shows the statistics and the resulting curve. The remainder of the electrical equipment weight (WEL1) is combined into one expression. The total electrical equipment on top (WELE) is the sum of these two.

Controls - The non-hub mounted controls include auxiliary pump and motor, hydraulic actuators and the control electronics. The controls weight (WCTR) reflects analytical weight estimates for two WGS sizes; 100 KW and 1000 KW. For the purpose of trending, it was assumed that the primary sizing parameter is rotor diameter.

Pintle Assembly - The pintle or turntable assembly (WPTL) consists of a bedplate (WBPT) to carry all the components on top of the tower, an enclosure (WENC) to house the drive system and electrical equipment, and a hydraulically powered mechanism (WYAW) which, by use of a ring gear, rotates the bedplate. All of the weight equations are based on analytical weight estimates from the conceptual design phase. These weight estimates are correlated with the supported weight (WRDG) which is the total of rotor drive electrical and controls weight. Therefore, the pintle weight is scaled with the weight carried by the bedplate.

Tower - The tower is of steel truss construction in which members are sized for loads or for stiffness, whichever criteria results in a higher weight. Attached to the top of the tower are components such as the ring gear and pintle support while others, such as the main power cable, utility electric lights and auxiliaries (ladders, etc.) are distributed along the tower. A base support structure attaches the tower to the foundation.

The ring gear weight (WRGR) equation was derived from analytical estimates correlated with the total weight it carries. The weight of ladders, gratings and railing (WLAD) are scaled from estimates, using manufacturers' data as a guide. The weight of the portion of electric power cable and electrical utilities that is attached to the tower was also derived from manufacturers' data. The electrical utilities (WEL2) represents utility lights and other electrical lines. The ladders, cable and electric utilities weights are assumed to be distributed along the tower for the purposes of tower sizing.

The main structural members of the tower are the tower legs or chords, diagonal braces and horizontal braces. These are sized to meet the three strength and

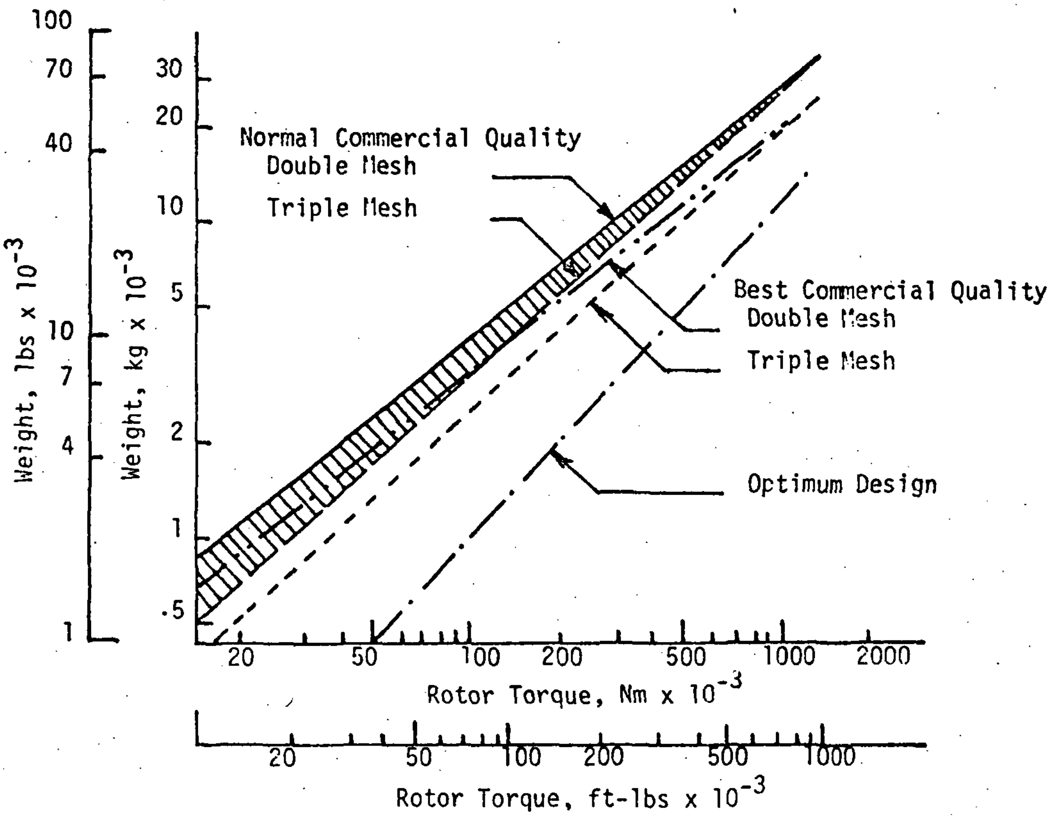


Figure 3-16. Fixed Ratio Gear Gearbox Weights

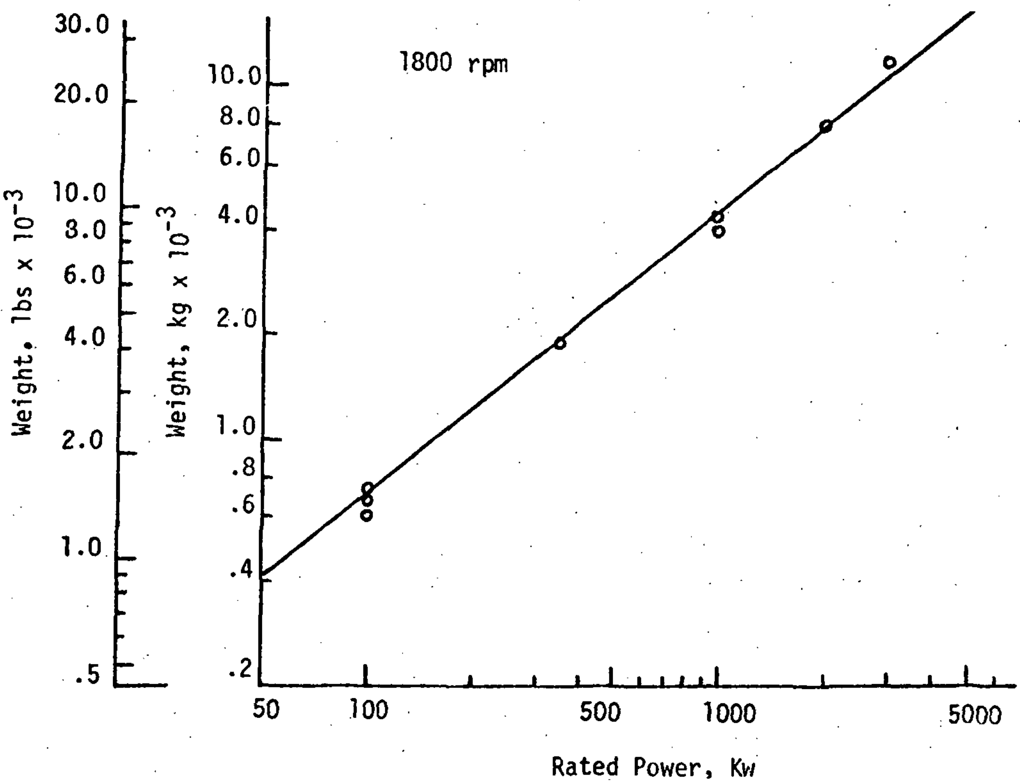


Figure 3-17. Weight of Synchronous Generators

two stiffness requirements as follows:

1. Blowover load - blades stopped in vertical position with the control system failed so that the blades are exposed flatwise to the maximum wind.

2. Normal operation - the loads at rated wind speed with seismic loads superimposed.

3. Gusts - loads superimposed on normal operation loads.

4. Bending frequency - stiffness requirements which set tower natural frequency at 1.5 times the rotor operating rpm.

5. Torsional frequency - stiffness requirements which set tower natural frequency at 2.5 times the rotor operating rpm.

A detailed discussion of these conditions is given in Section 6, Structure Subsystem.

An analytical sub-model was constructed to represent the loading of the members. Then, for given maximum allowable stress, calculated loads allow determination of cross-sectional areas needed and thus the weights of members. In this manner the chords (WCRD) are sized for the combined effect of weight on top, horizontally applied load, overturning moment and tower torque. The diagonal (WDIA) and horizontal bracing (WBRC) are sized similarly for tower torque. In addition, the model was formulated to relate stiffness to natural bending frequency. The chords are sometimes sized by bending frequency requirements and the diagonals and horizontal braces are sometimes sized by torsional frequency requirements, depending on the size of the system and the prescribed conditions.

After the tower main members are sized, other tower components, such as pintle or turntable support (WPTS), gussets (WGUS), base support (WBAS) and miscellaneous attachments (WMSC) are calculated. These were analytically determined fractions of the main member total weights sized for strength requirements only. The total structural steel weight (WSTL) is then the total of the structural members alone.

An alternate to the steel truss tower is a pre-stressed concrete shell tower which is not treated in the model, but which was investigated later in the study. Although slightly higher in cost, the concrete tower is aesthetically more appealing than the truss tower. It is believed that substitution of the concrete tower for the truss tower in the model would not significantly affect the results of the parametric analysis.

3.3.3.5 Direct Capital Cost

Direct capital cost is the cost to the utility, acting as its own prime and general contractor, for purchase and installation of all WGS components and subsystems, including the WGS site. In arriving at the direct capital cost for a complete wind generator system on a production basis, it was assumed that complete subassemblies would be provided essentially off-the-shelf by a well-established, mature, wind energy industry. Such an industry would provide the utility with components that could be simply bolted in place without the need for additional manufacturing or subassembly operations. Installation costs included in the parametric cost analyses reflect this concept.

It was assumed in this study that all components and subassemblies would be procured from vendors and suppliers as direct material purchases, not as construction or fabrication procurements. Therefore, direct labor and overhead rates, general and administrative expenses, and fees were assumed to be included in vendor pricing on which the parametric cost models were based. Direct capital cost presented herein does not apply multiple cost burdens which result from multi-level subcontractor procurements. The assumption that the utility acts as its own prime and general contractor also eliminates fees normally charged by such agencies. No allowance has been made for costs incurred internally by the utility for its prime and general contractor activity, for start-up costs, interest on construction loans, or contingencies.

Direct capital costs of individual WGS components are calculated in the parametric model using analytically or statistically derived equations, each representing the cost in terms of the component rating, weight or dimensions. The costs are estimated in constant 1975 dollars. The WGS direct capital cost is the sum of the individual costs. The direct capital cost, when divided by the WGS rated power, results in direct capital cost per kilowatt ($/kW). The direct capital cost and its elements also form the basis of determining yearly energy costs of the system.

Table 3-10 gives the WGS cost elements calculated by the computer program. In general, the items comprising direct capital cost parallel those in the weight breakdown of Table 3-9. All of the electrical system components are included (those at ground level, as well as those on the tower), and the foundation is added. In addition, costs for installing the system on site, and costs of the site and site preparation are included.

The cost equations are for a 1000 unit production level. Since most component and installation related costs do not vary much with production level, these same estimates should be good for other production levels as well. The exception is the rotor which, because of its new product status, would have a learning curve effect and a high tooling amortization effect.

Rotor Blade Cost - Because the rotor blades represent the most costly components of the system, a separate study was made of the costs associated with the manufacture of these components. The objective was to devise relationships by which the major cost elements (materials, fabrication and assembly) could be estimated parametrically on the basis of one or more key variables, such as rotor diameter and blade weight.

Kaman has, over a period of several years, been engaged in a number of advanced rotor blade design programs whose objectives included improvements in blade performance, field repairability and cost. At the time the parametric model was being developed, the company was just completing a preliminary design study for

TABLE 3-10. WGS COMPONENT COST BUILDUP

WGS DIRECT COST	CDIR
ROTOR	CROT
Hub	CHUB
Housing	CHSG
Hub Mounted Controls	CRCL
Pitch Mechanism	CPCH
TOTAL COMPLETE BLADE	CBLS
Flap (one)	CFLP
Blade (one)	CBLD
DRIVE SYSTEM	CDRS
Slip Device	CSLP
Clutch	CCLH
Brake	CBRK
High Speed Coupling	CCP2
Low Speed Coupling	CCP1
Gearbox	CTRN
Shaft	CSFT
ELECTRICAL SYSTEMS	CELC
Electric Utilities	CEL4
Equipment for Pwr. Gen.	CEL3
Transformer	CTRF
Cable (Total)	CCAB
Generator	CGEN
PINTLE ASSEMBLY	CPTL
Ring Gear	CRGR
Yaw Mechanism	CYAW
Bed Plate	CBPT
Enclosure	CENC
CONTROLS	CCTR
TOWER (Incl. Installation)	CTOW
Foundation	CFND
Structural Steel	CSTL
Ladders, Grating	CLAD
INSTALLATION (Excl. Tower)	CINS
Controls	CINC
Electrical Installation	CINE
Drive Installation	CIND
Pintle Installation	CINP
Rotor Installation	CINR
SITE	CSIT
Pad	CPAD
Shed	CSHD
Fencing	CFNC
Land Clearing	CCLR
Land Acquisition	CLND
TOTAL YEARLY COST	CYR
Carrying Charges	CCAR
Operation and Maintenance	COAM
Operation	COPS
Total Maintenance	CMNT
Maintenance of Site	CMAS
Maintenance of Tower	CMAT
Maintenance of Pwr. Sys.	CMAP
Maintenance of Rotor	CMAR

an all-composite advanced rotor blade for the Army's AH-1Q helicopter. Data developed under this and other programs were examined to construct realistic cost estimating relationships.

A standard cost approach was taken where the cost of the rotor blade is expressed as a function of material volume, blade length and number of parts (complexity). This expression is:

$$\text{Total Cost} = \text{Cost of Materials} + \text{Cost of Fabrication} + \text{Cost of Assembly}$$

$$= (V_{Pieces} \times C_{Material}) + (V_{Total} \times C_{Fabrication}) + [(L_{Bld} \times N_{Parts}) \times C_{Assembly}]$$

$$= [W_{Bld} \times C_{Material\ and\ Fabrication}] + [(L_{Bld} \times N_{Parts}) \times C_{Assembly}]$$

where: V = Volume (pieces and total)

L_{Bld} = Blade length

N_{Parts} = Number of parts

W_{Bld} = Blade weight

$C_{Material}$ = Material costs by volume and weight

$C_{Fabrication}$ = Fabrication (labor) costs by volume and weight

$C_{Assembly}$ = Assembly (labor) costs

As shown, the approach was to add the material costs (based on material volume) to the fabrication costs (also by volume of subassemblies) and the assembly cost (by size of blade and number of parts) to calculate total costs. Volume of material for a given blade design and choice of materials can be replaced by blade weight, sometimes a more convenient parameter.

The Field Repairable/Expendable Main Rotor Blade (FREB), designed and developed by Kaman for the Army's UH-1H helicopter, was analyzed first. The cost factors developed for this aluminum spar, composite afterbody blade are shown in the first section of Figure 3-18. These same factors were then applied to the all-composite AH-1Q rotor blade with the results shown in the next section of Figure 3-18. Lastly, the cost factors, modified to account for the more highly automated assembly methods involved with metal construction, were applied to the NASA/Lockheed 100 KW WGS blade as shown in the lower third of Figure 3-18.

On the basis of these three applications, it was concluded that a good estimate of rotor blade costs could be obtained from estimates of blade length, weight and number of parts. Separate equations were used for the two blade designs being considered: metal spar with composite afterbody and all-composite construction. The cost equation for the all-composite design ultimately decided upon is given in Appendix B under the variable name CBLD.

EXTRUDED SPAR/CONSTANT CROSS SECTION (FREB - Kaman Analysis)

		\$ 1741	Materials	
$7443\ in^3 \times \$\ .05/in^3$	=	372	Fabrication	\$ 10.90/lb + Assembly
22 ft x 34 parts x \$ 2.30	=	1720	Assembly	
		\$ 3833	(FREB = \$ 3831)	

ALL COMPOSITE (AH-1Q Army Analysis)

		\$ 4863	Materials	
$9757\ in^3 \times \$\ .05/in^3$	=	488	Fabrication	\$ 23.50/lb + Assembly
19 ft x 27 parts x \$ 2.30	=	1180	Assembly	
		\$ 6531	(Army = \$ 6011; Kaman = \$ 6890)	

FABRICATED METAL (NASA/Lockheed Blade - C-130 Wing Analogy)

		\$ 6790	Materials	
$30256\ in^3 \times \$\ .05/in^3$	=	1513	Fabrication	\$ 4.15/lb + Assembly
60 ft x 400 parts x \$ 1.15	=	27600	Assembly	
		\$35903		

Figure 3-18. Blade Cost Reference Designs

Other Cost Relations - As mentioned in the previous section on weight, most other component costs, as given in Appendix B, are based on component weights. These included such items as bearings, shafts, weldments and sheet metal assemblies, typical examples of which are given below:

\$4.4/kg (\$2/lb)	Machined parts, such as rotor shaft
\$9.9/kg (\$4.5/lb)	Bearings
\$6.0/kg (\$2.7/lb)	Gearboxes
\$1.2/kg (\$.55/lb)	Structural steel for tower

The hub cost, for example, is calculated using the weights of the housing, pitch bearings and control linkages and multiplying by a \$/unit weight factor. For the housing, the flexbeam hub concept used in the model envisioned a large aluminum plate machined to required dimensions. For a given size hub (diameter), the plate cost plus the machining cost was related to the final hub weight to form the basic \$/lb value. Pitch bearing costs were based, as were all bearing costs, on their weight for similar applications and duty cycle. The control actuators were considered machined parts and used the value per pound given above.

Some of the cost estimates were correlated with the height of the tower, such as cost of ladders, gratings and railing (CLAD). Others, particularly those for the Electrical Subsystem, were correlated with power rating of the units. The equation for the synchronous generator is illustrated in Figure 3-19, for example. In some cases, the cost was found to be related to item complexity and functions performed and did not vary appreciably with size or weight. The cost of such items, the system controls for example, were treated as constants.

The foundation cost (CFND) represents the cost of excavating and the cost of concrete. The three loading conditions for the tower were used to cost the foundation, the blowover load, the maximum thrust operation with earthquake load and the maximum thrust operation with gust load. The most severe of these conditions was selected for foundation sizing.

The cost of installation is also a capital cost and was estimated for the rotor (CINR), pintle assembly (CINP), drive system (CIND) and controls (CINC) as constants; size was not found to be a significant factor in installing most of the system components. Similar equipment and just as much preparation and effort are required to mount a WGS scale blade, irrespective of exact size. Exceptions are the electrical and tower installations; for the former (CINE), an expression was derived based on cost of electrical equipment; in the case of the latter, the tower erection cost is included in the cost of tower structural steel, which is standard practice.

It was estimated that the site (ALND) should be sized at 5 times the rotor diameter squared as a minimum size for security and safety, and that an area 3 times rotor diameter squared should be cleared (ACLR) in the vicinity of the WGS unit. Finally, an area equal to diameter squared should be fenced. The land acquisition cost (CLND) and clearing cost (CCLR) were estimated at \$.12/m^2

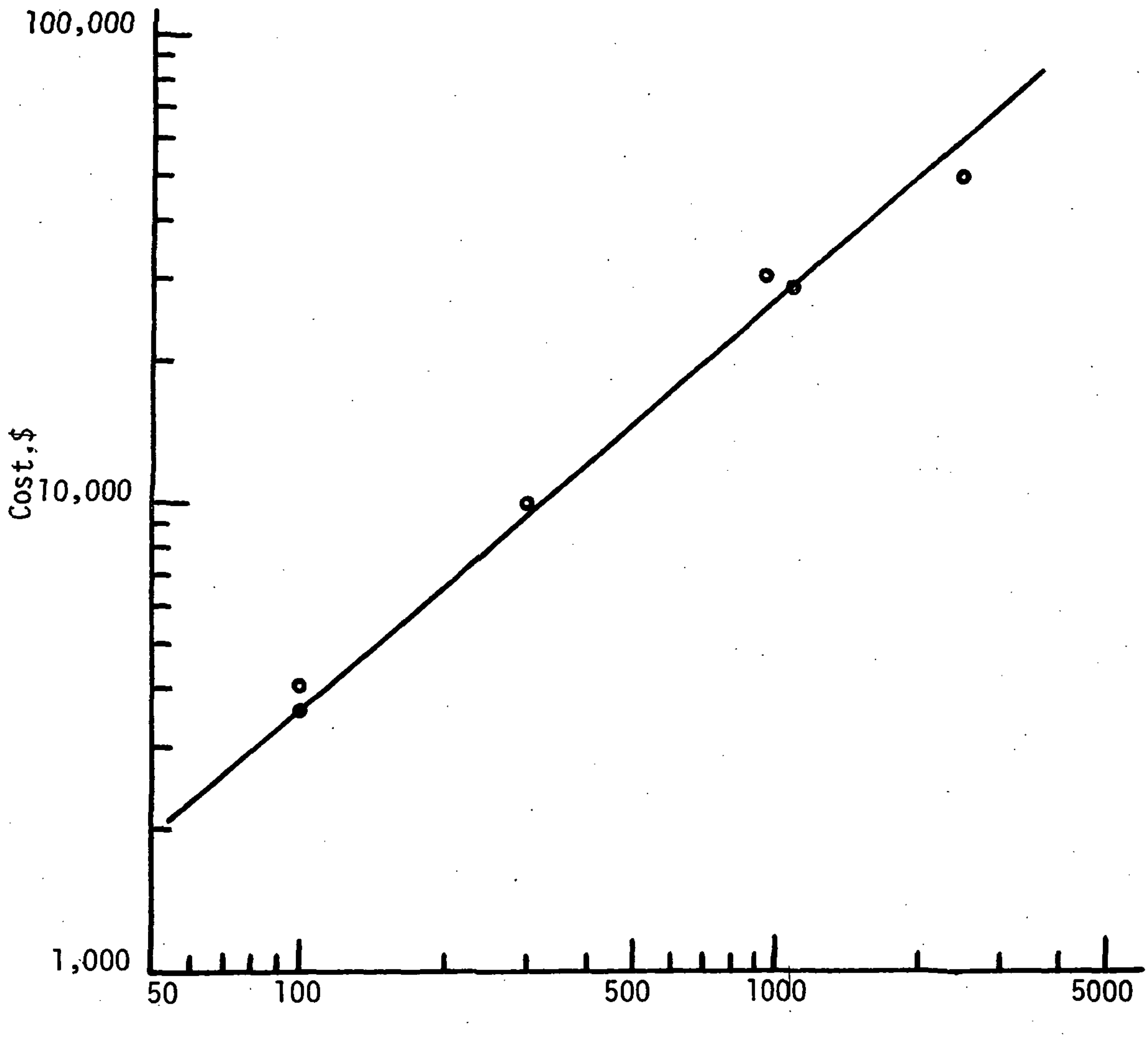


Figure 3-19. Cost of Synchronous Generators

($500/acre) and $.37/m^2 ($1500/acre), respectively, and the fencing cost at $40/m ($12/ft) from Northeast Utilities data. The land acquisition cost may vary somewhat from location to location; however, as will be seen from the model results, it is a very small portion of the total cost and thus, the variability of land acquisition cost does not have a significant impact on the results of the study.

Provisions for owner's costs were included in the model for direct capital cost determination. Although estimates were obtained on the order of 25% of total hardware, installation and site cost, owner's costs were not included in this study because they vary widely from utility to utility. Each utility reviewer can add his particular estimate to calculate total capital cost.

3.3.3.6 Yearly Operating Cost

Normal utility cost procedures break total yearly operating costs into three categories: carrying charges, operations and maintenance costs, and fuel costs. For the WGS, the last cost, obviously, is zero. This procedure was followed in the model, as shown below. The elements of yearly operating cost are shown in MODULE NAME-COST, Appendix B, page B-22, lines 65 - 73. Energy cost is calculated in MODULE NAME-PLANT EFFICIENCY, Page B-22, Line 8.

Carrying Charges - Northeast Utilities uses, for financial planning purposes, a carrying charge rate which expresses the average annual cost of a capital investment, excluding operations and maintenance, over its anticipated useful life. The carrying charges, which cover depreciation, debt service, return on equity and taxes are calculated by Northeast Utilities using a financial planning model. Total expenses over the life of the investment are converted into an average annual premium or carrying charge, taking into account the time value of money.

Based on financial planning factors supplied by NASA, Northeast Utilities' model was used to calculate an annual carrying charge rate for the WGS. The 14.7% rate calculated by Northeast's financial model was rounded to 15% for use in the parametric modeling. Section 9 of this report contains a more detailed discussion of carrying charges and operations and maintenance costs.

Operations and Maintenance - Operations and maintenance costs are commonly expressed as a yearly percent of initial component or subsystem cost. These are given by class of equipment, type of generating plant and other factors established by Federal Power Commission guidelines.

Yearly operations and maintenance costs were, with the exception of rotor maintenance, obtained from rates experienced by Northeast Utilities for comparable plants and equipment. Rotor maintenance costs were estimated from helicopter experience, and are considered conservative. Annual maintenance costs used in the model, expressed as a percentage of initial capital investment, are given below:

	PERCENT OF INITIAL COST
Rotor	10
Power (Drive, Electrical, Controls)	3.5
Tower	0.5
Site and Facilities	1.0

Yearly operating cost was estimated as 1.5% of direct capital cost, from the operating cost experience of Northeast Utilities.

3.3.3.7 System Evaluation

The mathematical model allows estimation of the direct capital cost, as well as total yearly operating costs. For comparison and evaluation purposes, unit costs are better suited. Direct capital cost is thus referred to rated power output to obtain $/KW. Similarly, the total yearly operating cost is referred to system yearly net energy output to result in unit energy cost, ¢/KW-hr.

Energy cost was the prime parameter of interest in the development of the model, although as the study progressed, it became clear that utilities also are concerned with the actual cash outlay level of the WGS (capital cost). However, in the system optimization and evaluation, other parameters were also used, particularly plant utilization and physical sizes.

Plant utilization parameters include start-up and shutdown wind speeds, rated wind speed and plant factor, or effective percent of energy actually produced by the WGS referred to the maximum possible (if the WGS ran at rated power continuously). Also important is the percent time the unit is on-line, producing power. These parameters are important to a utility from a capacity planning and plant type mix strategy it will pursue in the future, and should have some influence on the choice of design parameters. (These issues are discussed in Section 9.)

Physical sizes and ratings of components are also a factor in system optimization and planning. Rotor diameter is, perhaps, the best single parameter to express technical risk in WGS development, since the rotor represents the greatest technical challenge in the development of utility compatible WGS designs. Tower height is another parameter of importance which, while it does not represent technical risk, does impact public acceptability and public safety (see Section 9).

These and other parameters calculated by the model must be considered in parallel with energy and capital costs when evaluating system designs. Therefore, the model must be exercised with all of the significant parameters to the degree appropriate to the particular study. How the model was used (and not used) for this purpose is discussed in paragraph 3.4, Results of Parametric Analyses.

3.3.4 Computer Implementation

The WGS is represented by a lengthy and complex mathematical model. As anticipated, a number of variations, modifications and improvements were made to the model during the course of this study. For this reason, the model was programmed using the Kaman-developed ZODIAC II computer language.

3.3.4.1 ZODIAC Description

ZODIAC is a general computer program which allows calculations to be programmed in a very flexible and easy to use language. It does not presuppose any input format, equations are easily modified or added, and output is printed out by a simple print statement with values automatically identified. General expressions are essentially the same as with FORTRAN. Further detail is found in the ZODIAC II User's Guide in Reference 3-1. An updated version of this document has been submitted to NASA Lewis Wind Energy Project Office. A previous application of this tool can be found in References 3-1 and 3-2.

ZODIAC provides a flexible calculating tool for the engineer. Many of the characteristics of FORTRAN which tend to make minor or major changes in a model or in the logic difficult are improved upon. These characteristics include format statements, argument lists and common allocation. In addition, automatic table look-up and iteration and numerous logic checks are included.

Programs executed using ZODIAC II are slower in CPU time than programs written in FORTRAN; however, calendar time required for programming, debugging and modification is reduced manyfold.

3.3.4.2 ZODIAC Implementation

ZODIAC II is characterized by a control module, data, tables and any number of modules.

Control module - The control module calls out certain groupings (modules) of equations to be evaluated. In addition, control modules permit simple arithmetic operations, GO TO statements, READ, PRINT and ITERATE statements. In this application, all data are printed out through the control module so as to achieve uniformity in format.

The program flow becomes at once apparent by looking at the control module as seen in the listing in Appendix B. The Program Flow Diagram, Figure 3-20, closely parallels the listing; the difference being that the Program Flow Diagram omits some expressions which are used to save constants for printout.

Tables - ZODIAC II performs automatic table look-up. The tabular data are entered in an easy to read form. In this program, two tables are used, presenting wind occurrence statistics and rotor aerodynamic efficiency. Data are entered in sufficiently small increments to minimize interpolation errors for nonlinear relationships.

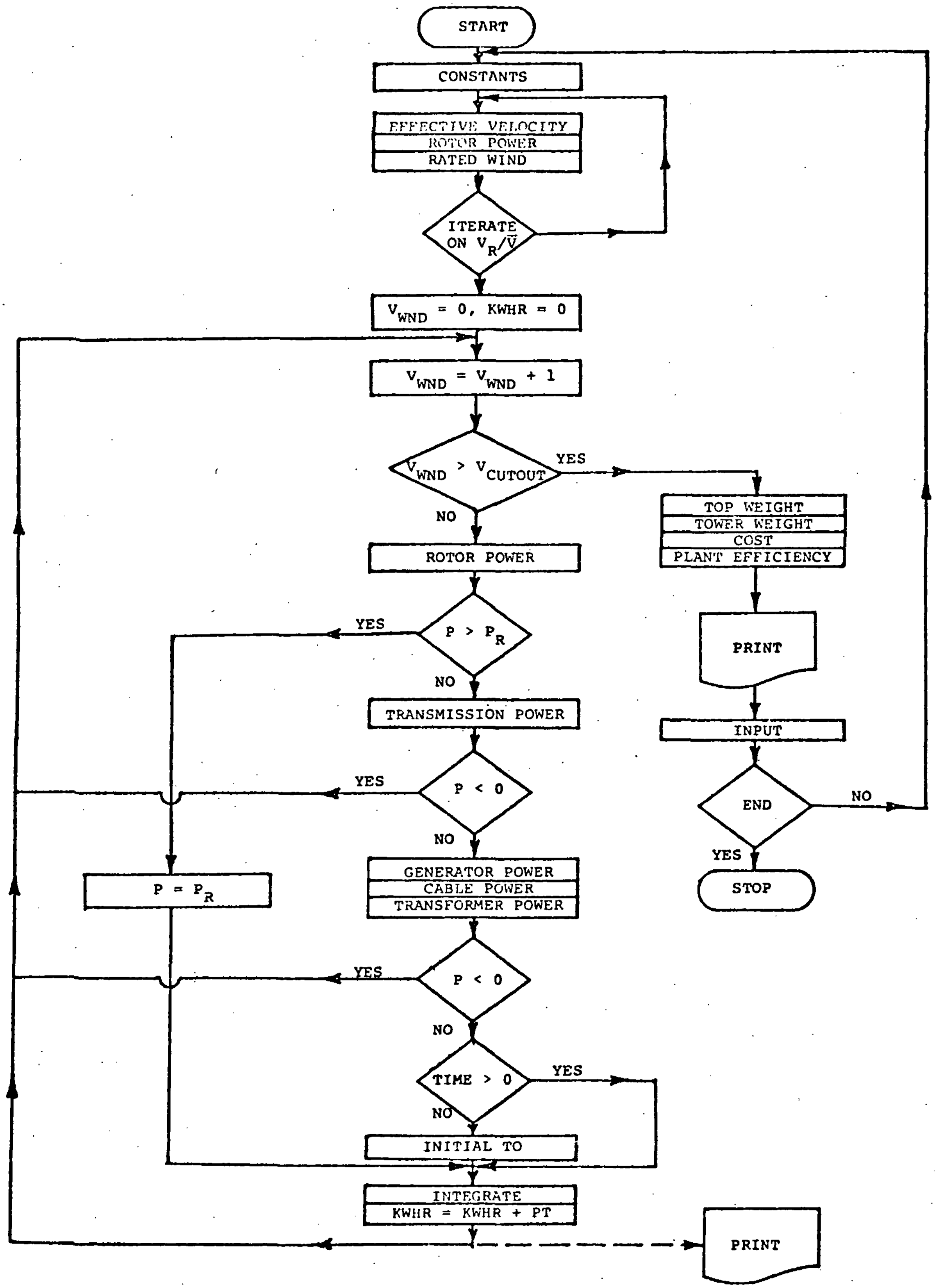


Figure 3-20. ZODIAC II WGS Program Flow Diagram

Data - Input items most likely to vary are separated from the rest of the program and specified for the baseline case. This speeds setting up cases and printout truncation.

Modules - Most of the program consists of equations. These are divided into groups called modules. Each module represents a set of related equations and is given a representative name. In addition to evaluation of equations, the module can perform simple logic, automatic iteration and automatic table look-up. A given module may be recalled repeatedly, much like a subprogram.

The listed modules in Appendix B are in the order in which they are first used; however, they may be in any order. All modules (and also the control module) have COMMON statements in which all variables used in other modules must be listed. Within a given module, however, a name used in another module is permissable.

Nomenclature - Variable names can contain up to four characters. Because of the great number of variables used in this program, a nomenclature was adopted in which the first letter identifies the category of the variable (weight, cost, etc.). Appendix B illustrates this further.

3.3.4.3 Methodology

The function of the mathematical model is, broadly, to do the following:

Size components

Integrate yearly energy output

Determine weights and cost

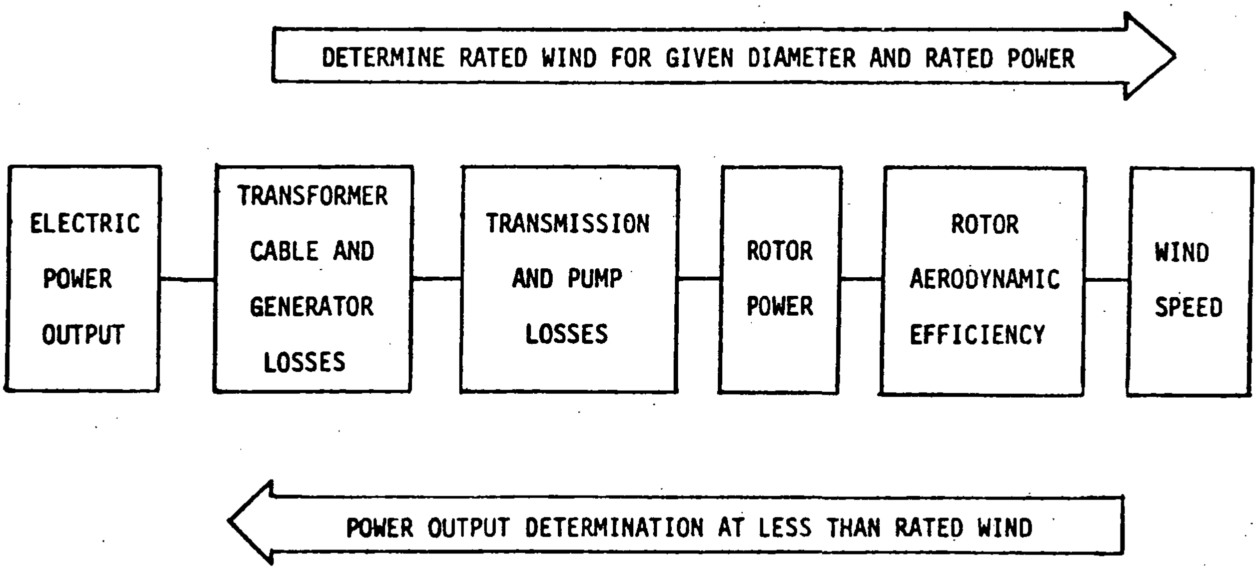
Calculate evaluation parameters.

Detail arrangements can vary, depending upon requirements, and different versions were devised for the study. The particular program listed in Appendix B is an arrangement devised to evaluate the system when the rotor diameter is specified. It is the model which leads to the 1500 KW selected WGS design. Besides the listed program, the other significant version constructed is one in which $V_R/\overline{V}$ is specified and the diameter is determined to correspond.

The following discussion refers to the model as presented in Appendix B and summarized by the flow diagram of Figure 3-20.

Component Sizing - Component sizing is effected by modules "CONSTANTS," "EFFECTIVE VELOCITY," "ROTOR POWER," and "RATED WIND." Most dimensions and losses are determined in the first module. The last three are used to determine the rated wind speed by an iterative process. Here, V_R is the wind speed at which rotor capabilities match the system rating. The process is illustrated by the schematic below:

CALCULATION PROCEDURE



Yearly Energy Output - Output power at any wind speed is calculated using modules "ROTOR POWER," "TRANSMISSION POWER," "GENERATOR POWER," "CABLE POWER," and "TRANSFORMER POWER."

The part power losses of electrical components are defined, customarily, in terms of output power. The modules were set up consistent with this, making it easy to change should substitution of new values be desired. Since the input to the module is component input power, the part power losses have to be calculated by an iterative procedure.

The process of output power determination is the reverse of that used to determine V_R, as illustrated above. Wind speeds are taken in .45 m/s (1 mph) increments from zero to cut-out wind speed. The module "INITIAL TO" determines time of occurrence of wind speeds at which power output, and thus energy output, is positive. The integrated value of all positive energy outputs is the yearly energy output.

Weights and Costs - Weight of components above foundation is determined from modules "TOP WEIGHT" and "TOWER WEIGHT"; the latter contains an iterative process because the tower weight itself has an effect on the size of members. The "COST" module determines all component and operating costs.

Evaluation Parameters - The last module, "PLANT EFFICIENCY" adjusts gross energy output for "housekeeping" losses and combines it with costs to determine energy unit costs. Plant factor is also determined.

3.4 Results of Parametric Analysis

The mathematical model described in previous sections was used to optimize all of the major system component parameters for minimum energy cost. Tradeoff and sensitivity studies were also conducted to further understand relationships of the WGS parameters to the system and its components. This investigation lead to recommendations for the parameters of the WGS preliminary design. Subsystem cost trends are shown in Figures 3-54 to 3-59, Section 3.7.

3.4.1 Major System Parameters

The essential characteristics of a WGS are described by median wind speed of the site ($\overline{V}$), rated output power of the system (P_R) and rated wind speed (V_R). Median wind speed is indicative of the inherent energy available at the site. Power rating determines the scale of the drive system and electrical components. Rated wind speed, the lowest wind speed at which the system achieves its design rating, determines rotor size. Thus, these three parameters determine the size of the WGS and its yearly energy output.

The ranges of some of the parameters studied are given in Table 3-11. These ranges were chosen to bracket the optimum conditions. When examining these major parameters, other system parameters were held at constant values determined to be near their optimum values or near their practical limits from preceding investigations. These are also listed in Table 3-11.

Figures 3-21 and 3-22 show energy cost (¢/KW-hr) and direct capital cost ($/KW) as functions of $\overline{V}$, P_R and $V_R/\overline{V}$ ($V_R/\overline{V}$ is a convenient parameter for rated wind speed). From Figure 3-21, it is seen that median wind speed has the major effect on the energy cost of a WGS. However, at a given $\overline{V}$, energy cost does not vary much with P_R or $V_R/\overline{V}$ when near its minimum value. This suggests that P_R and V_R (or diameter) may be selected at other than minimum energy cost conditions from other considerations with only a small energy cost penalty. Figure 3-22 shows that since direct capital cost improves with increasing $V_R/\overline{V}$ (the result of the rotor being smaller and its cost reduced), direct capital cost reduction could be achieved by accepting a small energy cost penalty.

Of all the results obtained in the parametric analysis, the data shown in Figure 3-21 are the most significant. The obvious conclusion which was drawn about these results can be stated in another way: in the region of its optimum (minimum) energy cost, considerable flexibility in the selection of system parameters (direct capital cost, plant factor, rotor diameter, cut-in wind speed, etc.) can be exercised with minimal impact on energy cost. This gives the system designer wide latitude in the selection of these variables to meet particular application requirements, technical development risk reduction, unit cost goals and other important factors. Figures 3-23 and 3-24 show plant factor vs direct capital cost and energy cost for the 5.4 m/s (12 mph) and 8.0 m/s (18 mph) median wind speeds, respectively. Optimum design values for minimum energy cost are shown also. Higher plant factors are associated with lower rated wind speeds and larger rotor diameters in proportion to the WGS power rating.

TABLE 3-11. PARAMETERS FOR SYSTEM OPTIMIZATION

MAJOR SYSTEM PARAMETER RANGES		
Rated Power		50 to 3000 KW
Rated Wind Speed, V_R		1 to 2 times Median Wind Speed
Median Wind Speed, $\overline{V}$		3.6 m/s(8 mph) to 10.7 m/s (24 mph)
BASELINE VALUES		
Site:	Maximum Wind Speed	53.6 m/s (120 mph)
	Ambient Temperature	15°C (59°F)
	Elevation	SL
Rotor:	Diameter	Calculated for required V_R
	Equivalent Solidity	.03
	Tip Speed	Selected to obtain maximum aerodynamic efficiency at .8 V_R
	Number of Blades	2
	Twist	Optimum
	Planform Taper Ratio (Effective)	3:1
	Root Cut Out Fraction	.1
	Airfoil Section	230 XX
	Airfoil Thickness	Variable
	Airfoil Surface Roughness	Standard
	Inclination to Wind	10°
Drive Efficiency		97%
Generator Speed		1800 RPM
Cable Cross Section		Standard

TABLE 3-11. PARAMETERS FOR SYSTEM OPTIMIZATION (continued)		
Tower:	Height	For 15 m (50 ft) Rotor/Ground Clearance
	Height to Base Width Ratio (Aspect Ratio)	4
	Bending Frequency/ Rotor RPM Ratio	1.5
	Torsional Frequency/ Rotor RPM Ratio	2.5
Costs:	(% of Initial Cost) Carrying Charge Rate	15%
	Operations	1.5%
	Maintenance:	
	Rotor	10%
	Power System	3.5%
	Tower	0.5%
	Site & Facilities	1.0%
Component Life (Years):		
	Dynamic Components	30
	Static Components	50

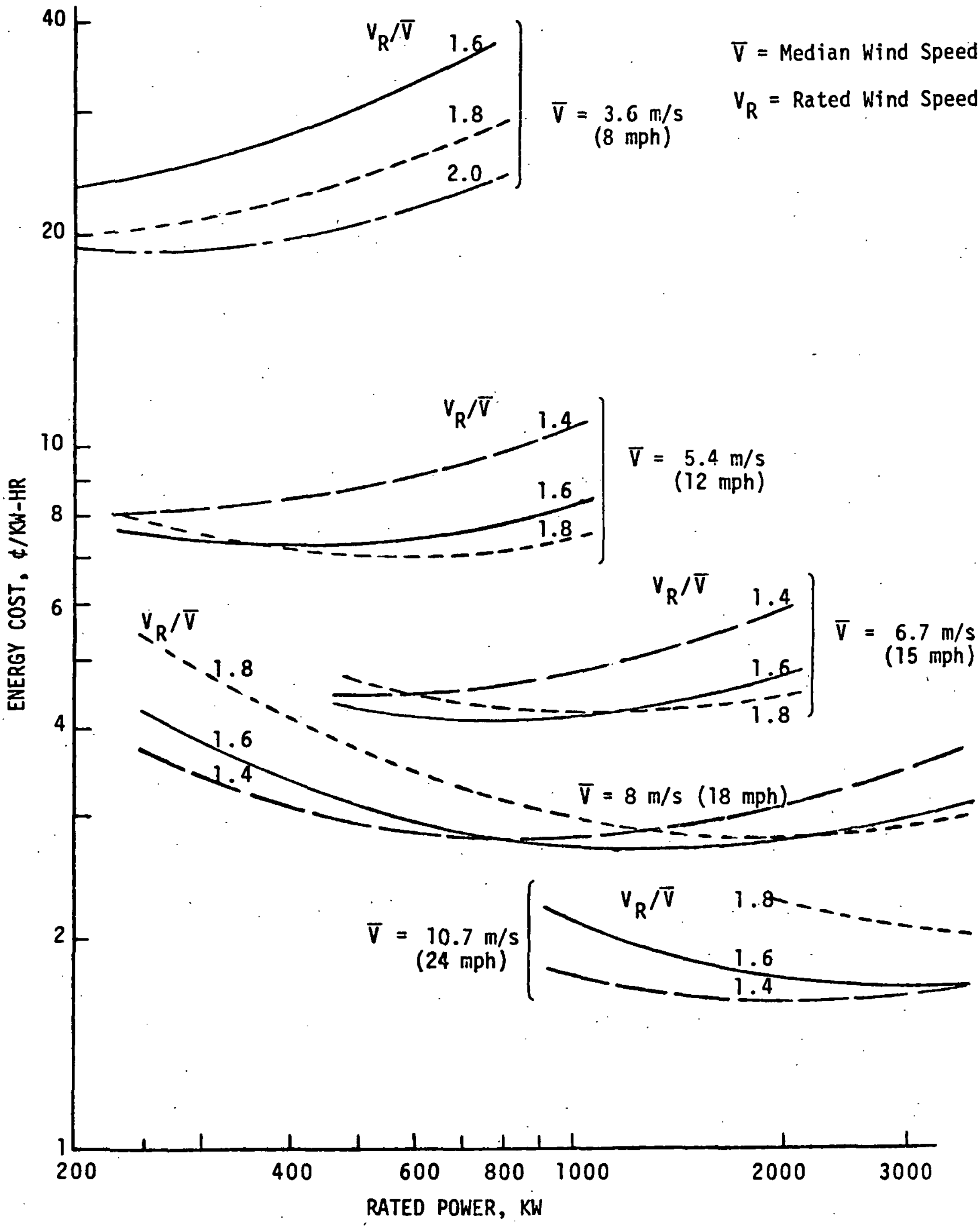


Figure 3-21. Energy Cost Variation With Major Parameters

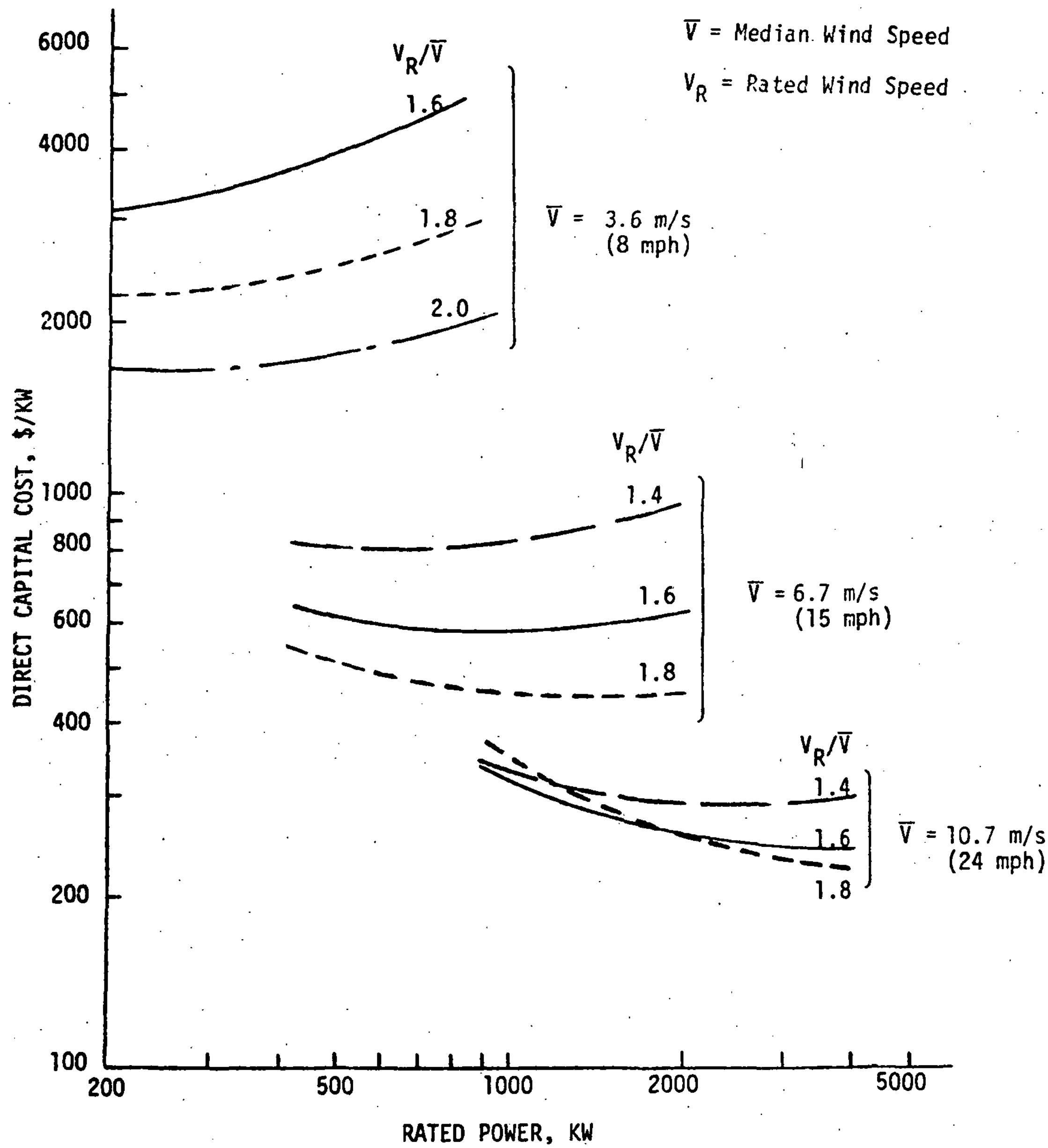


Figure 3-22(a). Direct Capital Cost Variation With Major System Parameters

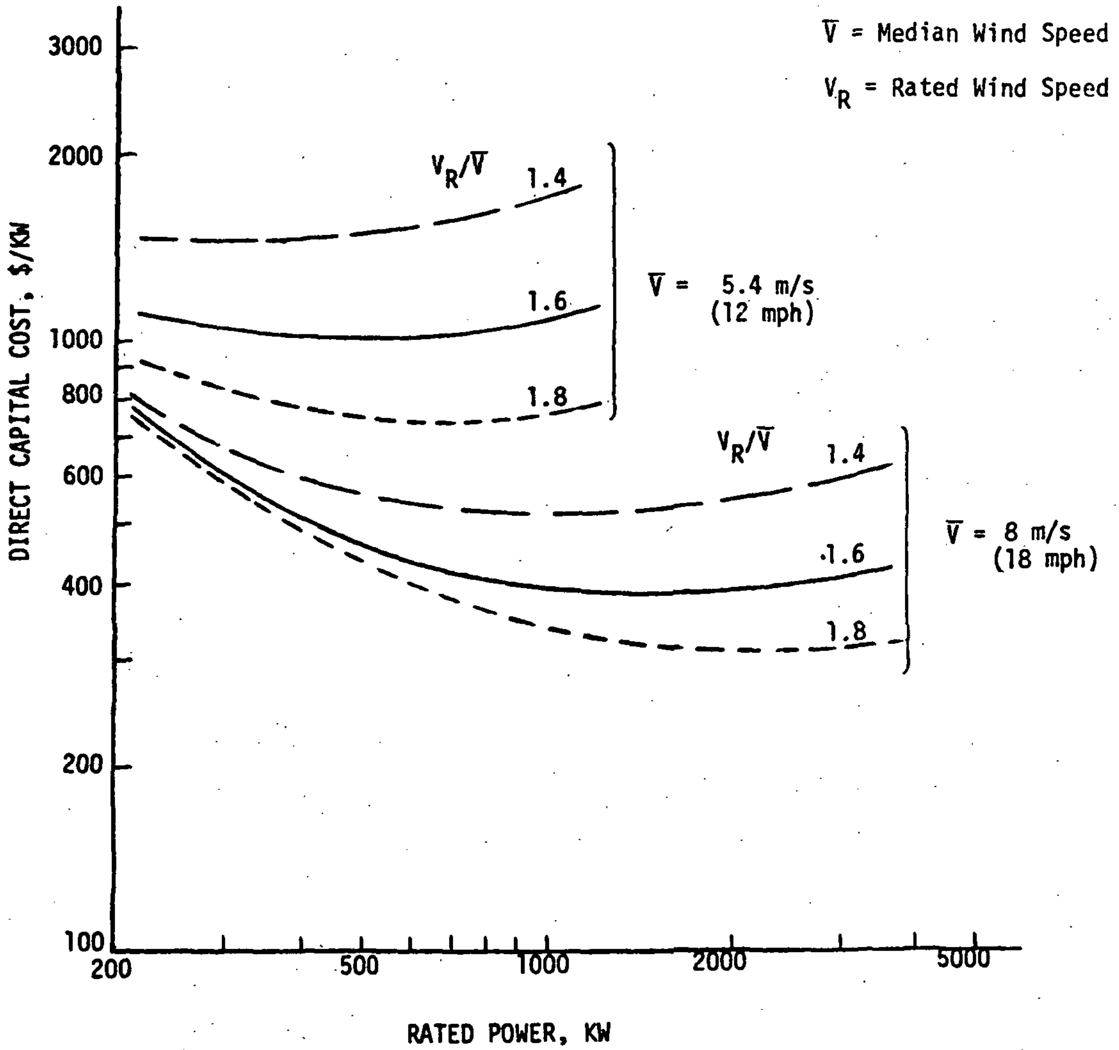


Figure 3-22(b). Direct Capital Cost Variation With Major System Parameters

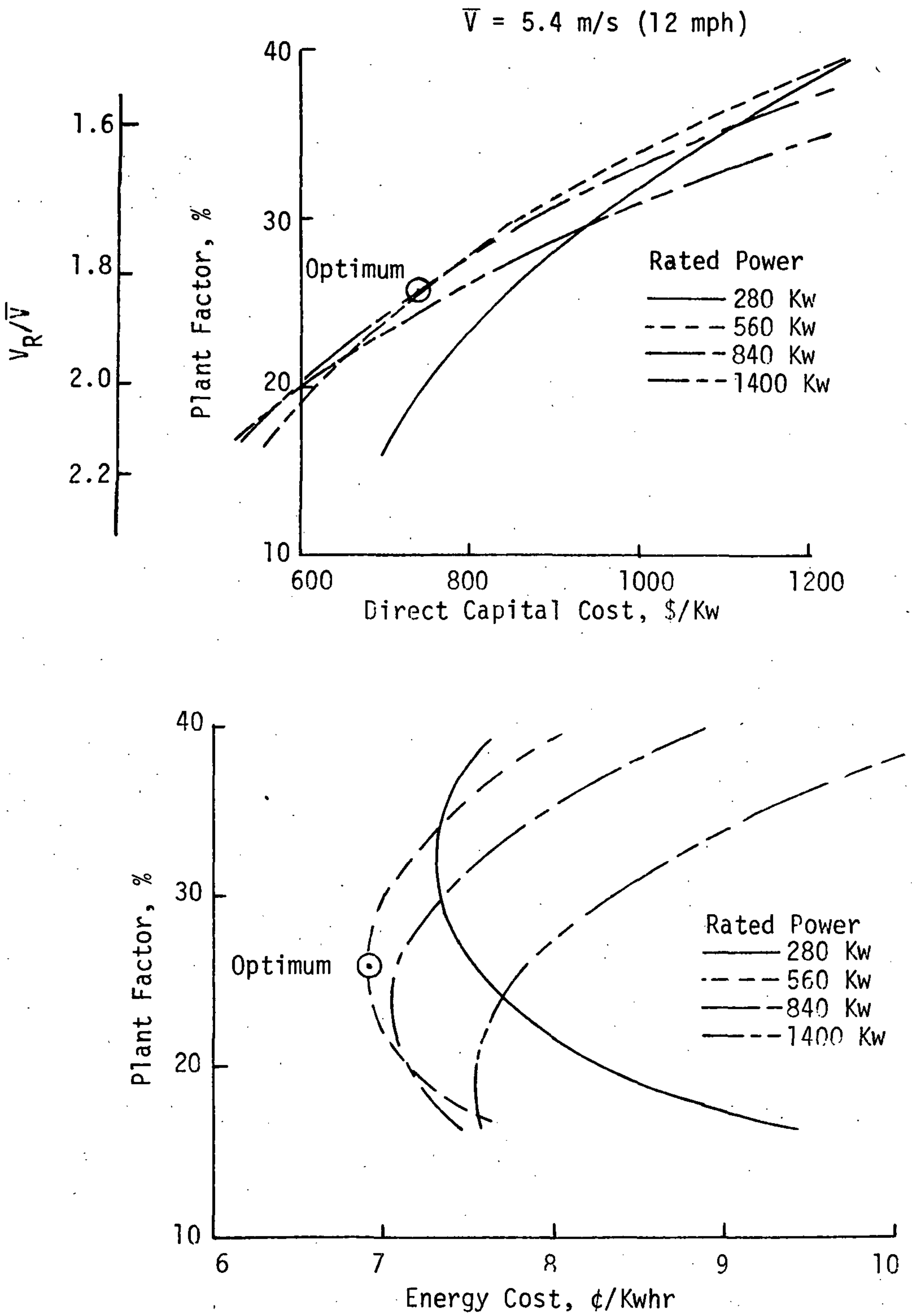


Figure 3-23. Plant Factor Sensitivity to Cost

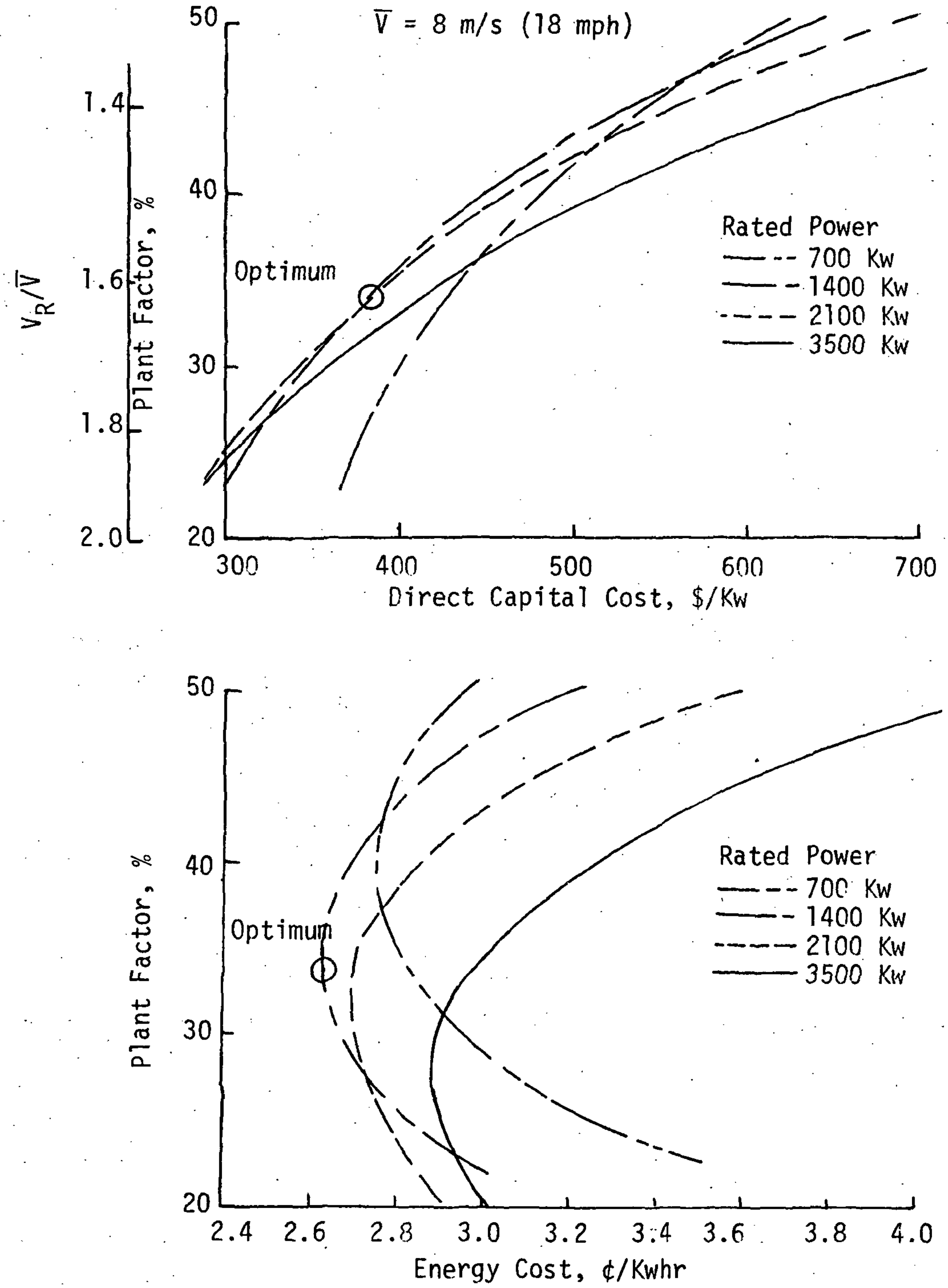


Figure 3-24. Plant Factor Sensitivity to Cost

In paragraph 3.3.3.4, it was illustrated that the two separate rotor models used in the study had significant weight (and hence, cost) differences between them. This resulted in significant differences in energy cost and capital cost characteristics for the WGS using one or the other rotor model. The data shown here are for the current WGS model using the improved rotor model. However, it should be noted that the conclusions drawn about $\overline{V}$, V_R and P_R effects on system characteristics above were found to be true using the earlier rotor model as well. Hence, it appears that these conclusions are good guides for decision making for the WGS parameters.

3.4.2 Optimum Systems

The preceding data can be used to determine optimum P_R and $V_R/\overline{V}$ when using minimum energy cost as the criterion for optimization. Figures 3-25 to 3-26 show these optimum values (for minimum energy cost) and related parameters as a function of $\overline{V}$. Energy cost is seen to decline significantly with $\overline{V}$ because of the increase in wind energy available. The rated power is increased and, correspondingly, the unit direct capital cost is decreased since the optimum rating grows faster with $\overline{V}$ than do the total direct capital costs. Rotor diameter is nearly constant regardless of wind speed as a result of opposite trends of optimum rated power and optimum rated wind speed ratio shown in Figures 3-25 and 3-27, respectively. Optimum $V_R/\overline{V}$ reduces with $\overline{V}$; although V_R increases with $\overline{V}$, it increases at a lesser rate. Plant factor increases with $\overline{V}$ as the result of greater time occurrence of winds in the operating envelope of the system.

The data in Figures 3-25 to 3-27 show considerable change in system parameters of rated power, rated wind speed, etc., as the site median wind speed is changed. However, for the same reasons mentioned in the previous section, substantial changes in these values at a given median wind speed are possible without significantly affecting energy cost. This is important to note since paragraph 3.5, Site Adaptability Studies, reports the results of studies which determine that a given system designed for a given site (e.g., designed for a 5.4 m/s [12 mph] median wind speed site) is not far off optimum if used at another site (e.g., a 6.7 m/s [15 mph] median wind speed).

3.4.3 Major Component Parameters

Major component parameters are those which have significant effect on WGS performance, weight and cost. These include rotor tip speed, rotor solidity, number of blades and tower height. Selection of optimum values for these parameters is described below.

3.4.3.1 Rotor Tip Speed

Rotor tip speed affects WGS yearly energy output and capital cost of drive system components sensitive to rotor torque. Higher rotor speeds require lower torque and, consequently, lower cost drive system components. Yearly energy output is maximized by the selection of a rotor tip speed which maximizes aerodynamic efficiency over the particular range of wind speeds in which the rotor will operate. Figure 3-9 shows, for a given rotor solidity, that maximum efficiency occurs at a specific velocity ratio, V_{TIP}/V_{EFF}. The effective velocity,

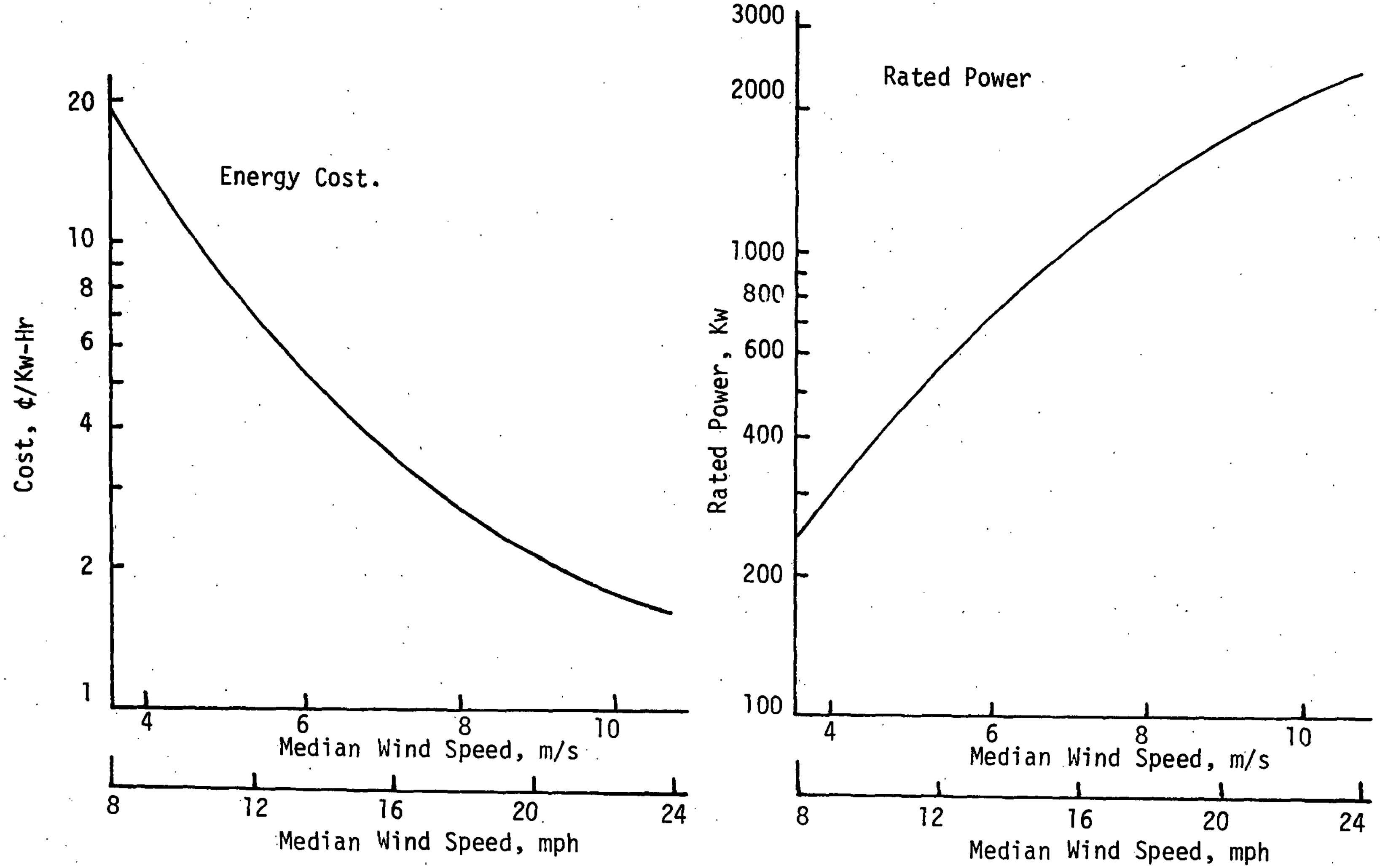


Figure 3-25. Rated Power and Energy Cost for Minimum Energy Cost Systems

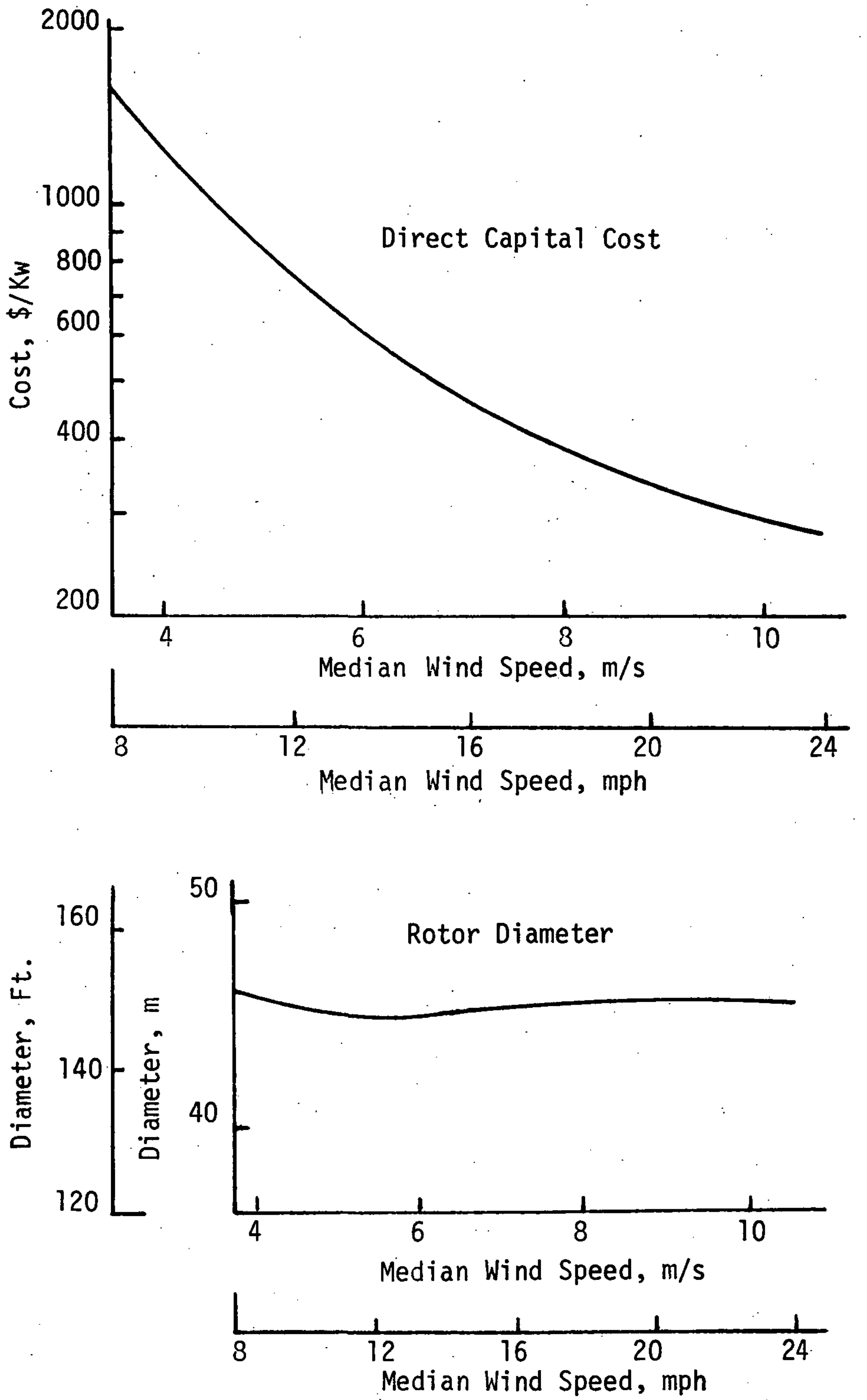


Figure 3-26. Direct Capital Cost and Rotor Diameter for Minimum Energy Cost Systems.

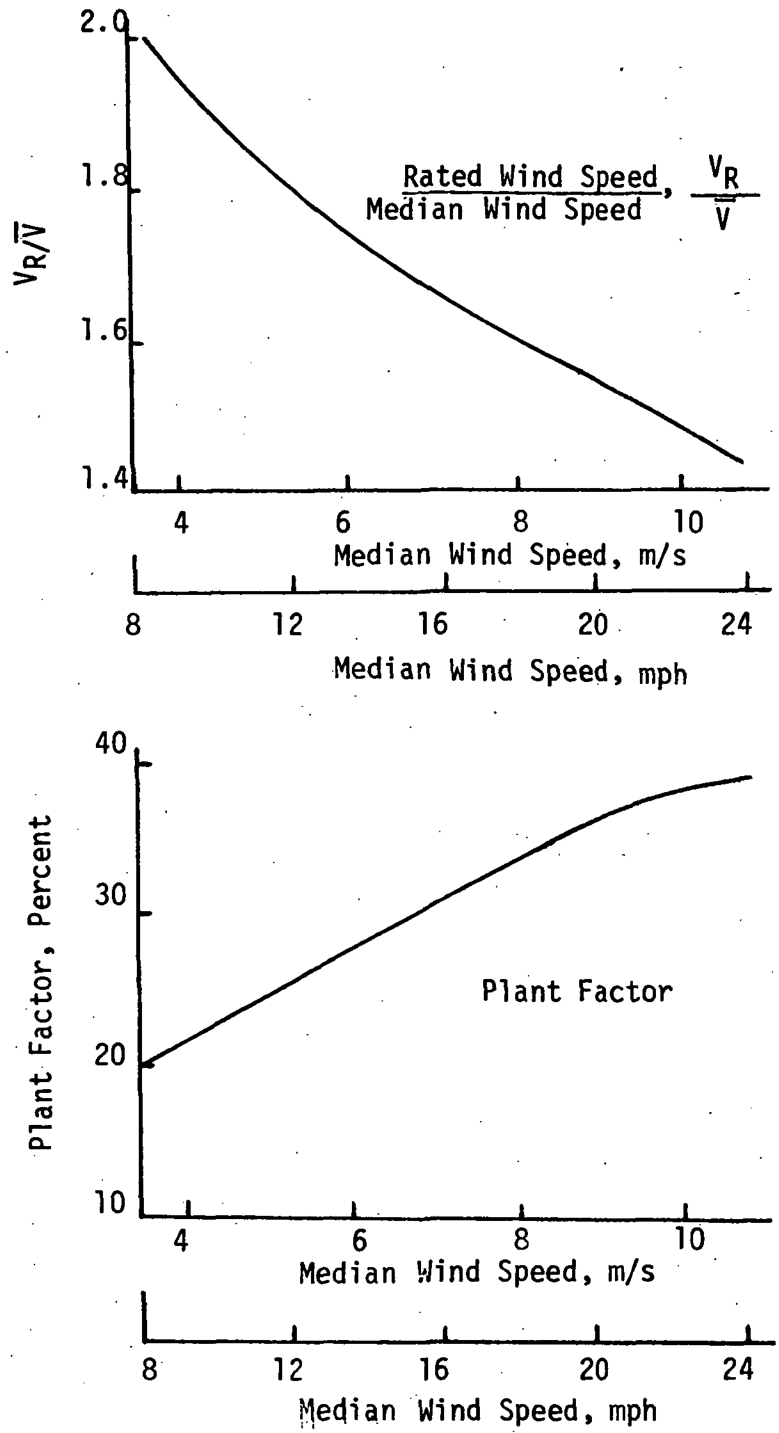


Figure 3-27. Plant Factor and Rated Wind Speed for Minimum Energy Cost Systems.

V_{EFF}, is defined in paragraph 3.3.3.1 as a function of wind velocity at the 9.1 m (30 ft) reference height, V_{REF} (or V_{WND}). The design speed, V_D, is that selected V_{REF} at which aerodynamic efficiency is maximized. The parameter V_D can be expressed in terms of rated wind speed, V_D/V_R, to generalize the analysis for any WGS site.

Results of the study to optimize V_D/V_R are shown in Figure 3-28, in which the optimum V_D/V_R is seen to occur at .76 and .77 for the two $\overline{V}$ conditions investigated. These values were used for final preliminary design sizing although most investigations are based on $V_D/V_R = 0.8$.

Figure 3-29 shows how the system power output profile and direct capital costs vary with V_D/V_R for a fixed system rating and rotor diameter (1500 KW and 54.9 m being the values selected for the high power preliminary design WGS). As shown, capital costs decrease with increasing tip speed, primarily because drive system torque is reduced. However, as tip speed (and V_D/V_R) are increased, the power output profile of the system steepens, resulting in less total energy production. This counterbalances the reduced cost of the system, resulting in higher energy cost.

The results shown in Figure 3-29 illustrate another common occurrence in WGS optimization studies: changes that tend to reduce capital costs (and hence, operating costs) often reduce energy output. Hence, care must be exercised in evaluating cost reducing parameter changes to insure that corresponding energy output reductions do not negate or even reverse energy cost reductions.

3.4.3.2 Rotor Solidity

Solidity has no significant effect on rated wind speed or plant factor.

The effect of rotor solidity on energy and direct capital costs is shown in Figure 3-30. The data of Figure 3-30 show that both energy and direct capital costs are reduced as rotor solidity is lowered. This result was, as mentioned before, the impetus to examine the limits of rotor solidity and diameter for the WGS. This study did not find any inherent technical limits to solidities as low as 0.02 but, as shown in Figure 3-30, the incentive to reduce solidity below 0.03 is very small. Considering technical risk, the 0.03 value was selected for the preliminary designs.

The data of Figure 3-30 are for optimized systems (power rating, rated wind speed, rotor diameter, etc., selected at minimum energy cost). The effect of solidity on fixed system designs is more pronounced, as shown in Figure 3-31. Here, the effect of solidity on the selected preliminary designs is presented.

3.4.3.3 Number of Blades

Having established cost advantages of minimum solidity and the necessary rotor diameter to provide rated power at rated wind speed, it now remained to select the number of blades. This involved consideration of blade loading and aspect

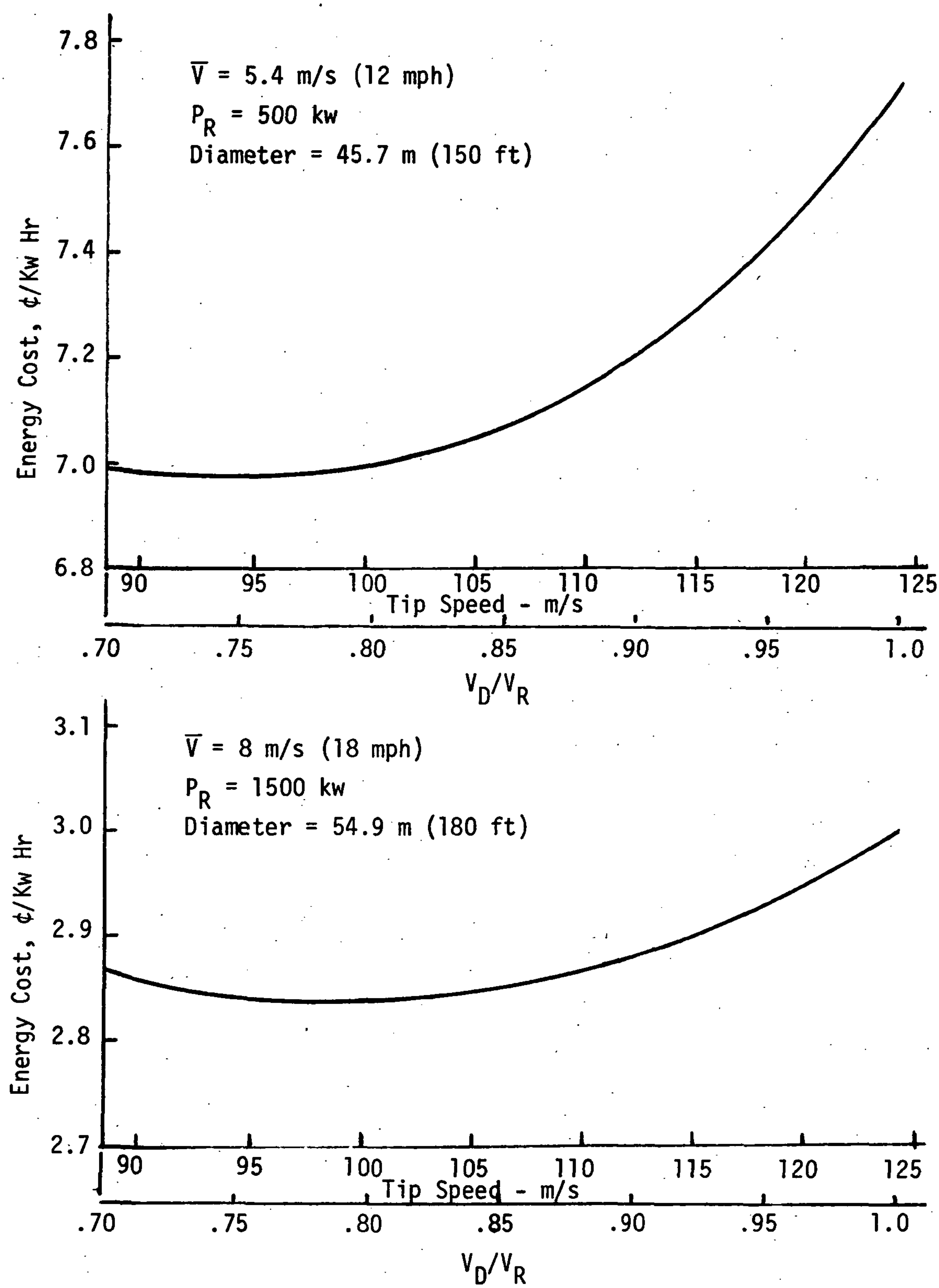


Figure 3-28. Tip Speed Optimization - Energy Cost.

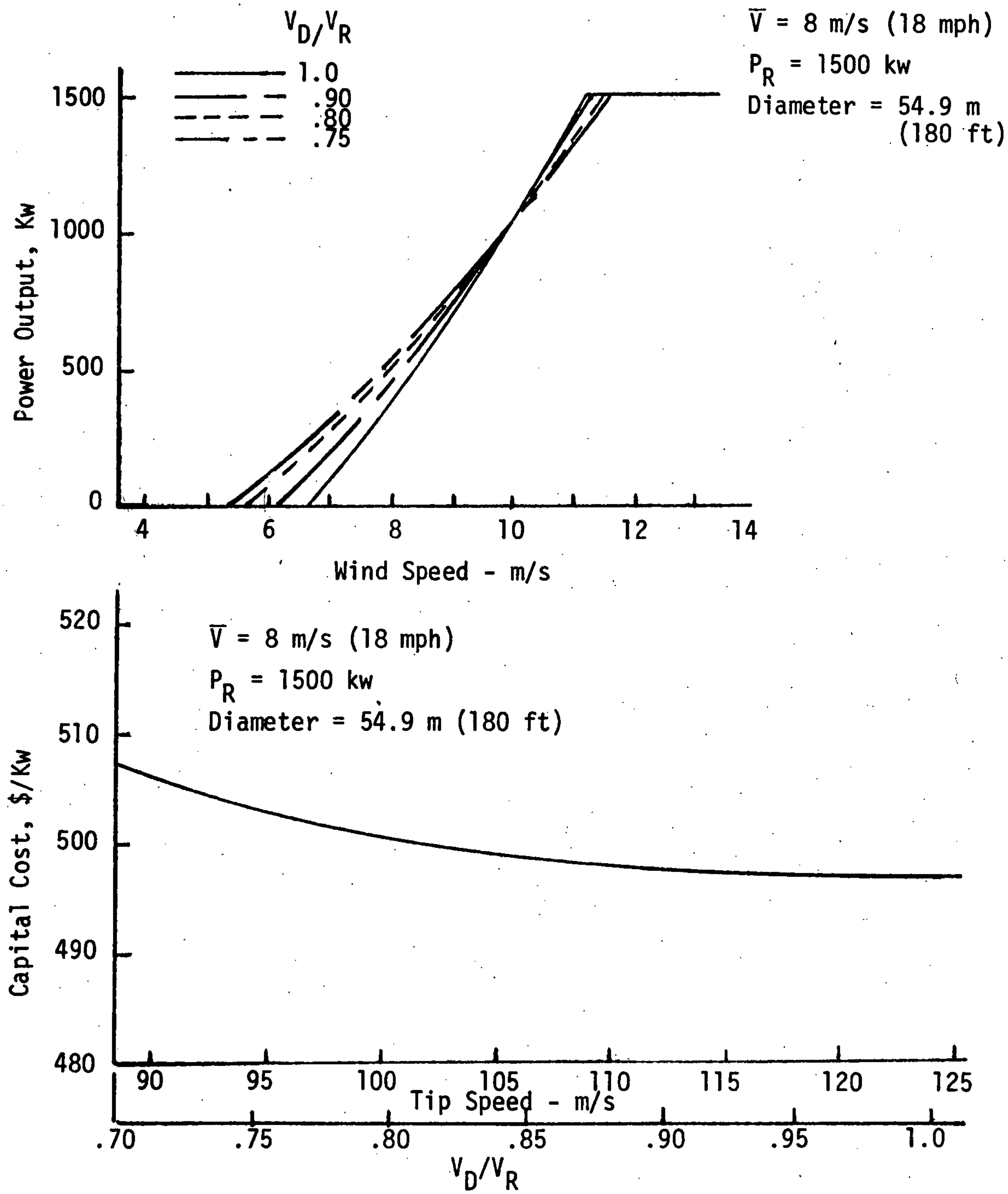


Figure 3-29. Tip Speed Optimization Study
Capital Cost and Power Output

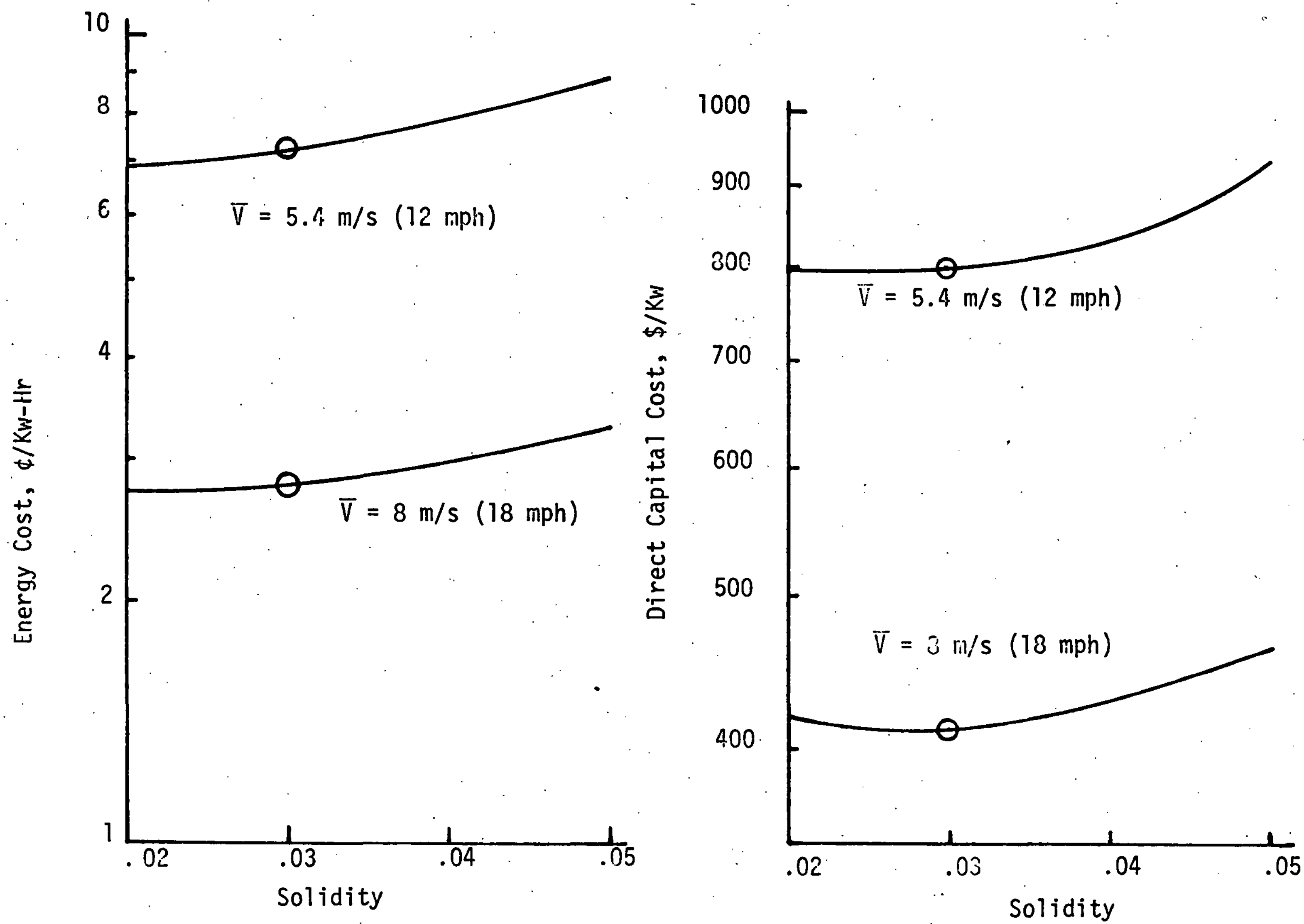


Figure 3-30. Rotor Solidity Effect on Costs of Optimum Systems.

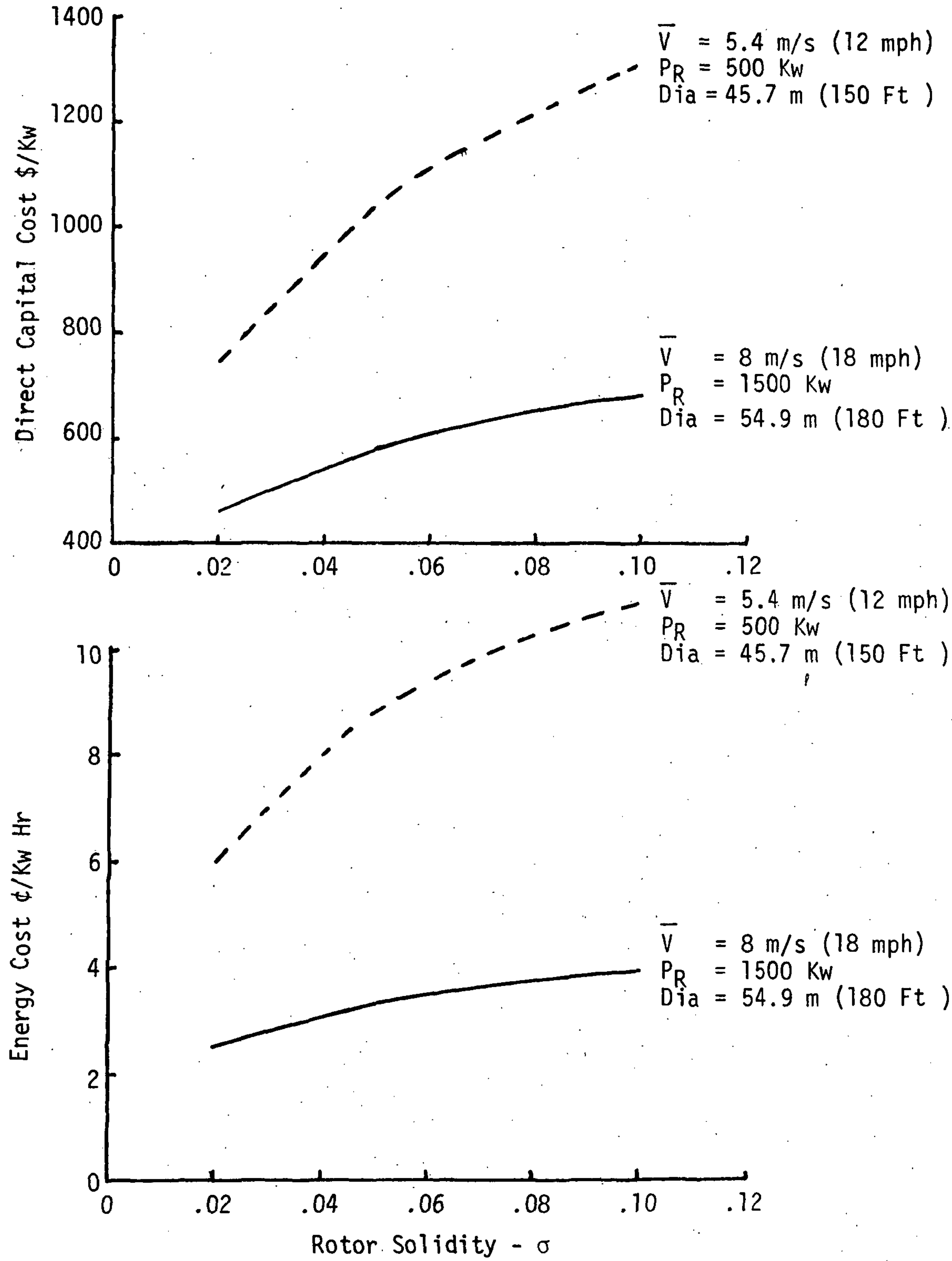


Figure 3-31. Rotor Solidity Effect - 2 Bladed Rotor

ratio (slenderness). In the case of helicopters, the number of blades is usually established to maintain blade loading within a limit of approximately 5000 N/m^2 (100 lbs/ft^2). The WGS blade loading is already very low, in the order of 1440 N/m^2 (30 lbs/ft^2). Aspect ratio, therefore, became the limiting technical factor from a structural standpoint and a two-bladed rotor is already considered to be at the practicable aspect ratio limit to provide the necessary blade strength and tuning.

Therefore, the addition of another blade with the same aspect ratio adds to total blade area (solidity) and attendant component cost and resulting energy cost. It should be noted that one advantage of a higher number of blades is reduced vibratory load input to the drive shaft, transmission, tower and other components. These higher loads in a two-bladed rotor are not difficult (costly) to handle; larger gears, heavier hub sections, more tower steel are all low cost tradeoffs against the substantial increase in cost of three blades versus two blades. Consequently, they were not found sufficiently important to outweigh the above energy cost disadvantages of the three-bladed rotor. The vibration characteristics of the selected two-bladed rotors were analyzed so that proper response characteristics would be built into the blades, drive system and tower.

The results of analyses on the number of rotor blades are shown in Figure 3-32. The figure gives energy and capital costs, and plant factor for a two-bladed rotor WGS and a three-bladed rotor WGS for the nominal power ratings and rotor diameters of the preliminary designs. As noted, the three-bladed rotor has a higher solidity to insure that it does not exceed the maximum blade aspect ratio established by previous studies. The blade aspect ratio for the three-bladed rotor was kept the same as the two-bladed rotor, resulting in a 50% increase in solidity.

The results indicate the three-bladed rotor WGS energy and capital costs are significantly higher than the two-bladed WGS. The main reason is the higher cost of the rotor due to the additional blade. Also, the higher solidity of the three-bladed rotor results in lower rotor tip speeds and higher torque, which increases drive subsystem costs. Hence, the two-bladed rotor offers significant cost advantages over the three bladed rotor, and it was carried as the baseline concept for the remainder of the study.

3.4.3.4 Tower Height or Rotor Ground Clearance

Figure 3-33 shows various system parameters as functions of rotor ground clearance. Increased plant factor and rated wind speed are both a reflection of more energy available to the rotor as tower height is increased. Unit direct capital cost is increased because of the increased cost of the tower. Unit energy cost is the result of interaction of these variables and optimizes the rotor ground clearance between 15 and 45 m (50 - 150 ft). Final selection of ground clearance was made at 15.2 m (50 ft) for purposes of the preliminary design study and was used as a baseline value for subsequent investigations.

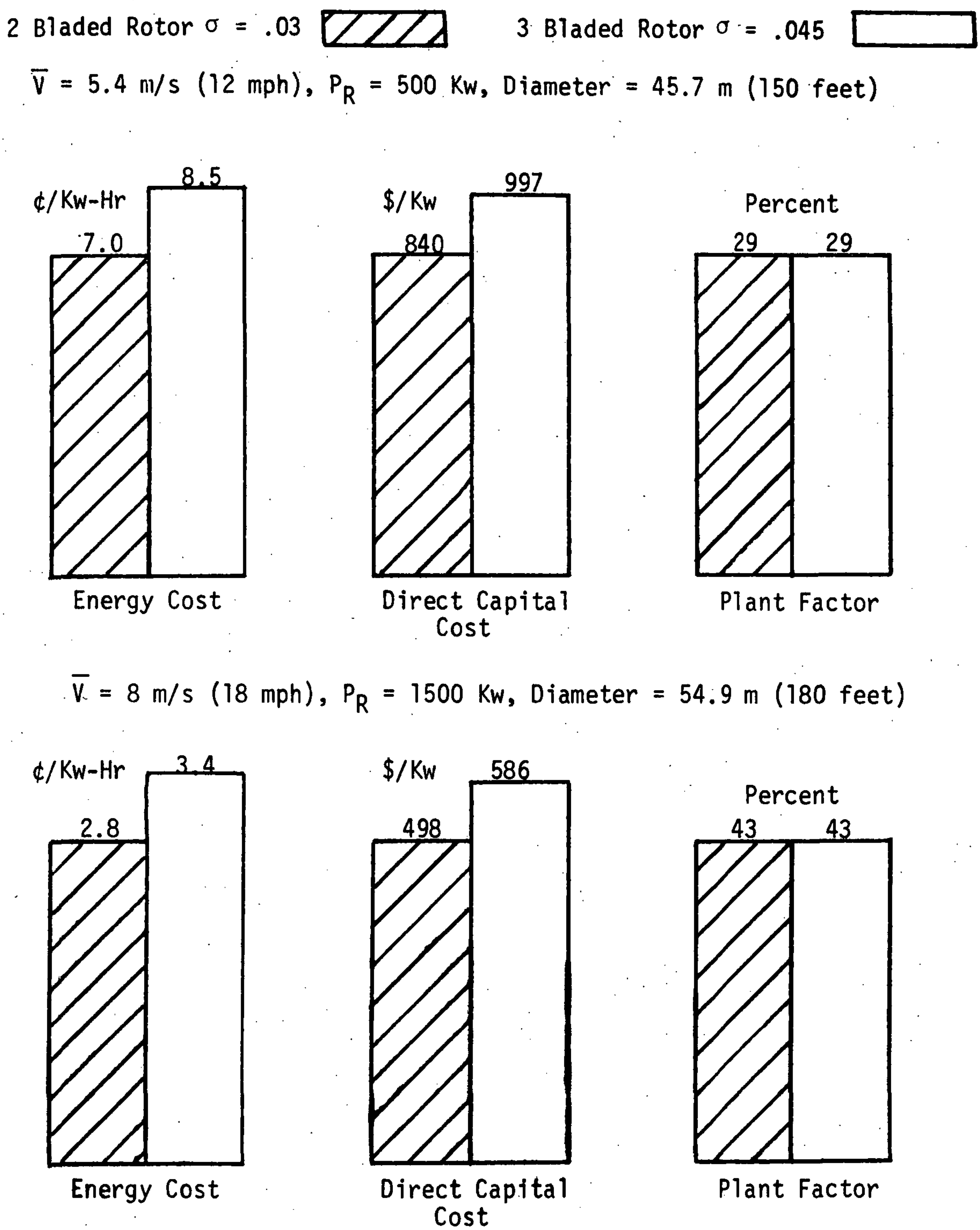


Figure 3-32. Effect of Number of Blades on System Parameters at Minimum Blade Aspect Ratio.

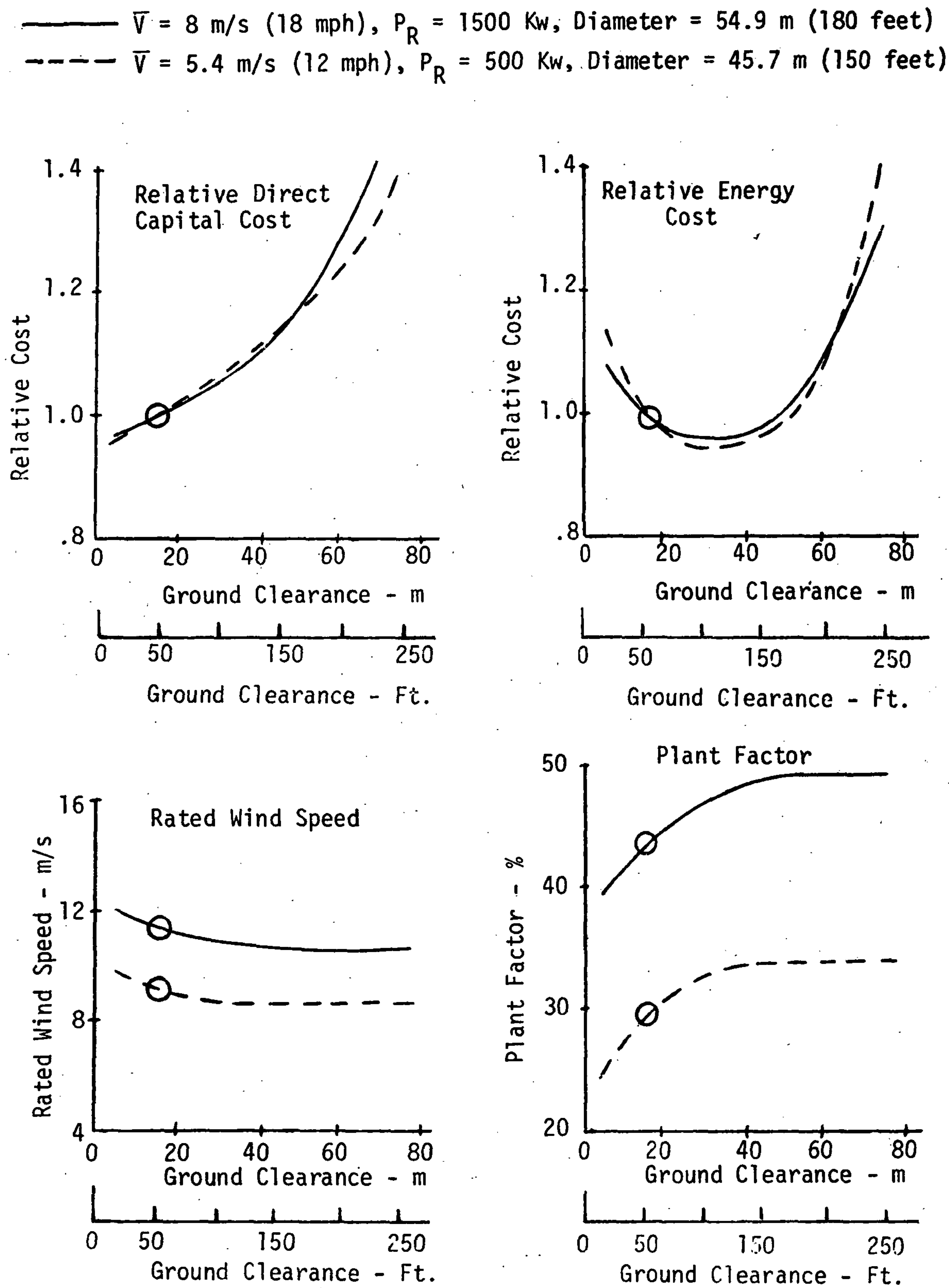


Figure 3-33. Ground Clearance Effect on System Parameters

3.4.4 Tradeoff and Sensitivity Studies

A number of component and system parameter tradeoff and sensitivity studies were conducted on other system parameters. Some of these parameters are true tradeoff or sensitivity study items, such as component efficiencies which always "optimize" at 100%. Others, such as blade aerodynamic properties, are of less importance to overall system selection than those above and are also treated as tradeoff or sensitivity items. In addition, sometimes these parameters are discrete, such as airfoil section, and are best handled as sensitivity parameters.

3.4.4.1 Rotor Geometry

The primary effect of rotor blade geometry differences, such as twist, thickness, taper, airfoil shape, etc., is to change the aerodynamic efficiency of the rotor and thereby the power output of the WGS. The flexibility of construction of composite blades makes them insensitive to cost and weight effects of geometry differences if the rotor diameter is held constant. However, if rotor diameter is resized for the changes in aerodynamic efficiency so as to hold WGS rated power and rated wind speed constant, then the WGS direct capital cost and energy cost will vary. This variation is defined by the parametric weight and cost relationships in the model.

Figure 3-34 shows the variation of cost, size and utilization parameters as a function of an incremental shift of the aerodynamic efficiency. In this sensitivity study, the aerodynamic efficiency map (presented in Figure 3-9) is increased (or decreased) at any velocity ratio by a fixed percentage increment. The data, presented in their normalized form in Figure 3-34, are representative of a broad range of WGS designs and include the effect of resizing WGS diameters to maintain rated power at rated wind speed.

The effect of changing blade characteristics on efficiency was determined using a program based on vortex theory. Table 3-12 lists these changes and their corresponding effect on cost and diameter (to adjust for efficiency changes) using the generalized data of Figure 3-34. As shown in Table 3-12, changes to airfoil, thickness, smoothness, twist, cut-out and taper have small effects on energy cost if properly selected. This confirms that cost and durability, not aerodynamic sophistication, are the prime items of concern for the WGS rotor.

Various conventional airfoil sections were selected for evaluation during the systems analyses phase of the WGS study program. Considerations, such as long periods of unattended operation and a 30 year component life requirement, led to selection of standard roughness airfoil data as more representative than smooth airfoil data for WGS application. This conservative approach accounts for gradual degradation of blade surface smoothness by such environmental effects as sand, grit, rain and hail erosion, particularly at the blade leading edge.

Initial performance evaluations included NACA 4412, 23012, 23018 and 63_2-615 airfoil sections. The section characteristics for these airfoils were obtained from the data presented in NACA Report No. 824 (Reference 3-3) under standard roughness conditions at a Reynolds number of 6 million, which is the approximate Reynolds number of the outer portion of the rotor blades. These data were

$\overline{V} = 8$ m/s (18 mph), $P_R = 1150$ KW, $V_R = 13$ m/s (29 mph)

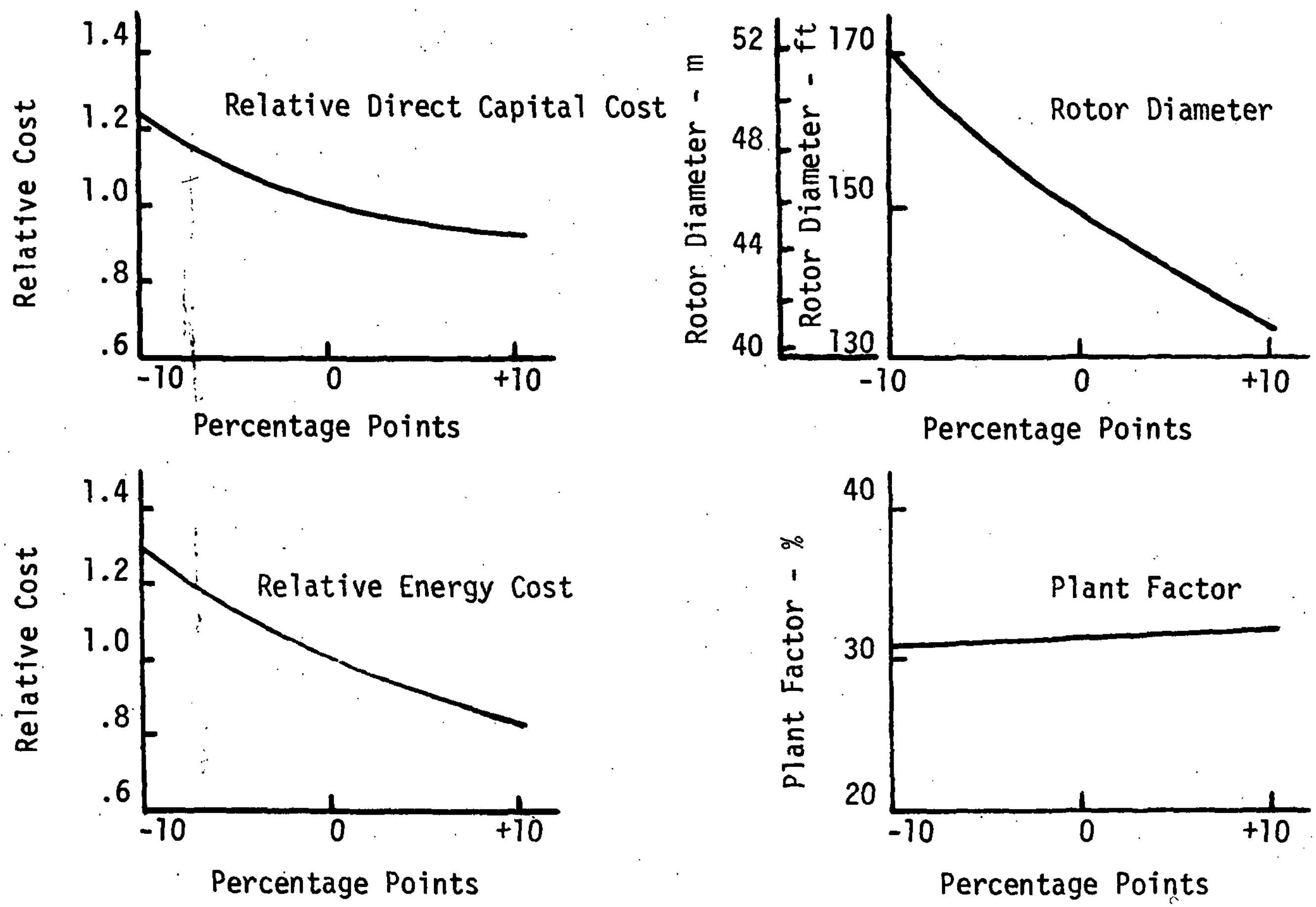


Figure 3-34. Sensitivity Effects of Incremental Changes in Rotor Aerodynamic Efficiency

TABLE 3-12. BLADE GEOMETRY EFFECTS ON ROTOR COST AND DIAMETER

	ROTOR PEAK EFFICIENCY PERCENT	PERCENT CHANGE FROM REFERENCE		
		ENERGY COST	DIRECT CAPITAL COST	DIAMETER
AIRFOIL SECTION (NACA)				
23012	38.6			
4412	38.6			
63-615	39.1	- .8	- 1	- .6
THICKNESS				
23012/23018*	38.			
23018 Constant	34.8	+ 5	+ 5	+ 3.8
23012 Constant	38.6	- 1	- 1	- .6
WGS Blade (Fig. 4-21)	36.1	+ 3	+ 4	+ 2.4
SMOOTHNESS				
Standard Roughness*	38			
Smooth	40.5	- 4	- 5	- 2.9
TWIST				
Optimum*	38			
0°	33	+ 8	- 10	+ 5.9
15° Linear	37.5	+ 0.8	- 1	+ .6
23° Linear	35.4	+ 4	+ 5	+ 3.1
ROOT CUT OUT				
10% Cut Out*	38			
20% Cut Out	36	+ 3	+ 4	+ 2.4
35% Cut Out	31.7	+ 10	+ 13	+ 7.4
TAPER				
3:1*	38			
2:1	37.7	+ .5	+ .6	+ .3
1:1 (Lowest Chord)	37	+ 1.6	+ 2	+ 1.2
START OF 3:1 TAPER				
Mid Span Taper*	38			
Full	38.3	- .5	- .6	- .3

*Baseline Configuration for optimization study.

obtained from two-dimensional tests in the Langley Low Turbulence Pressure Tunnel, and are considered valid for the low Mach number operation of wind turbines. Section characteristics for the above mentioned airfoils are presented in Figures 3-35 through 3-37.

Although L/D ratios for the NACA 4412 and 63_2-615 airfoils are higher than for the 23012 airfoil, the rotor optimization study yielded very little difference in aerodynamic efficiencies for the entire rotor. Consequently, other considerations, such as lower blade aerodynamic pitching moments, better producibility and satisfactory past performance in helicopters and fixed wing aircraft, led to selection of the 230 series airfoil for the WGS application. (230 series airfoils are easier to produce because of the absence of reflex curvature in the under surface, particularly important in filament-wound composite construction.)

After selection of the 230 series airfoil in the optimization phase of this WGS design study, it was retained for the remainder of the program. However, prior to a final design phase, the question of airfoil selection should be reopened, in more depth than could be undertaken in this study, to further optimize aerodynamic efficiency.

During the preliminary design phase of the wind generator system study, blade tuning requirements necessitated increased bending stiffness at the blade root end. Thus, airfoil thickness ratios were increased inboard from 18% at the mid-span, linearly to 40% at the 0.2 radius station. Reference 3-3 was again used as the source of 230 section airfoil data up to a thickness ratio of 24% at 0.4 radius. For thicker sections inboard of this radius, data for the 230 section were obtained from Reference 3-4 tests of 23012, 23021 and 23030 sections in the Langley 7 x 10 atmospheric wind tunnels. The section drag data for a 23012 airfoil from the higher turbulence conditions of this tunnel appear in close agreement with standard roughness data from the low turbulence tunnel tests of Reference 3-3. For the 30% thickness section, therefore, these higher turbulence drag data were considered representative of the roughness condition expected, and were used directly.

In order to avoid reflex surface curvature in the thicker root section, a transition was made to a symmetrical 4-digit NACA 0040 section at the 20% radius. Section characteristics were estimated from extrapolation of the results of Reference 3-5 and 3-6 on 4-digit symmetrical sections varying from 9 to 35% thickness ratio with suitable corrections for the effect of roughness from Reference 3-3. The resulting airfoil section characteristics for the 40% thickness ratio section are presented in Figure 3-35, along with the higher thickness ratios of the 230 sections.

3.4.4.2 Altitude and Temperature

The rotor sees altitude and temperature changes as changes in air density. This effect is most conveniently studied in terms of density altitude.

Figure 3-38 shows density altitude effects on the major parameters of optimized systems; i.e., rotor diameter is permitted to vary. The graph shows that increases in density altitude cause size and cost to increase while the plant factor remains essentially unaffected.

AIRFOIL CHARACTERISTICS

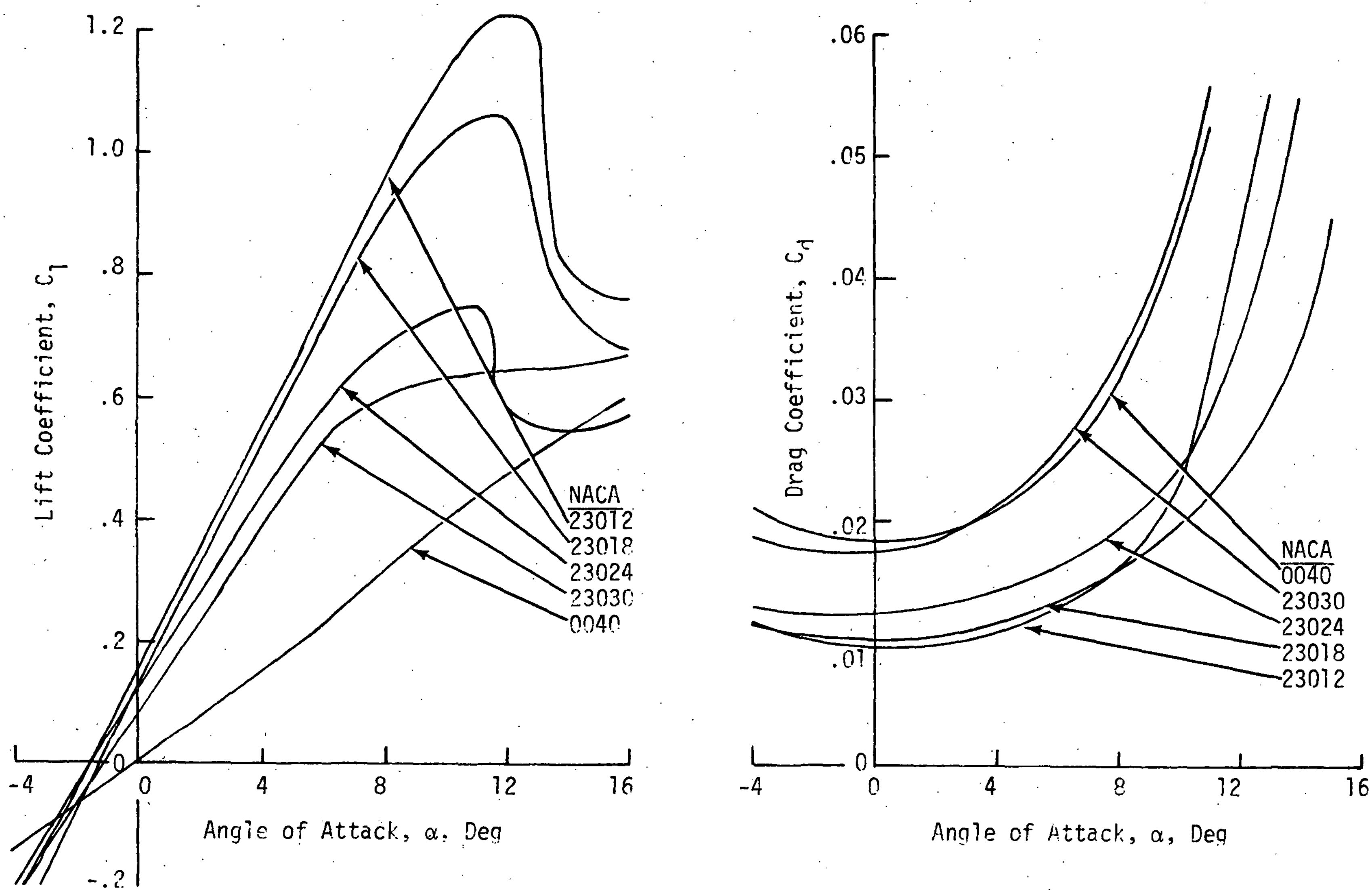


Figure 3-35. 230 Series and 0040 Airfoil Section Lift and Drag Characteristics (Standard Roughness).

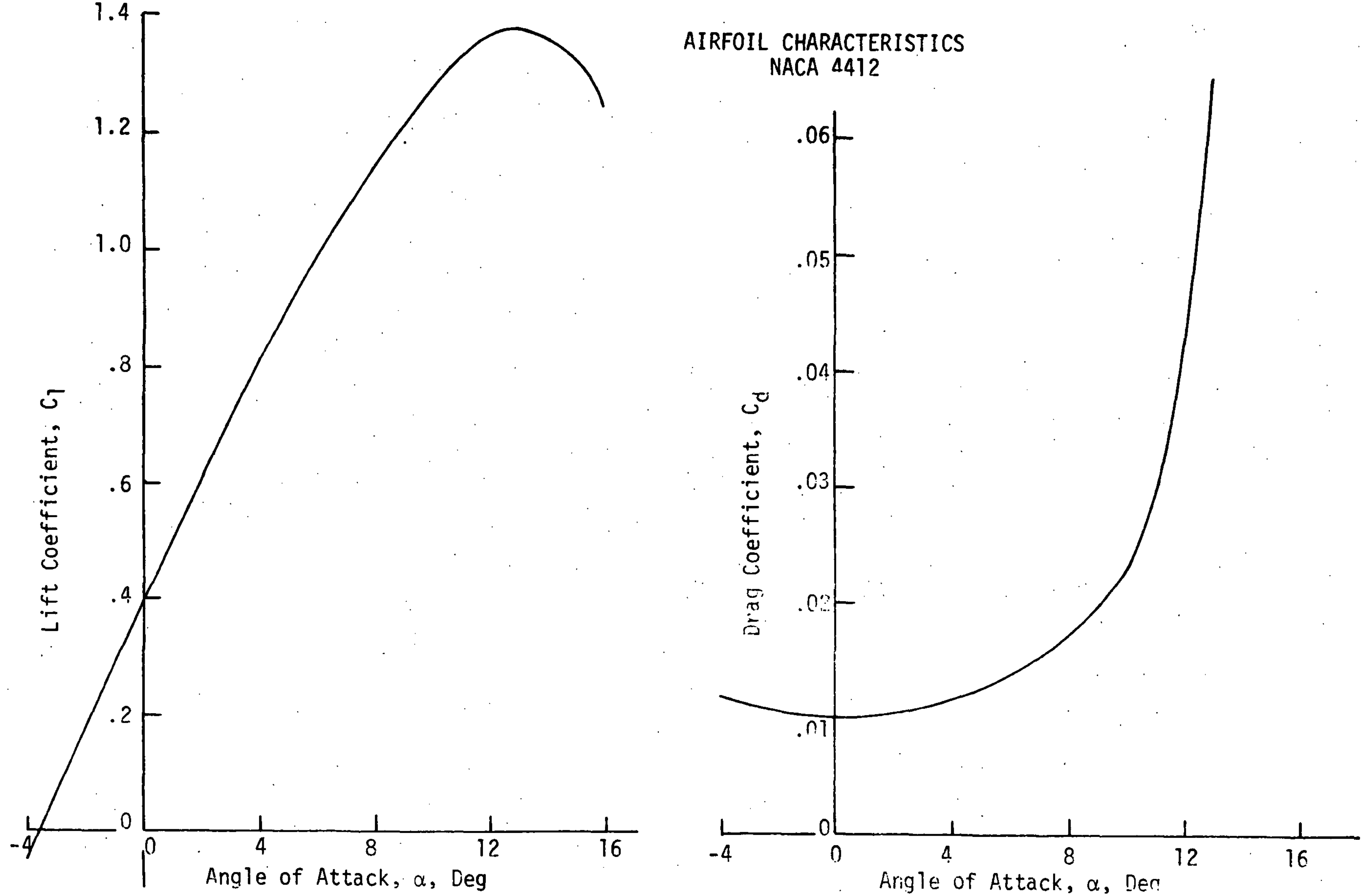


Figure 3-36. NACA 4412 Airfoil Section Lift and Drag Characteristics (Standard Roughness).

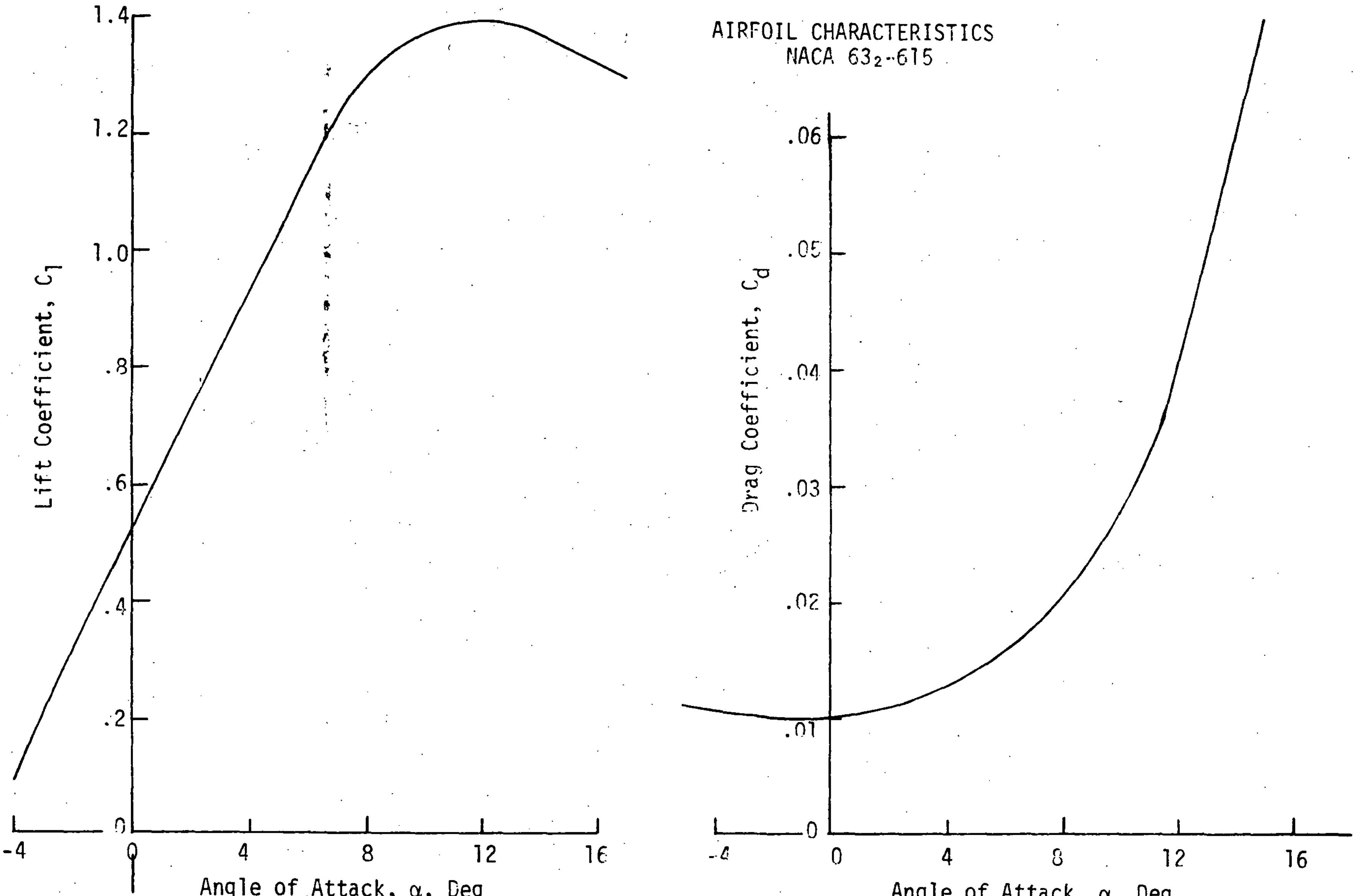


Figure 3-37. NACA 63_2-615 Airfoil Section Lift and Drag Characteristics (Standard Roughness).

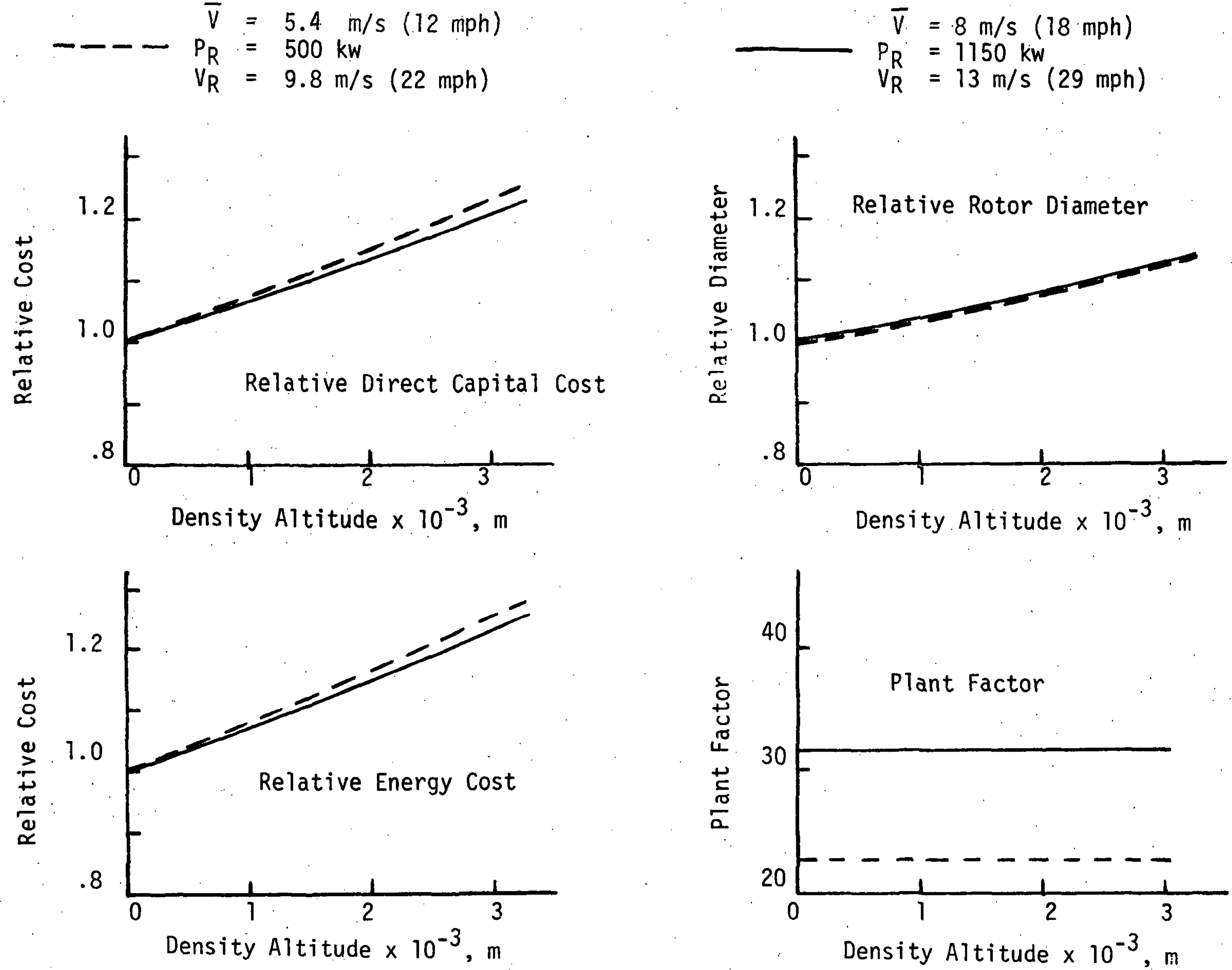


Figure 3-38. Density Altitude Effects on System Parameters.

3.4.4.3 Drive and Electrical Subsystem Components

Figure 3-39 illustrates the sensitivity of costs, size and utilization to changes of gearbox efficiency. The results may be generalized to be indicative of any efficiency shift of any component in the power line. Figure 3-40 shows penalties due to lowering generator speed. There is no improvement predicted above the selected 1800 rpm.

Figure 3-41 shows cable size effect. The curves indicate that the smallest practical cable is the most cost effective.

3.4.4.4 Tower Aspect Ratio

Tower slenderness or aspect ratio effects on cost and utilization parameters are shown in Figure 3-42. Aspect ratio is the ratio of height over base width. Narrow or high aspect ratio towers are designed for tower stiffness. Low aspect ratio towers are designed for blowover conditions. The shown optimum aspect ratio of 4 lies near the intersection of these criteria.

In this particular investigation, the angle between the tower side surface and the plane normal to the shaft (rotor inclination) was held at 10°. Shaft overhang and rotor coning increase the clearance to beyond that provided by the 10°.

3.4.5 Selected Systems for Preliminary Design

The information summarized in previous sections was generated primarily to select the optimum WGS design for two wind speed regime sites: a 5.4 m/s (12 mph) median wind speed site and an 8 m/s (18 mph) median wind speed site. Originally, it was anticipated that energy cost would be the dominating criterion for selection, but during the course of the study other parameters of significance also were recognized. These include capital cost and plant factor. Carpet plots for the two median wind velocities, 5.4 m/s (12 mph) and 8.0 m/s (18 mph), are given in Figures 3-43 and 3-44, respectively. These plots summarize the relationships between rated wind speed and power, rotor diameter, plant factor, energy cost and direct capital cost.

3.4.5.1 Selection Rationale

With multiple decision criteria available from the model results, plus other program and technical considerations not produced by the model, an evaluation of all study results was conducted by Kaman and NASA. The result of this evaluation is presented in Table 3-13, which summarizes the essential features of the optimum (minimum energy cost) WGS designs for both wind speed regimes and the system characteristics selected for the preliminary designs.

As shown, the selected low power system is very close to the nominal optimum design. However, the high power system has been selected to have a substantially higher plant factor than the optimum WGS would have. To achieve this, a larger rotor diameter is required, and a higher capital cost, although energy cost is practically the same as for the optimum system.

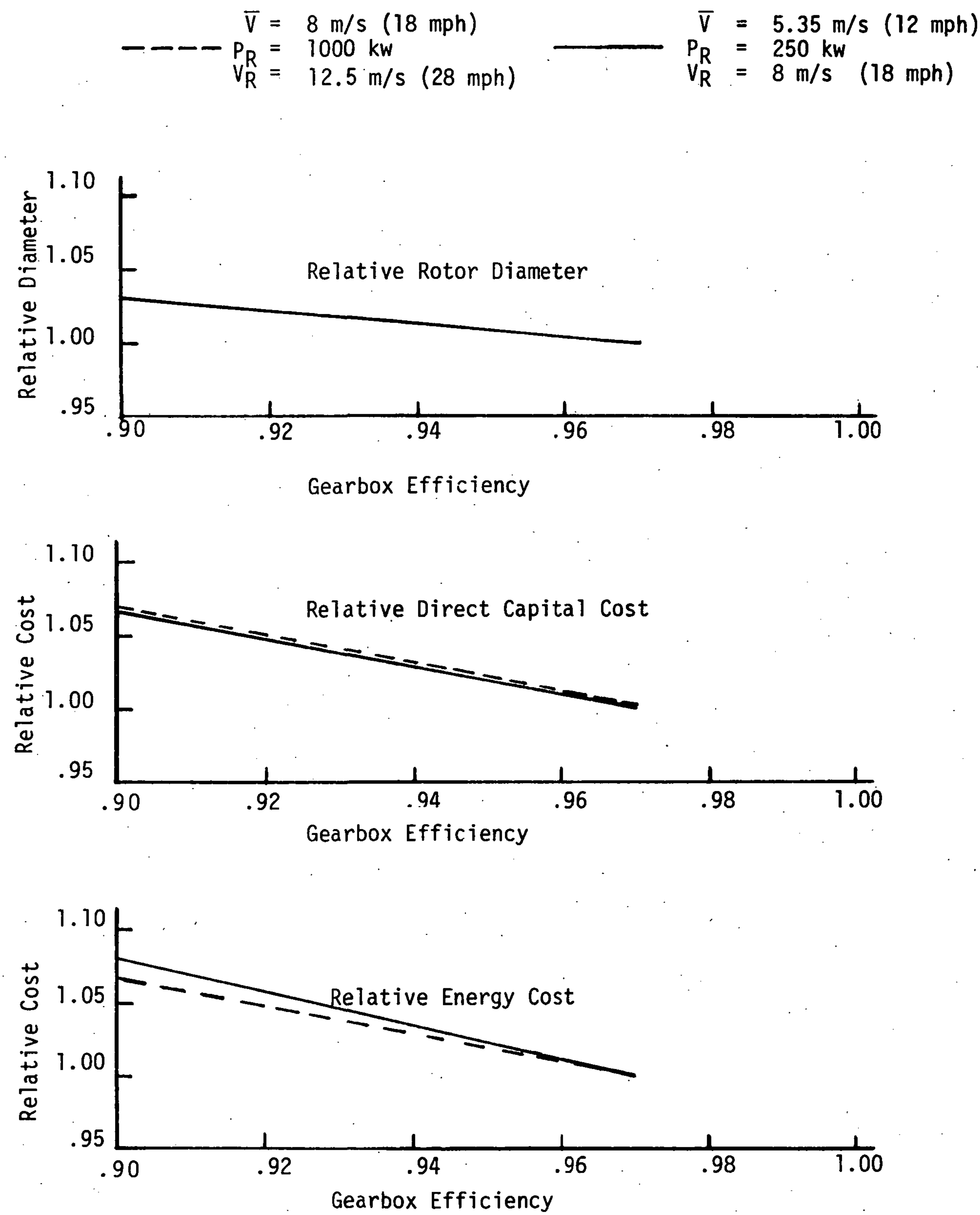


Figure 3-39. Gearbox Efficiency Effect on System Parameters.

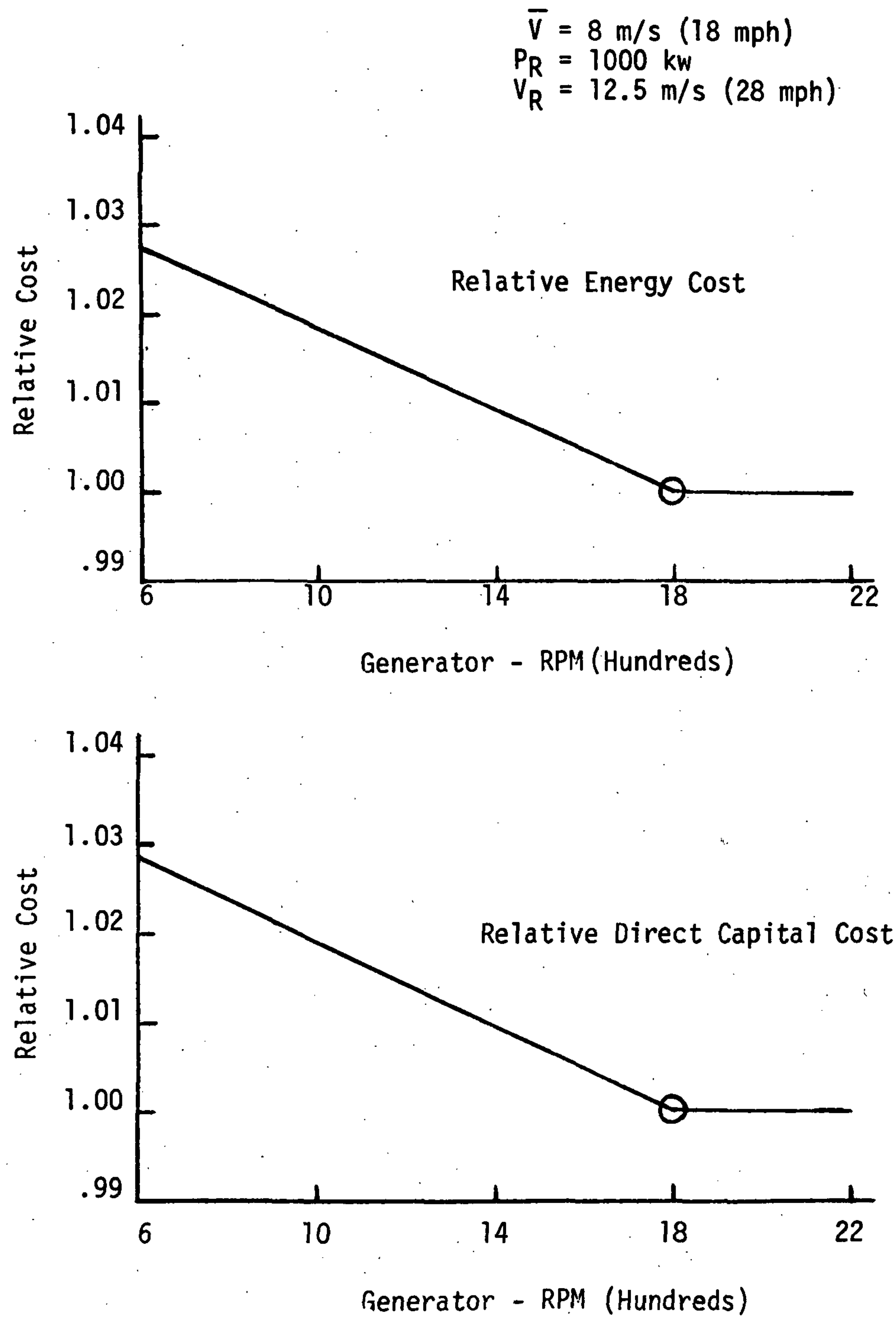


Figure 3-40. Generator RPM Effect on System Parameters

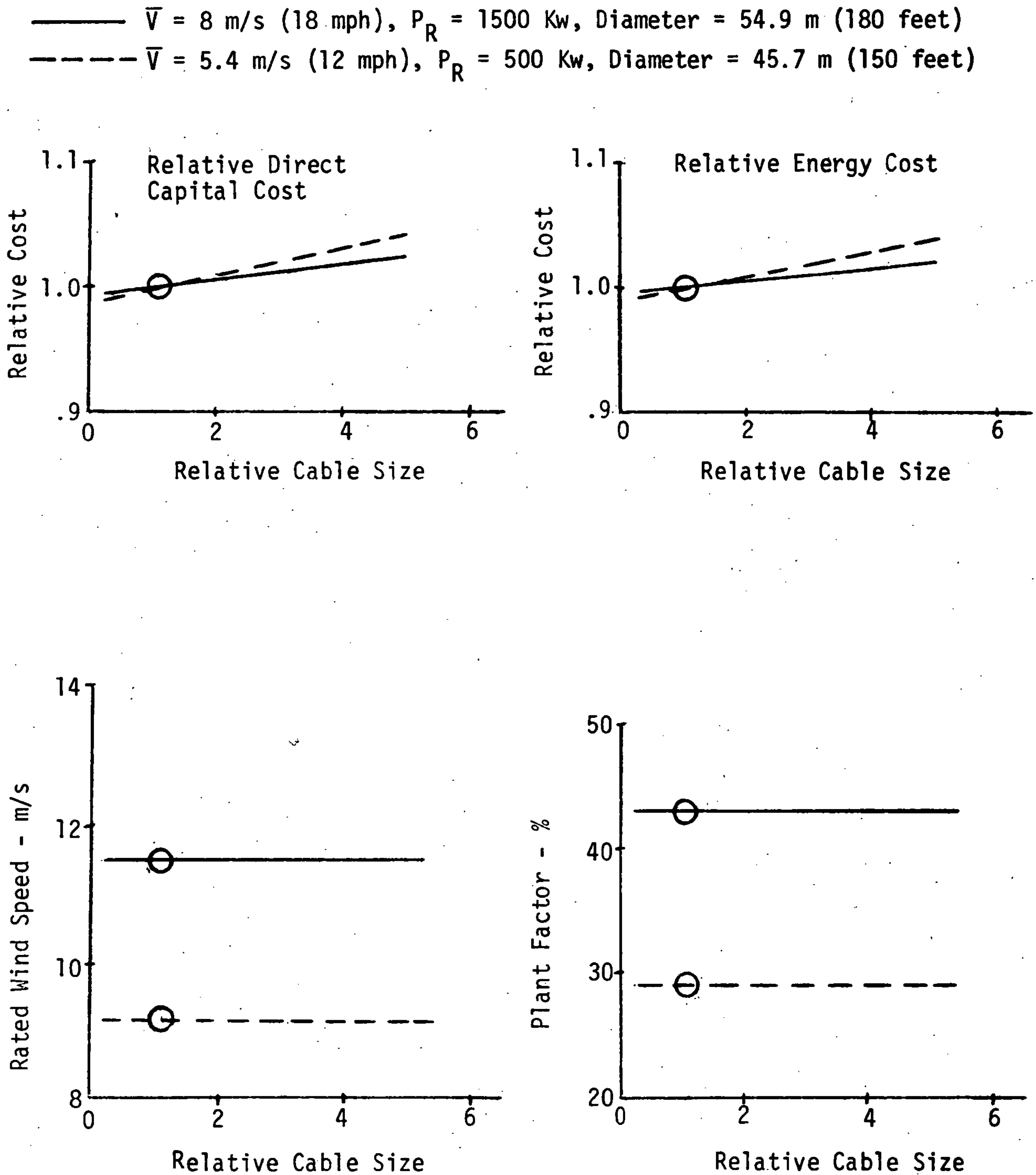


Figure 3-41. Cable Size Effect on System Parameters

$\overline{V}$ = 8 m/s (18 mph), P_R = 1500 Kw, Diameter = 54.9 m (180 ft)

$\overline{V}$ = 5.4 m/s (12 mph), P_R = 500 Kw, Diameter = 45.7 m (150 ft)

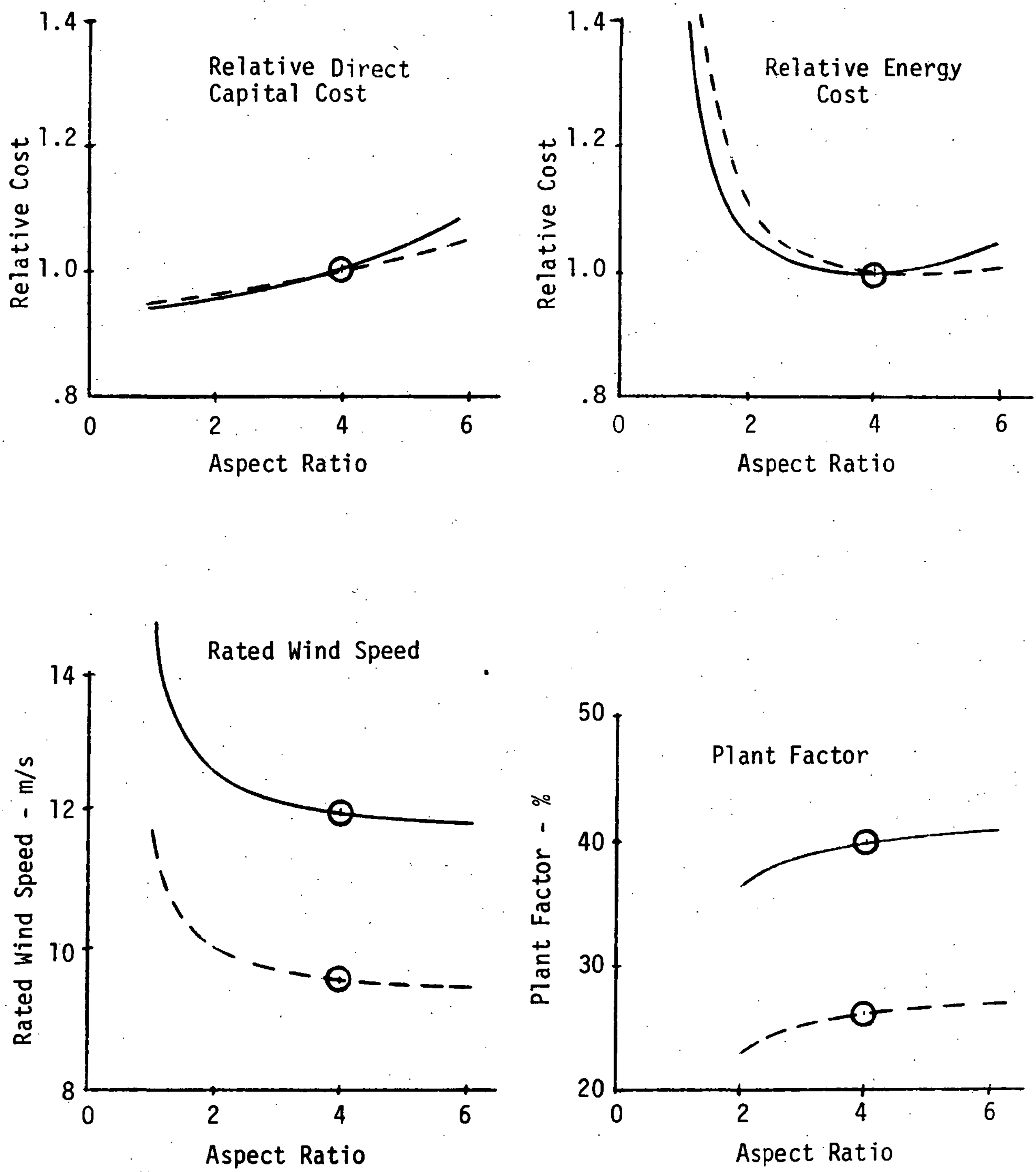


Figure 3-42. Tower Aspect Ratio Effects on System Parameters

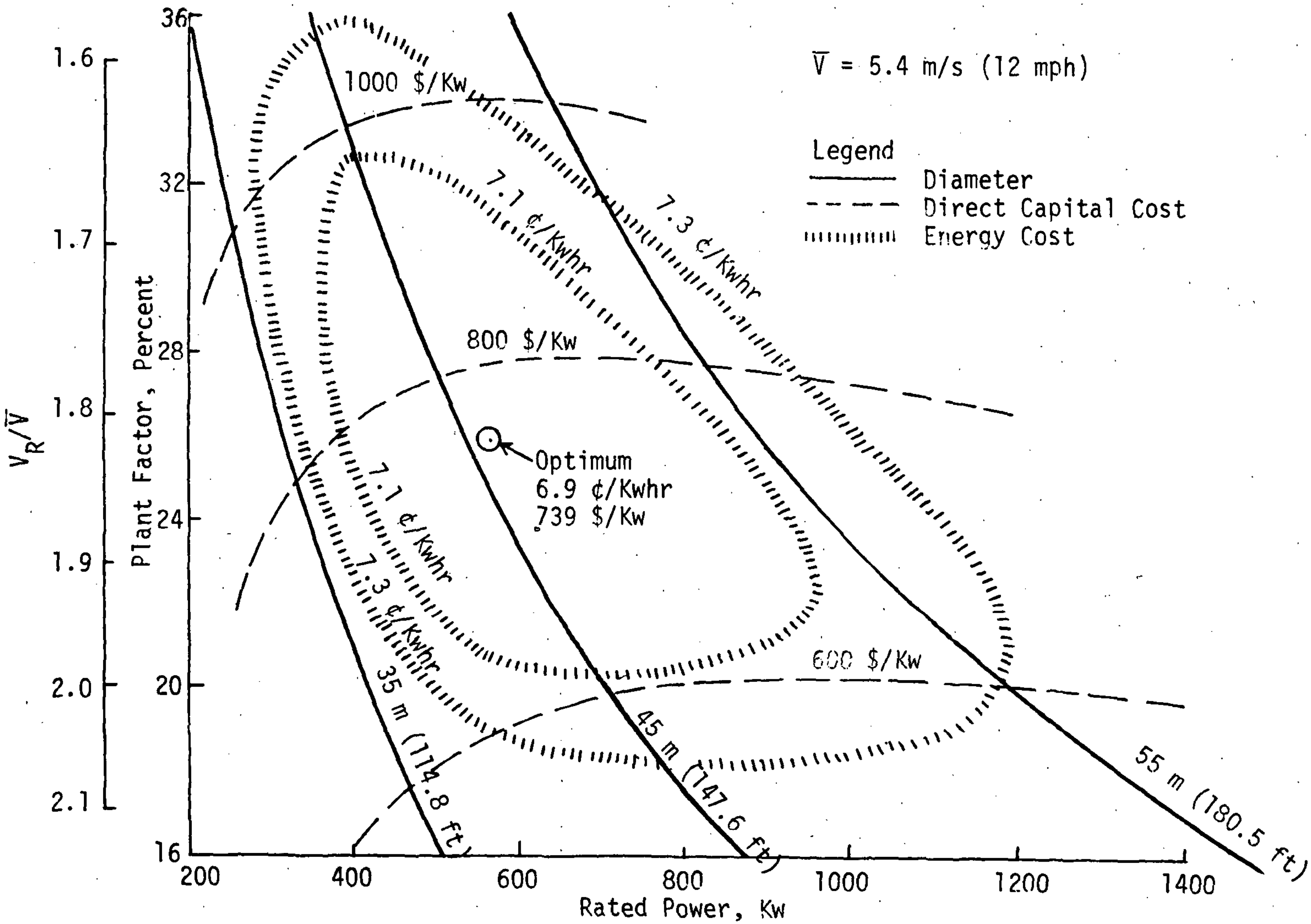


Figure 3-43 . Relationship of Evaluation Parameters Near Optimum Condition

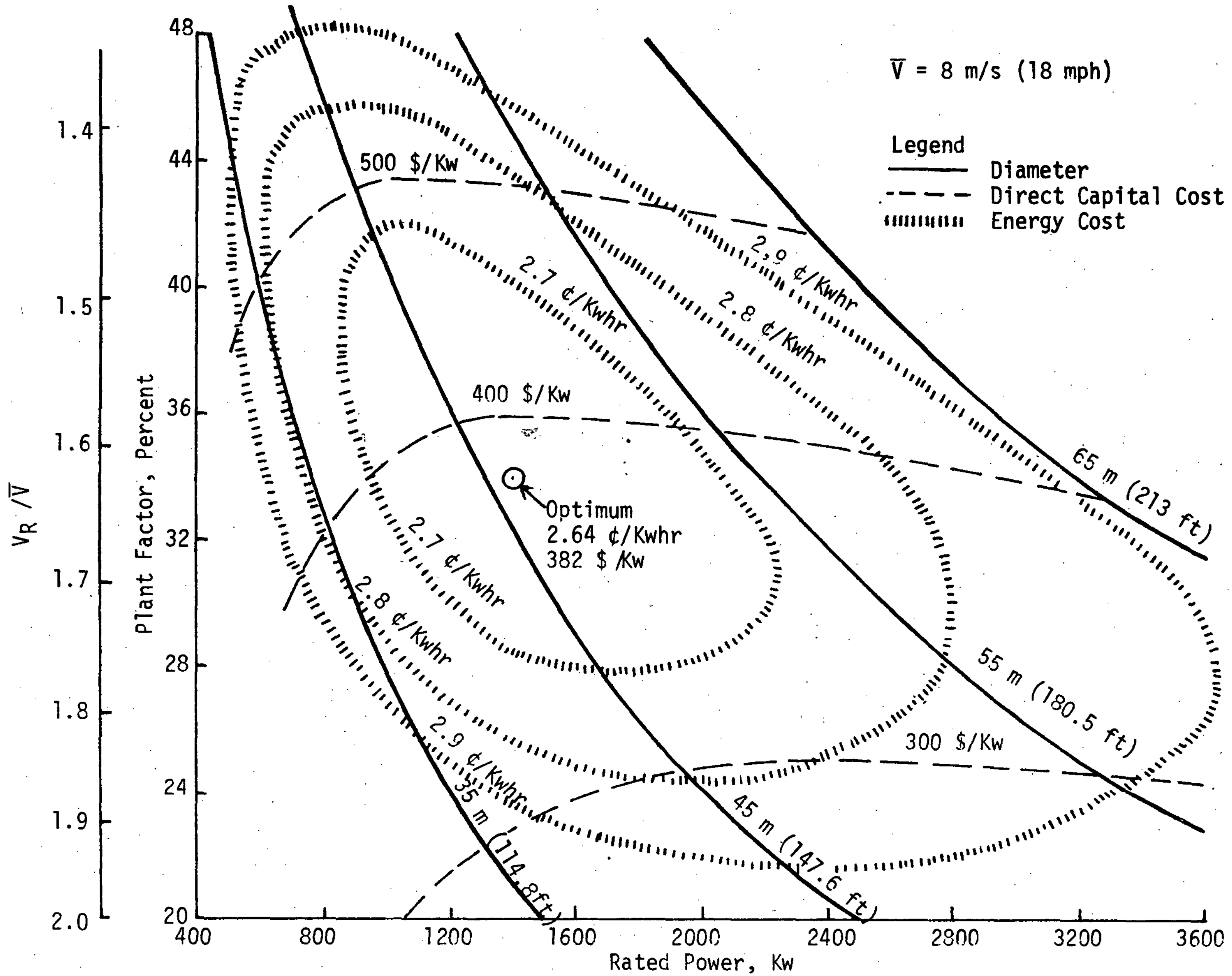


Figure 3-44. Relationship of Evaluation Parameters Near Optimum Condition

TABLE 3-13. MINIMUM ENERGY COST AND SELECTED SYSTEMS

	LOW POWER SYSTEM		HIGH POWER SYSTEM	
	MINIMUM ENERGY COST	SELECTED	MINIMUM ENERGY COST	SELECTED
Median Wind Speed, m/s (MPH)	5.4 (12)	5.4 (12)	8 (18)	8 (18)
Rated Power, KW	560	500	1,400	1,500
Rated Wind Speed, m/s (MPH)	9.7 (22)	9.3 (21)	13 (29)	11.5 (26)
Rotor Diameter, m (Ft)	44.9 (147)	45.7 (150)	45.4 (149)	54.9 (180)
Rotor Solidity	.03	.03	.03	.03
Rotor Speed, RPM	34	32	48	34
Energy Cost, ¢/KW HR	6.9	7.0	2.6	2.8
Capital Cost, $/KW	739	846	382	499
Unit Cost, $	414,000	423,100	534,500	749,000
Plant Factor, %	25	29	34	43

Selecting the high power system at a point which does not appear to have minimum energy cost is, as discussed before, easily accomplished for only a slight energy cost penalty. However, selecting a system with a higher capital cost in return for a higher plant factor was not the only other quantitative element in the selection process.

One other factor was the recognition that both the low power and high power system rotors optimized at 45.7 m (150 ft) in diameter, directly attributed to the rotor model used. The earlier rotor model, discussed previously, had shown systems optimizing at larger diameter rotors. Hence, it would be prudent to recognize this factor and take it into consideration when making a final selection.

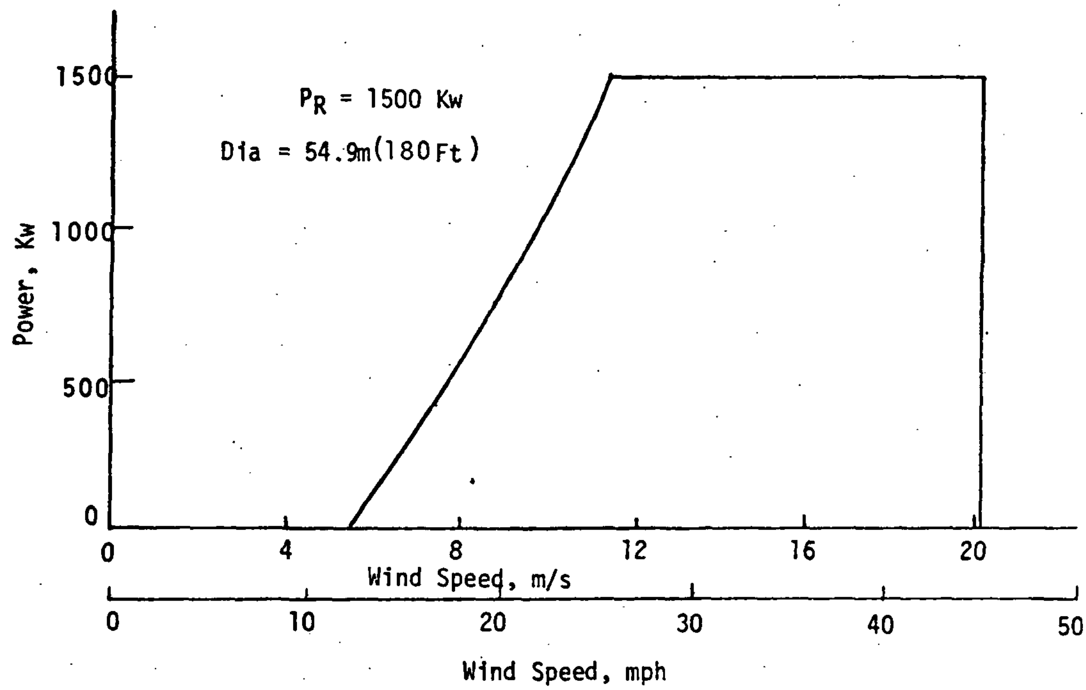
Another factor was the adaptability of the WGS design to non-design condition sites. If the low power and high power systems are designed with different diameters, the two systems provide a better choice of designs for non-design sites than if both machines have the same diameter.

Since energy cost for the high power system was not an important factor in the 46 m - 55 m (150 ft - 180 ft) diameter range, capital cost, plant factor, rotor model sensitivity and site adaptability were the quantitative parameters which were added to other program considerations in selecting the high power system characteristics.

3.4.5.2 Model Predicted WGS Characteristics

With the selection process completed, the preliminary design task proceeded for the selected concept with specific values for the high power and low power systems. The model was used to calculate overall system performance and system and component dimensions, weights, costs and operating parameters (speeds, loads, etc.) for the selected designs.

The model calculated data for the selected systems are presented in this section. Figure 3-45 presents the power profiles for the systems. Figure 3-46 breaks down the profiles to show the relative percent of operating time the systems spent in each operating mode. Table 3-14 presents the predicted dimensions of the systems. Table 3-15 gives the predicted weights and Table 3-16, the corresponding costs for the selected systems. Table 3-17 presents a power analysis of selected systems at rated conditions. The overall power loss of the power generation system is the combined losses of the transmission, generator, cable and transformer. Figure 3-47 shows the overall loss in terms of power generation system efficiency, which is defined as the ratio of transformer (output) power to rotor (input) power. The efficiency is shown for the two selected WGSs and is plotted as a function of rated power fraction. Each component of the power generation system, except the cable, has a loss component that is constant and a component that varies as a function of power transmitted. The shape of the overall power generation system efficiency reflects these characteristics. On the bottom of Figure 3-47, wind velocity fraction, V/V_R, is shown for reference.



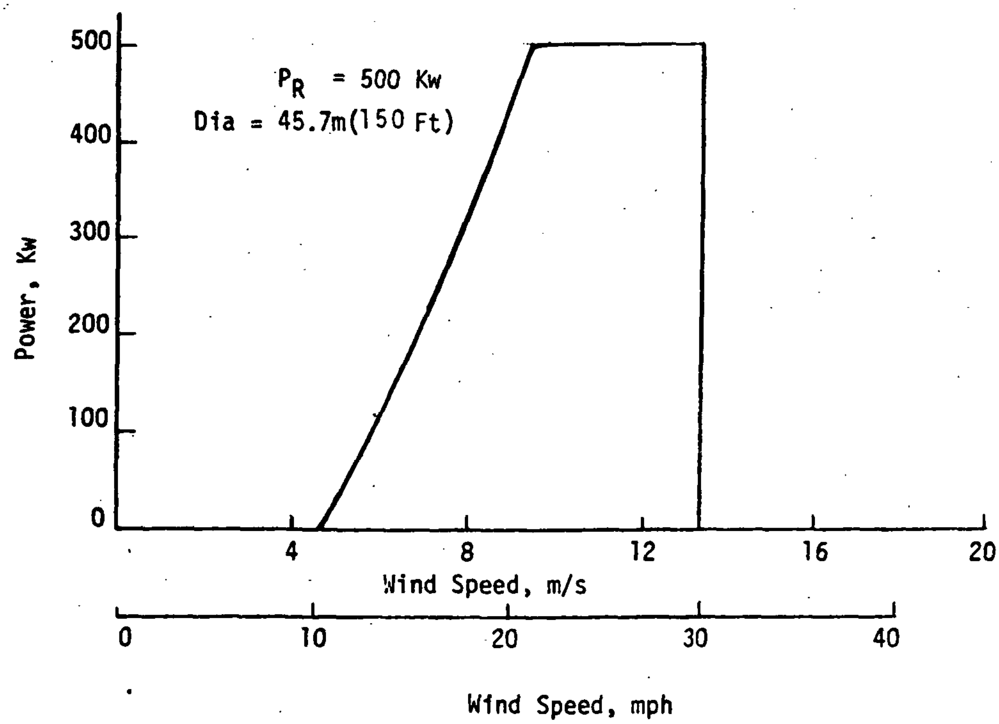
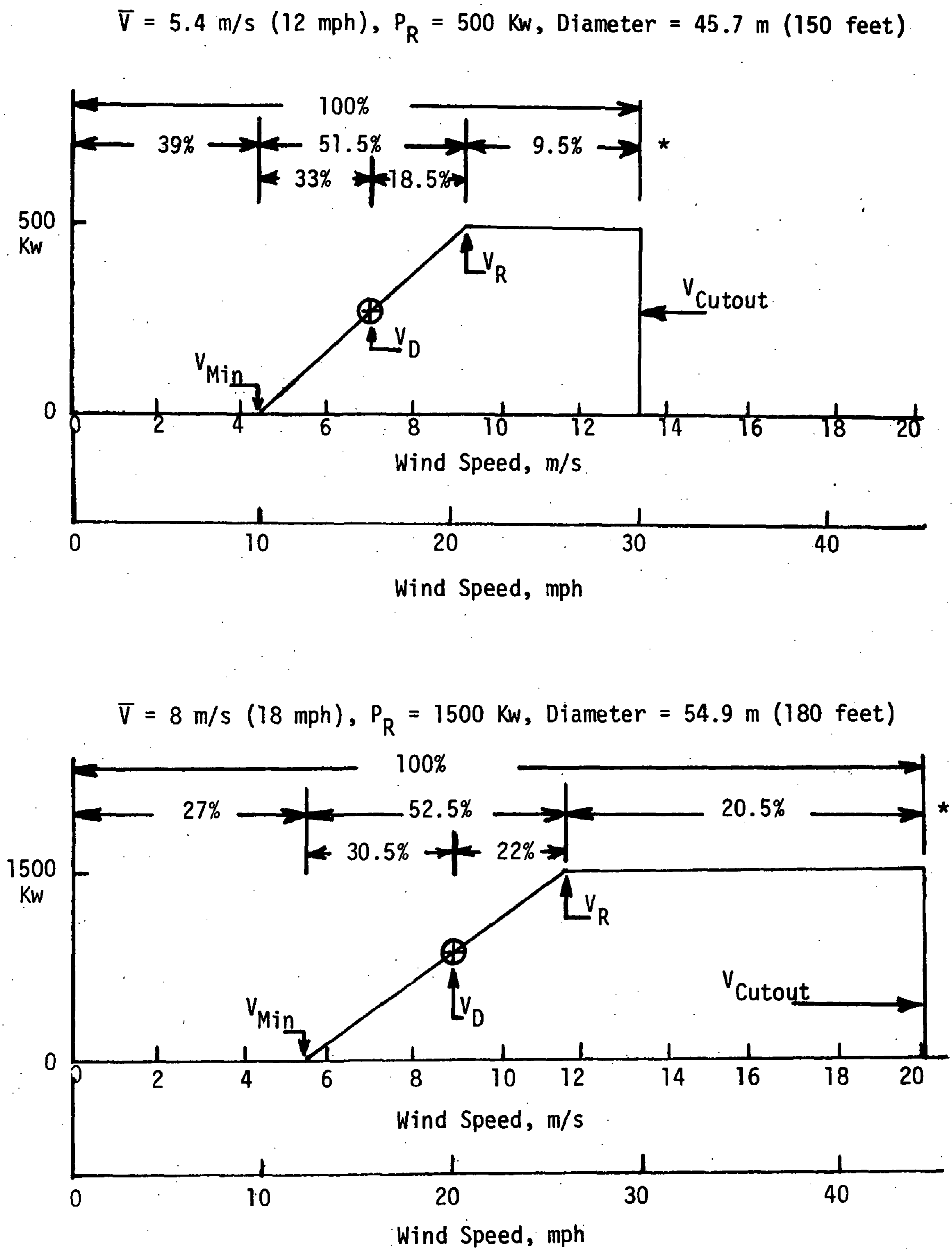


Figure 3-45. Power Output Profiles of Selected Systems



* Percentage of winds above V cutout is statistically insignificant

Figure 3-46. WGS Utilization Time Of Selected Systems.

TABLE 3-14. SELECTED GENERAL CHARACTERISTICS PREDICTED BY MODEL				
	WGS RATED POWER			
	500 kW		1500 kW	
CHARACTERISTICS	METRIC	ENGLISH	METRIC	ENGLISH
ROTOR:				
Diameter	45.7 m	150 ft	54.9 m	180 ft
Solidity	.03	.03	.03	.03
Number of Blades	2	2	2	2
Root Cutout Fraction	.1	.1	.1	.1
Tip Speed (V_{TIP})	77.1 m/sec	253 fps	99.1 m/sec	325 fps
Rotational Speed	32.2 rpm	32.2 rpm	34.5 rpm	34.5 rpm
Torque @ Rated Power	166,059 N-m	122,485 lb-ft	456,280 N-m	336,550 lb-ft
Efficiency @ Rated Wind Speed	.342	.342	.344	.344
GENERATOR SPEED	1,800 rpm	1,800 rpm	1,800 rpm	1,800 rpm
GEARBOX GEAR RATIO	55.88	55.88	52.21	52.21
WIND:				
Median Wind Speed, $\overline{V}$	5.36 m/sec	12 mph	8.05 m/sec	18 mph
Rated Wind Speed, V_R	9.30 m/sec	20.8 mph	11.6 m/sec	25.9 mph
Minimum Operating Wind Speed, V_{MIN}	4.47 m/sec	10 mph	5.36 m/sec	12 mph
Cutout Wind Speed, V_{CUTOUT}	13.4 m/sec	30 mph	20.1 m/sec	45 mph
Density ratio	1.0	1.0	1.0	1.0
SELECTED RATIOS:				
Tip Speed (Advance) Ratio at Design Point, V_{TIP}/V_{EFF}	8.63	8.65	8.63	8.65
Effective Wind to Reference Wind, V_{EFF}/V_{REF}	1.27	1.27	1.29	1.29
$V_R/\overline{V}$	1.73	1.73	1.44	1.44
Design Wind Speed to Rated Wind Speed, V_{DES}/V_R	.76	.76	.77	.77
TOWER:				
Height	38.1 m	125 ft	42.7 m	140 ft
Aspect Ratio	4	4	4	4
Rotor Clearance	15.2 m	50 ft	15.2 m	50 ft
YEARLY OPERATING TIME	5573 hrs	5573 hrs	5573 hrs	5573 hrs
PLANT FACTOR	.293	.293	.433	.433
YEARLY ENERGY OUTPUT	1.28×10^6 kWh	1.28×10^6 kWh	5.68×10^6 kWh	5.68×10^6 kWh

TABLE 3-15. MODEL PREDICTED WEIGHTS FOR SELECTED SYSTEMS

	WGS RATED POWER			
	500 kW		1500 kW	
	kg	lbs	kg	lbs
WGS Above Foundation	51,307	113,113	106,290	234,329
ROTOR SUBSYSTEM	6,440	14,208	12,593	27,763
Blades	2,011	4,434	4,031	8,887
Hub	4,430	9,774	8,562	18,876
Housing	1,782	3,929	3,442	7,588
Hub-mounted controls	337	742	651	1,435
Pitch mechanism	2,314	5,102	4,469	9,853
DRIVE SUBSYSTEM	12,518	27,597	28,160	62,083
Shaft	3,892	8,581	7,978	17,588
Gearbox	4,047	8,923	10,361	22,843
Low-speed coupling	626	1,381	1,645	3,626
High-speed coupling	161	30	37	81
Brake	68	150	68	150
Clutch	11	24	33	72
Yaw mechanism	3,352	7,390	7,419	16,355
Ring gear	507	1,118	621	1,368
ELECTRICAL SUBSYSTEM ON TOP	2,986	6,582	6,736	14,851
Generator	2,661	5,867	6,314	13,920
Electrical equipment on top	98	215	168	371
Cable	227	500	254	560
CONTROLS SUBSYSTEM	523	1,153	679	1,496
STRUCTURE SUBSYSTEM	28,836	63,573	58,122	128,136
Enclosure	1,596	3,518	2,368	5,221
Turntable	6,864	15,132	14,055	30,986
Miscellaneous	170	375	191	420
Ladders, platform	3,198	7,050	3,582	7,896
Structural steel	17,009	37,498	37,926	83,613
Pintle support	1,215	2,679	2,031	4,477
Chords	9,963	21,956	21,776	48,007
Diagonals	1,505	3,320	5,518	12,165
Horizontal braces	684	1,509	2,510	5,533
Gussets	1,215	2,678	2,031	4,477
Attachments	1,215	2,678	2,031	4,477
Base support	1,215	2,678	2,031	4,477

TABLE 3-16. MODEL PREDICTED COSTS FOR SELECTED SUBSYSTEMS

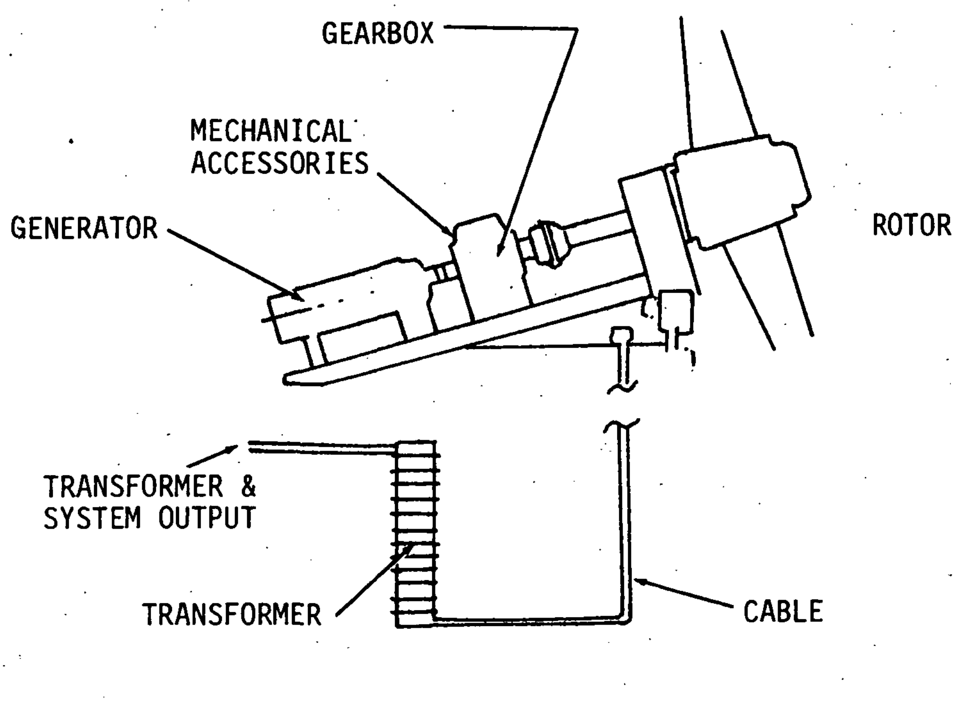
SUBSYSTEM	500 kW	1500 kW
WGS DIRECT COST	423,106	748,948
ROTOR	142,142	275,994
Blades	112,579	218,897
Hub	29,563	57,097
Housing	15,323	29,594
Hub-mounted controls	1,386	2,869
Pitch change mechanism	12,755	24,633
DRIVE	83,788	187,139
Shaft	28,533	58,481
Gearbox	24,092	61,677
Low-speed coupling	6,298	16,536
High-speed coupling	135	368
Brake	1,500	1,500
Clutch	247	728
Yaw mechanism	19,953	44,158
Ring gear	3,020	3,694
ELECTRICAL	54,413	84,988
Generator	14,913	37,944
Cable (total)	7,231	7,565
Transformer	5,389	12,149
Equipment for power generation	21,210	21,210
Electric utilities	5,670	6,120
ENCLOSURE	7,036	10,443
TURNTABLE	9,987	20,451
CONTROLS	11,350	11,350
TOWER (Including installation)	72,962	105,729
Steel	21,374	47,659
Ladders, gratings	13,250	14,840
Foundation	38,339	43,230
INSTALLATION (Excluding tower)	18,877	22,943
SITE PREPARATION	22,560	29,909
Land acquisition	6,457	9,298
Land clearing	6,973	10,041
Fence and shed	8,800	10,240
Pad	330	330

TABLE 3-16. MODEL PREDICTED COSTS FOR SELECTED SUBSYSTEMS (continued)

SUBSYSTEM	500 kW	1500 kW
OPERATION AND MAINTENANCE COSTS	25,781	48,302
OPERATION	6,347	11,234
MAINTENANCE	19,435	37,068
Rotor	14,214	27,598
Power systems	4,430	8,247
Tower	565	922
Site	226	299
CARRYING CHARGES	63,466	112,342
TOTAL YEARLY COSTS	89,247	160,644
DIRECT CAPITAL COST ($/kW)	846	499
YEARLY ENERGY OUTPUT (kWh)	1.281×10^6	5.684×10^6
ENERGY COST ($/kWh)	.0697	.0283
BREAKDOWN OF ENERGY GENERATION COST (¢/kWh) (Reference Section 9, paragraph 9.3)		
Rotor Maintenance	1.109	.485
Power Train Maintenance	.346	.145
Tower Maintenance	.044	.016
Site Maintenance	.018	.005
TOTAL MAINTENANCE	1.512	.651
Operations	.495	.198
TOTAL OPERATIONS AND MAINTENANCE	2.012	.849
Depreciation	1.122	.448
Debt Service	1.108	.442
Return on Equity (Stock Dividends)	1.416	.565
Taxes	1.307	.522
TOTAL CARRYING CHARGES	4.953	1.977
TOTAL ENERGY COST (c/kWh)	6.965	2.826

TABLE 3-17. MODEL PREDICTED POWER ANALYSIS AT RATED CONDITIONS

	500 kW	1500 kW
EFFICIENCY:		
Rotor Aerodynamics	.342	.344
Gearbox	.970	.970
Generator	.942	.954
Cable	.998	.999
Transformer	.982	.986
POWER OUTPUT (kW):		
Rotor	560	1,648
Gearbox	541	1,595
Generator	510	1,523
Cable	509	1,521
Transformer	500	1,500
POWER LOSS (kW):		
Hydraulic and Oil Pumps	2.34	3.48
Generator	31.28	72.86
Cable	1.07	1.12
Transformer	9.01	21.38



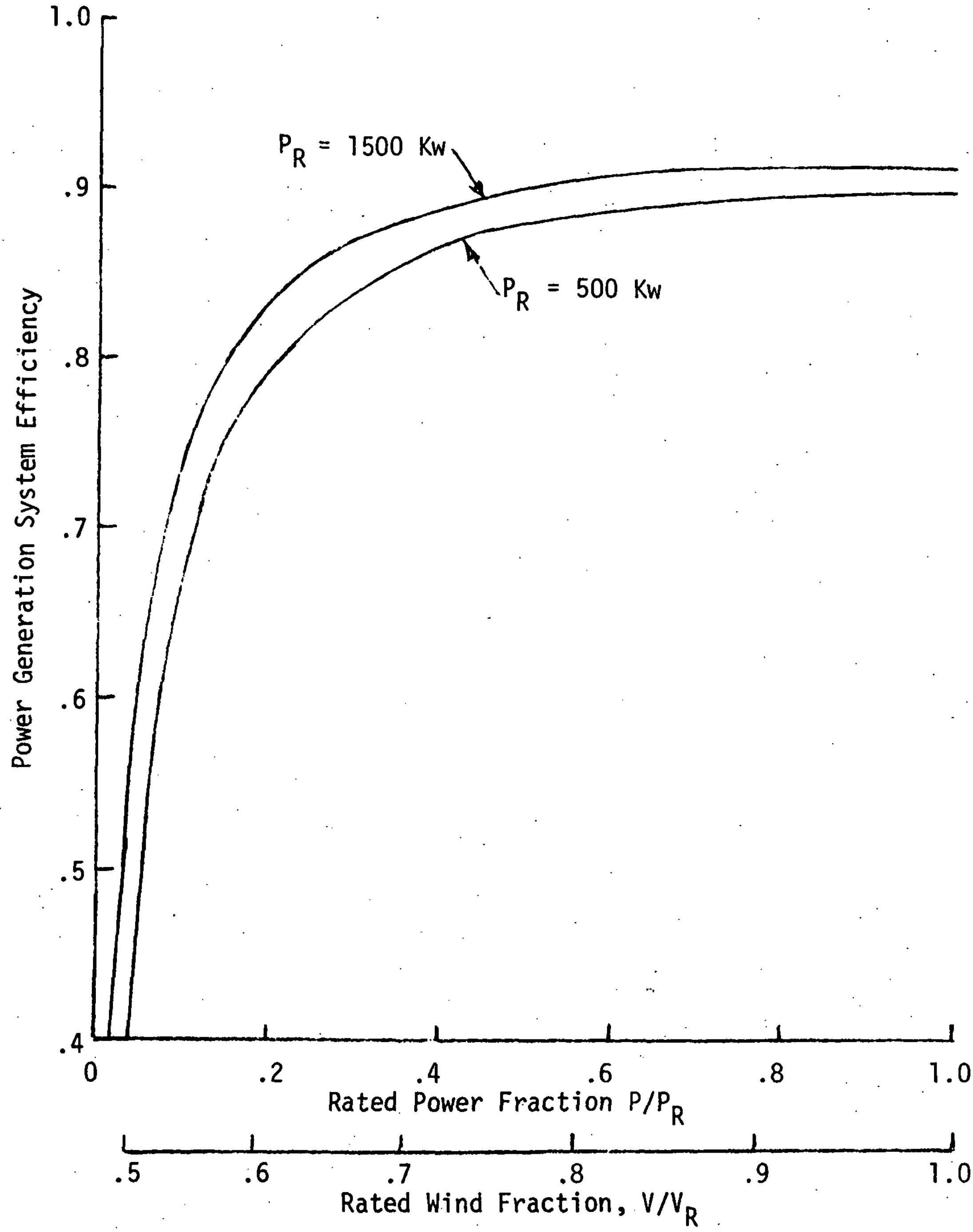


Figure 3-47. Power Generation System Efficiency

3.5 Site Adaptability Studies

The WGS designs described in the previous section were selected for the characteristics of two specific sites; a site with a median wind speed of 5.4 m/s (12 mph), at sea level, with a terrain roughness yielding a wind shear exponent of 0.17 (see paragraph 3.3.3); and a site with an 8 m/s (18 mph) median wind speed, also at sea level and with the same terrain roughness. The question then arises as to how adaptable these designs are for operation at other than their design sites.

Aside from technical considerations, which have a small effect on the designs operating at different sites, the principal issue is the cost of the energy produced by the WGS at a site other than that for which the machine was designed. The questions are: (1) how does the energy cost of a system designed for a specific site change when the machine is operated at a differenct site; (2) how wide a span in site characteristics should a given machine operate over; and (3) given two separate machine designs, how much overlap do these machines provide at a site for which neither was designed?

Several studies were conducted using the parametric model to examine these questions from different viewpoints. Although these studies were not all conducted for the specific selected designs described in the previous section, for reasons which are detailed below, it is believed that the results are generally applicable for any size WGS, and that the conclusions drawn can be applied to the specific machines selected for preliminary design.

3.5.1 Fixed vs Optimized Systems

One way of examining site effects on WGS characteristics is to compare the energy cost of a fixed machine at a variety of sites with that of WGS units designed for the specific site characteristics. This comparison will yield an estimate of the relative benefit of developing an optimized machine for a given site versus utilizing a machine designed for another site, and indicate whether the cost of the new development is worth the energy cost advantage.

In this type of analysis, all optimized systems must be utilized, so as not to distort trends. For this reason, energy cost was used as the determining optimizing variable and resulted in the use of the optimum high power system (designed for the 8 m/s site), with a rating of 1400 KW and a rotor diameter of 45.4 m (149 ft) (see paragraph 3.4.2). The selected low power system (designed for the 5.4 m/s site), rated at 500 KW with a 45.7 (150 ft) diameter rotor, was utilized for the study, since it was very close to the optimum system shown in paragraph 3.4.2.

3.5.1.1 Median Wind Speed Site Effects

Median wind speed is, as discussed before, the most significant WGS parameter, having the greatest impact on energy generated and hence, energy cost. For the high power (1400 KW) system, the change in major system parameters with site median wind speeds are shown in Figure 3-48. For comparison, the corresponding values of parameters of systems optimized for the site are also shown.

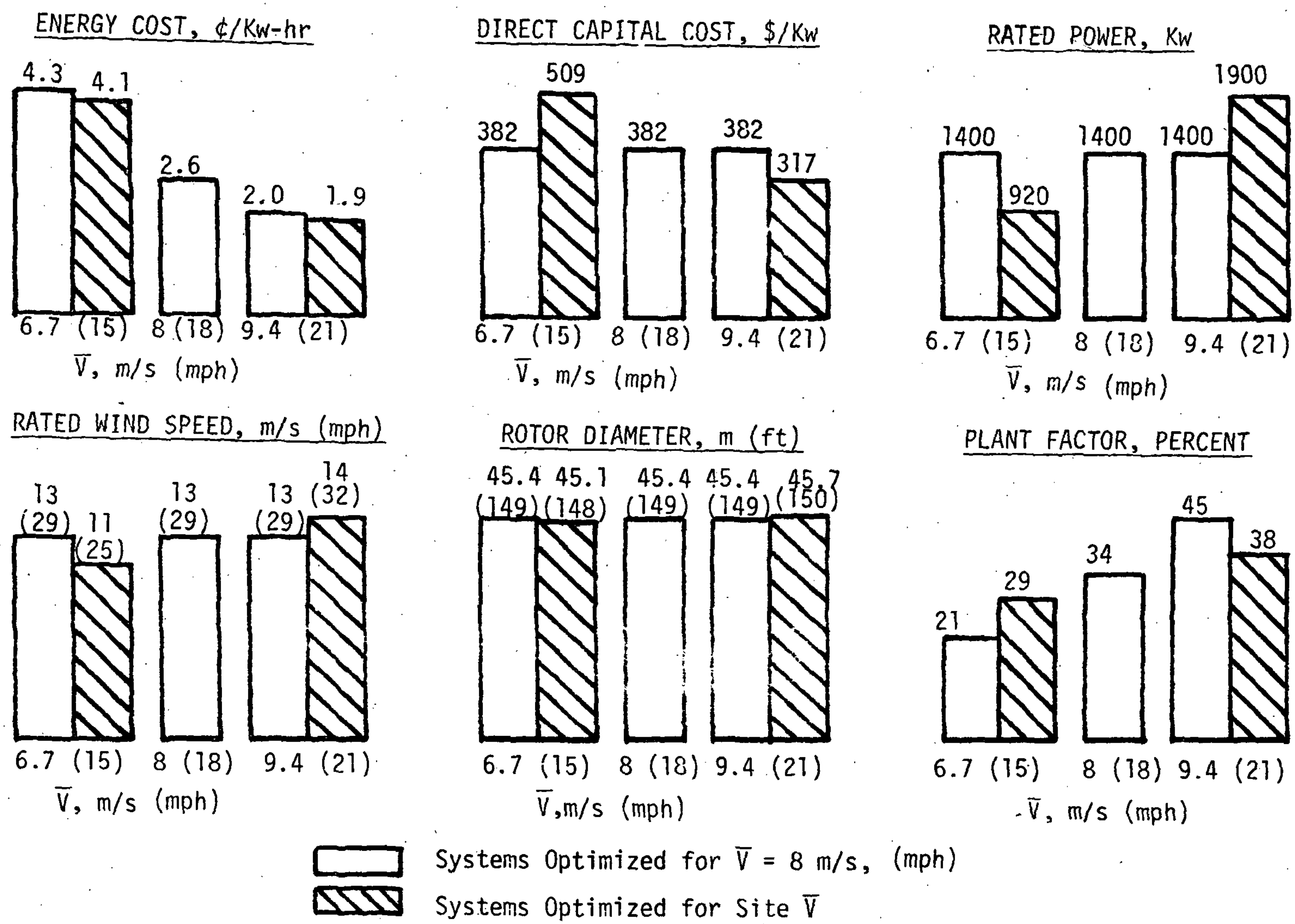


Figure 3-48. Site Median Wind Speed Effect on Fixed and Optimized Systems.

The data in Figure 3-48 show that when the system designed for the 8 m/s site is moved to a 6.7 m/s (15 mph) site, its energy cost increases from 2.6¢/KW-hr to 4.3¢/KW-hr and its plant factor declines from 34% to 21%. Since the machine is a fixed design, capital cost, rated power, rated wind speed and rotor diameter do not change. When the same machine is moved to a 9.4 m/s site (21 mph), its energy cost declines to 2¢/KW-hr, and its plant factor increases to 45%. The optimized systems (optimized for minimum energy cost) do have significantly different capital cost, rated power, rated wind speed and plant factor than the fixed machine design, but their energy cost remains virtually the same as the fixed machine values. This result is consistent with the conclusions reached in the optimization studies discussed in paragraph 3.4; large changes in system design parameters can be made without significant effects on energy cost. In fact, moving the fixed machine to the higher wind site results in a significant gain in plant factor which, for some applications, might outweigh the slight increase in energy cost.

The energy cost variation with site median wind speeds from 4 to 11 m/s (8 to 24 mph) for both optimized high power and low power systems is shown in Figure 3-49. These data illustrate that the two systems cover the entire range of median wind speed sites well, with their energy costs close to the systems which are optimized for each site. The low power design (500 KW) would be used for median wind speed sites less than 6.3 m/s (14 mph) and the high power system would be utilized for median wind speed sites greater than 6.3 m/s. As noted above, the advantage of utilizing a fixed machine vs an optimized machine often lies in the plant factor. Figure 3-50 illustrates that as site median wind speed increases, the trend to increase plant factor for optimized systems is lower than that for a fixed machine.

The data presented in Figures 3-48 through 3-50 indicate that the two WGS designs optimized for the 5.4 and 8 m/s median wind speed sites collectively have acceptable energy costs at median wind speed sites varying from 4 to 9 m/s, when compared to WGS units optimized for each specific site. It is believed that the same conclusion can be drawn concerning the designs selected for preliminary design, even though the selected high power machine characteristics are biased toward higher plant factor than the optimum machine. Thus, it can be concluded that these two machines should provide adequate coverage for WGS requirements at a wide variety of sites, spanning most of the attractive locations for wind generator systems available in the United States.

3.5.1.2 Site Terrain Effects

Terrain characteristics affect the wind shear exponent or vertical variation of wind speed at a given site. This effect and how it is included in the parametric model is discussed in paragraph 3.3.3. If the site terrain is rougher (heavily wooded, rough, rocky ground, etc.) than the reference site terrain (.17 wind shear exponent) then for the same median wind speed at the reference 9.1 m (30 ft) height, a higher average velocity will be seen by the rotor and hence, more energy will be generated by the machine.

To examine this effect, the optimized (as described in the previous section) 1400 KW high power WGS was analyzed to evaluate the changes in its major system

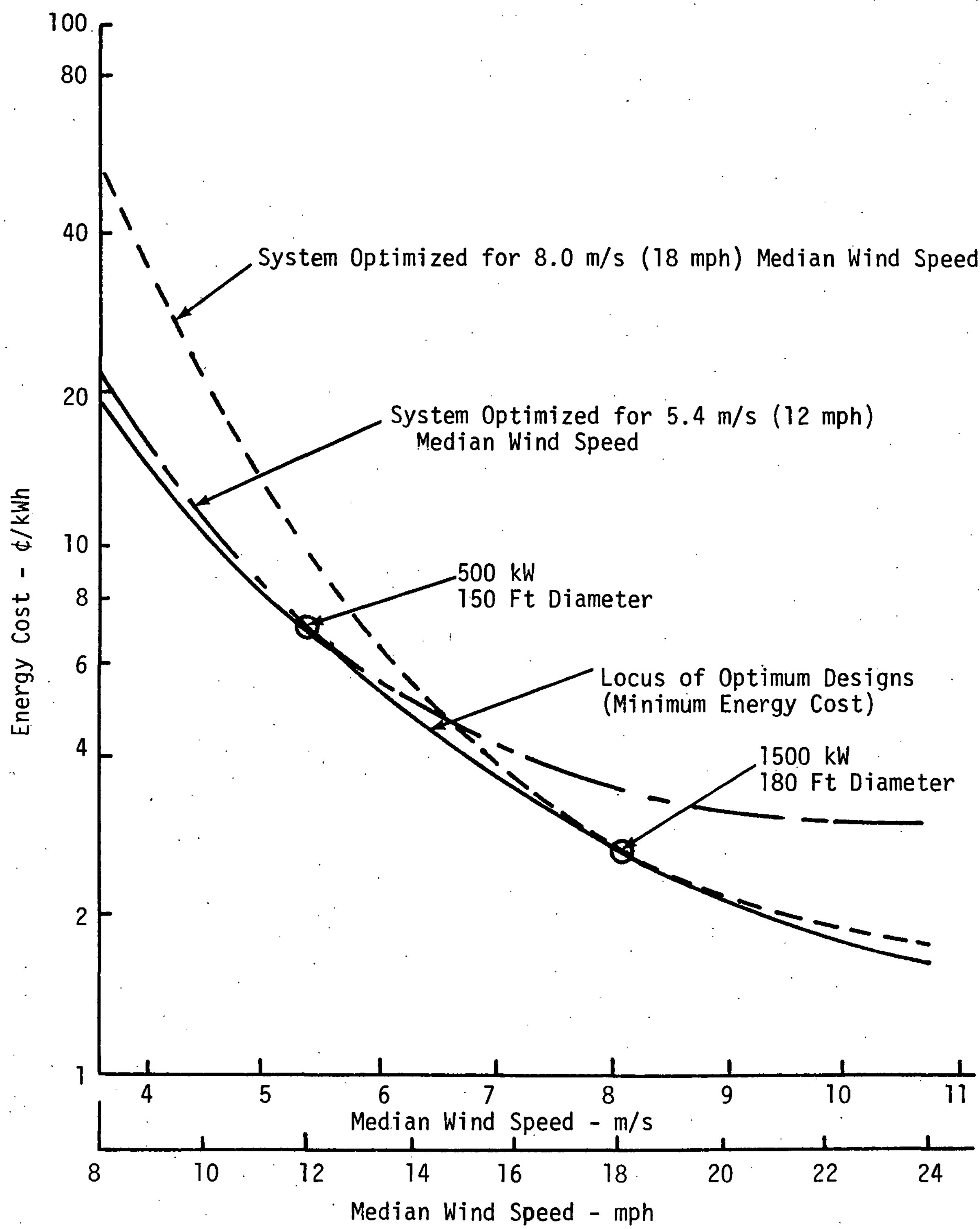


Figure 3-49. Energy Cost Variation With Median Wind Speed

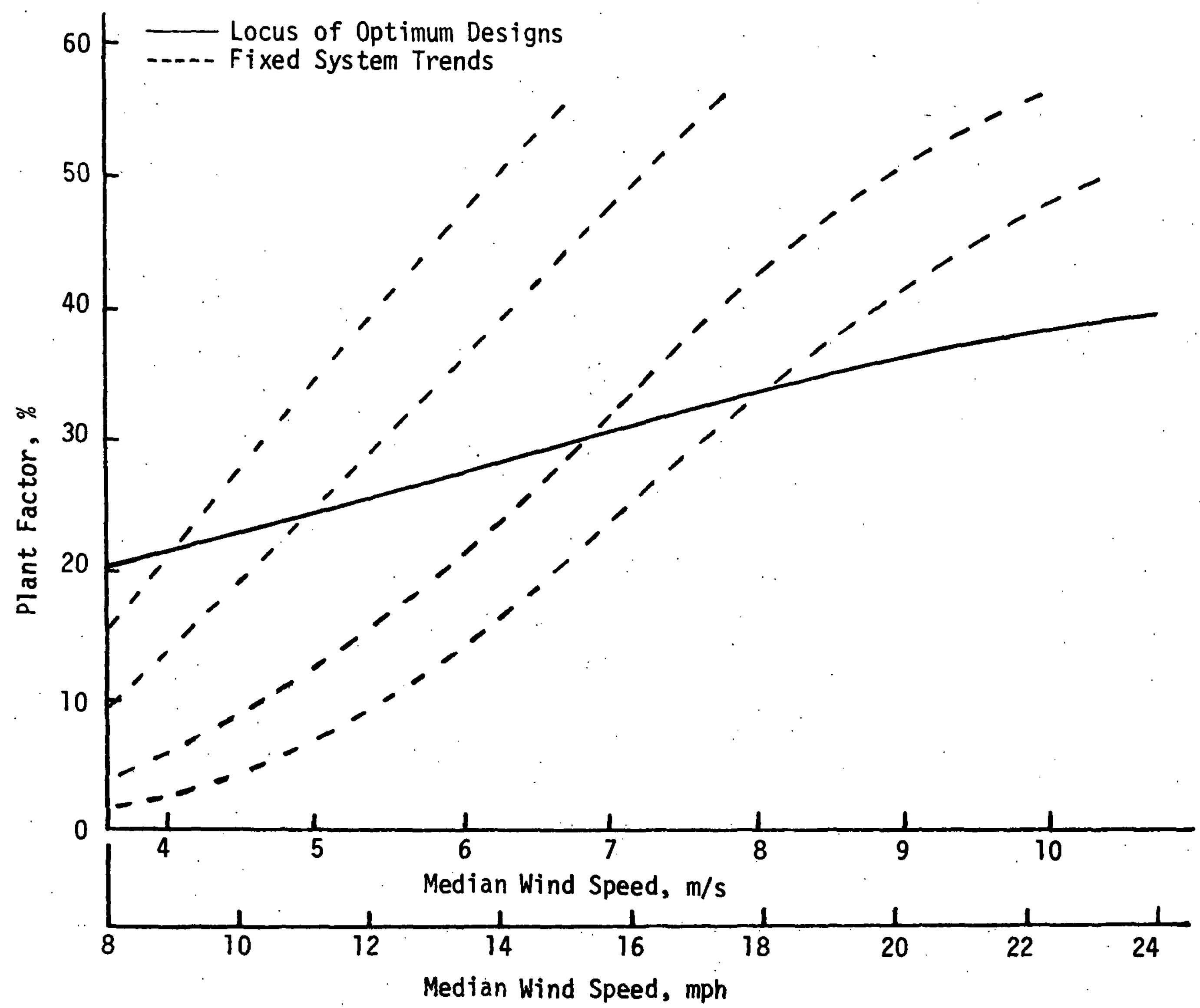


Figure 3-50. Plant Factor Variation With Median Wind Speed - Optimum and Fixed Systems.

parameters with wind shear exponent with the median wind speed at 9.1 m held constant. The results of this analysis are shown in Figure 3-51, with comparative values for systems optimized for the specific site wind shear.

For the range of wind shear exponents analyzed, the data of Figure 3-51 show that there are virtually no changes in energy cost at the different sites and only moderate differences between the fixed and optimized systems in direct capital cost, rated power and plant factor. For very large variations in wind shear exponent, larger changes in these variables would be experienced; however, if suitable sites with adequate preparation are selected for WGS installations, variation in site characteristics should be controlled within reasonable limits, and data of Figure 3-51 would indicate that little effect on energy cost would be experienced by a given machine.

Again, it is believed that these results, which are for the optimized high power system, would apply to the selected high power system and to the selected low power system, as well.

3.5.1.3 Site Altitude Effects

The effect of installing a WGS at a site higher than sea level is to reduce air density and hence, power output and energy production. This effect is shown in Figure 3-52, which compares the major system parameters for the optimized 1400 KW high power system (as before), operating at its design altitude, sea level, and at the maximum altitude specified by NASA for consideration, 1220 m (4000 ft). A system optimized for operation at the 1220 m altitude site is also shown for comparison.

As shown in the figure, rated wind speed and rotor diameter are approximately the same for both the sea level and optimized systems at 1220 m, but since the density is lower, the rated power of the optimum system is lower and hence, its direct capital cost is higher than the sea level machine. However, the overall effect on energy cost is very small, although some degradation in plant factor for the sea level machine does occur with increasing altitude. These results indicate that for the anticipated range of potential site altitudes, machines designed for sea level sites have acceptable characteristics when installed at higher altitude sites.

As was the case for the median wind speed and terrain effect analyses, it is believed that these conclusions apply for the selected high power system and low power systems, as well.

3.5.2 Component Options

Another method of examining the effect of installing WGS units designed for one site at a non-design site is to evaluate the machine characteristics if component options are utilized. In this approach, certain components of the WGS are changed when the machine is installed at non-design sites to either lower costs (if the machine is moved to a lower energy potential site), or to capture additional energy (if the machine is moved to a higher energy potential site). Components which are candidates for options are those which are either critical to system performance, or those which are easy and inexpensive to change.

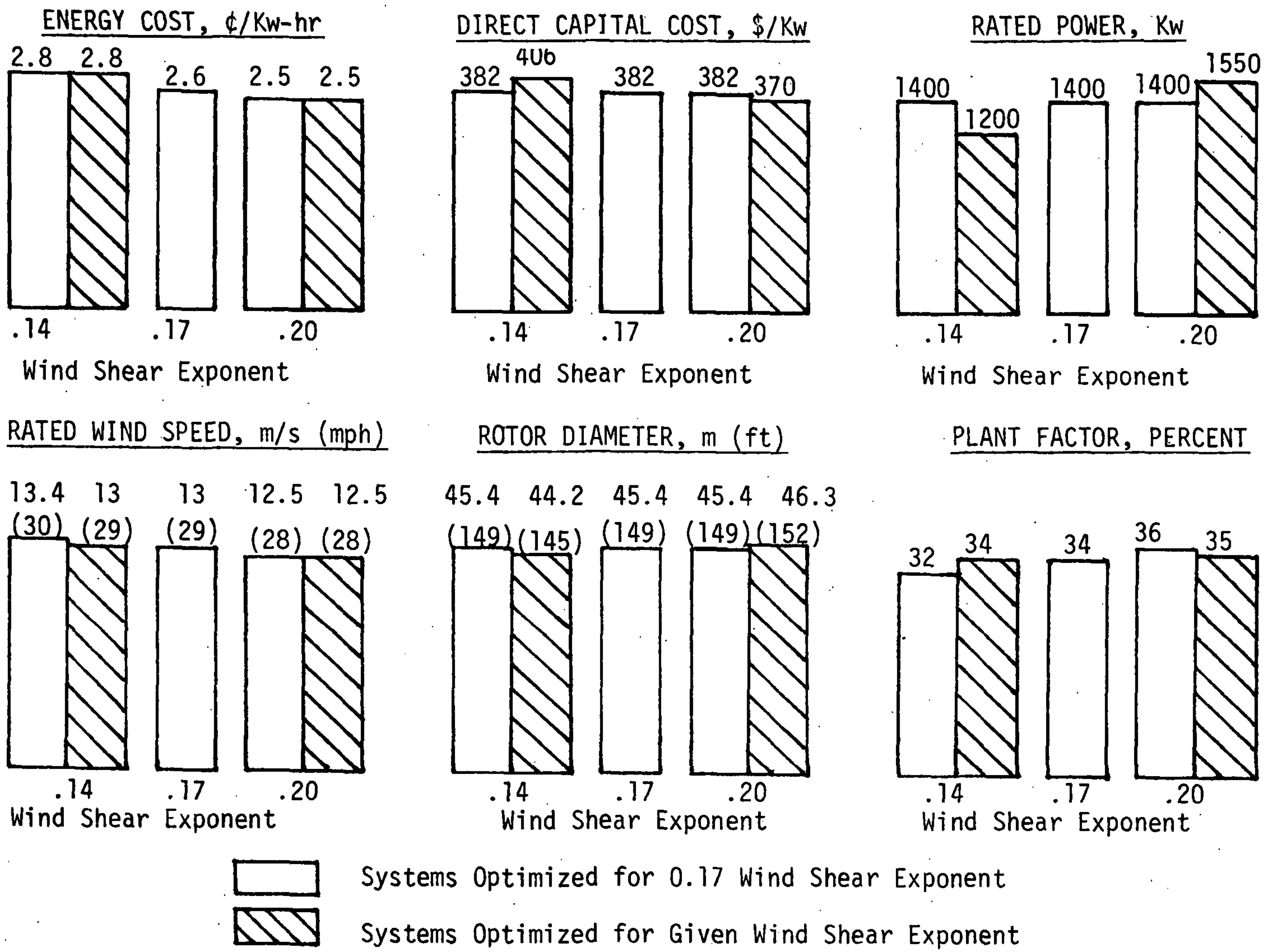


Figure 3-51. Site Terrain Effect on Fixed and Optimized Systems.

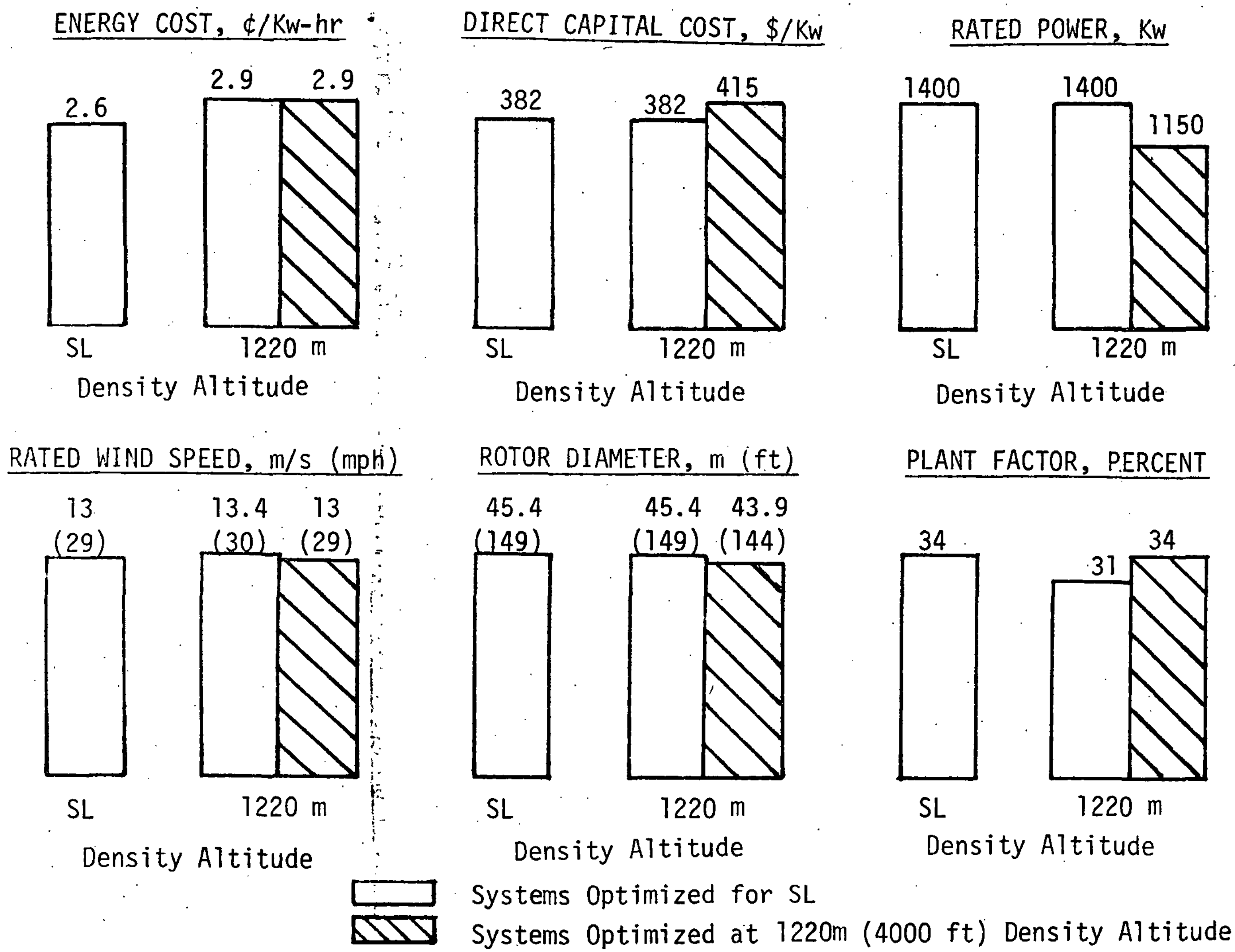


Figure 3-52. Site Altitude Effect on Fixed and Optimized Systems.

The analysis described below was conducted only for variations in site median wind speed because this site variable is the most significant to the system energy cost, and the results obtained can be extrapolated to conclusions concerning terrain and altitude effects.

The potential component options available to improve the characteristics of a WGS installed at a non-design site are summarized in Table 3-18. This table shows each component option, the benefit anticipated by exercising the option, and other significant system changes that may be required to accommodate the component option. For those sites where the winds are lower than design, changes to rotor diameter, rotor speed, gearbox and generator rating and tower strength were considered. The purpose of increasing rotor diameter or decreasing rotor speed when the site median wind speed is lower than design is to increase rotor output and/or rotor efficiency. Decreasing rotor speed is a straight-forward method of increasing rotor efficiency at lower wind speeds, since it is desired to keep the velocity ratio of the rotor at a high value as the wind speed is reduced (see Figure 3-9). Increasing rotor diameter to increase rotor output argues for a lower rotor speed, due just to the increase in diameter itself. In addition, some foundation strength increase may be necessary because of the larger blades and hence, higher blowover loads. The other option is to decrease gearbox and generator ratings, which would be accomplished together, to reduce cost. Since less energy is available at the given site, lower ratings on these components might be cost effective options. An additional minor option is to decrease the tower strength due to lower rotor loadings at the lower wind speed site, although savings here would be minimal.

Table 3-18 also shows the component options, potential benefits and other changes required when the site wind speeds are higher than design. The options available parallel those for the previous case. One option is to decrease rotor diameter, since more energy is available at the site and the larger rotor is unnecessary. This will reduce rotor cost. Increasing rotor speed, which is desirable to maintain the proper velocity ratio, would also be necessary to keep rotor efficiency high if the rotor diameter is decreased. And, increasing gearbox and generator ratings would have the objective of increasing the ability of the conversion machinery to utilize the additional energy available at the site.

Although the component options for the higher than design wind speed site parallel those for the lower than design wind speed site, their effects on the design are much more severe. Increasing rotor speed to maintain higher rotor efficiency at the higher wind speed site implies that more energy will be produced by operating the system at higher power levels, as well as for more hours of the year. Hence, the gearbox and generator rating increase really is necessary if the higher efficiency rotor option is exercised. And, if rotor speed is increased, it may be necessary to increase the tower stiffness to achieve acceptable lateral bending characteristics. Increasing the ratings of the power train may also require rotor structure modifications due to increased rotor loads. Hence, modifying the machine to take advantage of higher than design wind speeds implies parallel changes in many of the system components to achieve acceptable technical characteristics as more energy is harvested at high power levels.

TABLE 3-18. POTENTIAL TAILORING OPTION BENEFITS

COMPONENT OPTION	BENEFIT	OTHER SIGNIFICANT CHANGES REQUIRED
SITE WIND SPEEDS LOWER THAN DESIGN		
INCREASE ROTOR DIAMETER	INCREASED ENERGY OUTPUT	REDUCE ROTOR SPEED* INCREASE FOUNDATION STRENGTH
DECREASE ROTOR SPEED	INCREASED ROTOR EFFICIENCY	GEARBOX RATIO
DECREASE GEARBOX RATING	REDUCED COST	---
DECREASE GENERATOR RATING	REDUCED COST	---
DECREASE TOWER STRENGTH	(MINIMAL)	---
SITE WIND SPEEDS HIGHER THAN DESIGN		
DECREASE ROTOR DIAMETER	REDUCED ROTOR COST	INCREASE ROTOR SPEED*
INCREASE ROTOR SPEED	INCREASED ROTOR EFFICIENCY	GEARBOX RATIO GEARBOX RATING GENERATOR RATING TOWER STIFFNESS
INCREASE GEARBOX RATING	INCREASED ENERGY OUTPUT	POSSIBLE ROTOR STRUCTURE INCREASE INCREASE GENERATOR RATING
INCREASE GENERATOR RATING	INCREASED ENERGY OUTPUT	POSSIBLE ROTOR STRUCTURE INCREASE INCREASE GEARBOX RATING

*RECOMMENDED

A qualitative assessment of the impact of these changes is shown in Table 3-19, which gives the estimated impact on the system and its components due to the various options. As shown, any modifications to the rotor, either in diameter or operating speed, have significant impacts on the rotor design, even though the impact on other components may be minimal. These severe effects on the rotor design are largely due to the critical tuning requirements of large, flexible rotor systems, an effect which heavily influenced the WGS rotor preliminary designs. This requirement is discussed fully in Section 4, Rotor Subsystem.

It appears, therefore, that no minor options are available for the rotor and the only feasible component options are to increase and decrease gearbox and generator ratings. To assess the quantitative effects of these options, the optimum high power system (1400 KW, 46 m rotor) designed for the 8 m/s median wind speed site was analyzed. As was the case for the studies reported in the previous section, this system was chosen in lieu of the selected high power system so as not to distort results by utilizing non-optimum system designs.

The analysis was conducted by "moving" the reference design WGS to higher and lower median wind speed sites, varying the ratings of the gearbox and generator, and determining energy and direct capital cost changes. In this analysis, the other system components were assumed to be unchanged. This approach was taken to show the maximum benefit possible from component options, and may be indicative of the growth potential of the WGS.

The results of the analysis are shown in Figure 3-53, in non-dimensionalized form for generalization. As shown, the relative direct capital costs of the WGS will increase or decrease as the relative ratings of the gearbox and generator/transformer are varied. The major effect on capital cost (in terms of $/KW) is due to varying the ratings; the effect of gearbox and generator cost variations are relatively small.

For the 8 m/s (18 mph) median wind speed site, energy cost should minimize at a relative rating of 1.0, since the machine is optimized for minimum energy cost at this point. The reason that the energy cost declines slightly as relative rating is increased is due to the lack of secondary effects on the rotor, tower, etc., which are not accounted for in the analysis. However, the effect of rating increases is very small and reflects the lack of sensitivity of energy cost to rated power in the vicinity of the optimum rating.

At a lower wind speed site, in this case 6.7 m/s (15 mph), some saving in energy cost can be realized if the gearbox and generator/transformer ratings are decreased, as expected. The effect is very small, and can be partially attributed to the fact that the other system components (rotor, tower, control system, etc.) are not reduced in proportion to the rating. However, the saving would be small even if this effect were included.

At the higher median wind speed site of 9.4 m/s (21 mph), a greater benefit is realized by varying the gearbox and generator/transformer ratings. This is as expected, since the optimum rated power of a wind generator system increases with site median wind speed (as shown in Figure 3-25). Although the benefit appears to amount to as much as a 7% saving in energy cost as gearbox and generator/transformer ratings are increased, the savings may not be fully realized because of modifications which may be necessary to the rotor, turntable, tower,

TABLE 3-19. EFFECT OF WGS TAILORING OPTIONS ON SYSTEM DESIGN

	EFFECT ON SYSTEM AND COMPONENT DESIGNS					
COMPONENT OPTION	ROTOR	DRIVE	ELECTRICAL	TOWER	CONTROL	SYSTEM
SITE WIND SPEEDS LOWER THAN DESIGN						
INCREASE ROTOR DIAMETER	HI	---	---	LO*	---	HI
DECREASE ROTOR SPEED	HI	LO	---	---	---	HI
DECREASE GEARBOX RATING	---	LO	---	---	---	LO
DECREASE GENERATOR RATING	---	---	LO	---	---	LO
DECREASE TOWER STRENGTH	---	---	---	LO	---	LO
SITE WIND SPEEDS HIGHER THAN DESIGN						
DECREASE ROTOR DIAMETER	HI	---	---	LO	---	HI
INCREASE ROTOR SPEED	HI	LO	LO	LO*	---	HI
INCREASE GEARBOX RATING	LO*	LO	LO	LO	---	LO*
INCREASE GENERATOR RATING	LO*	LO	LO	LO	---	LO*

*POSSIBLE EFFECT MAY BE HIGHER

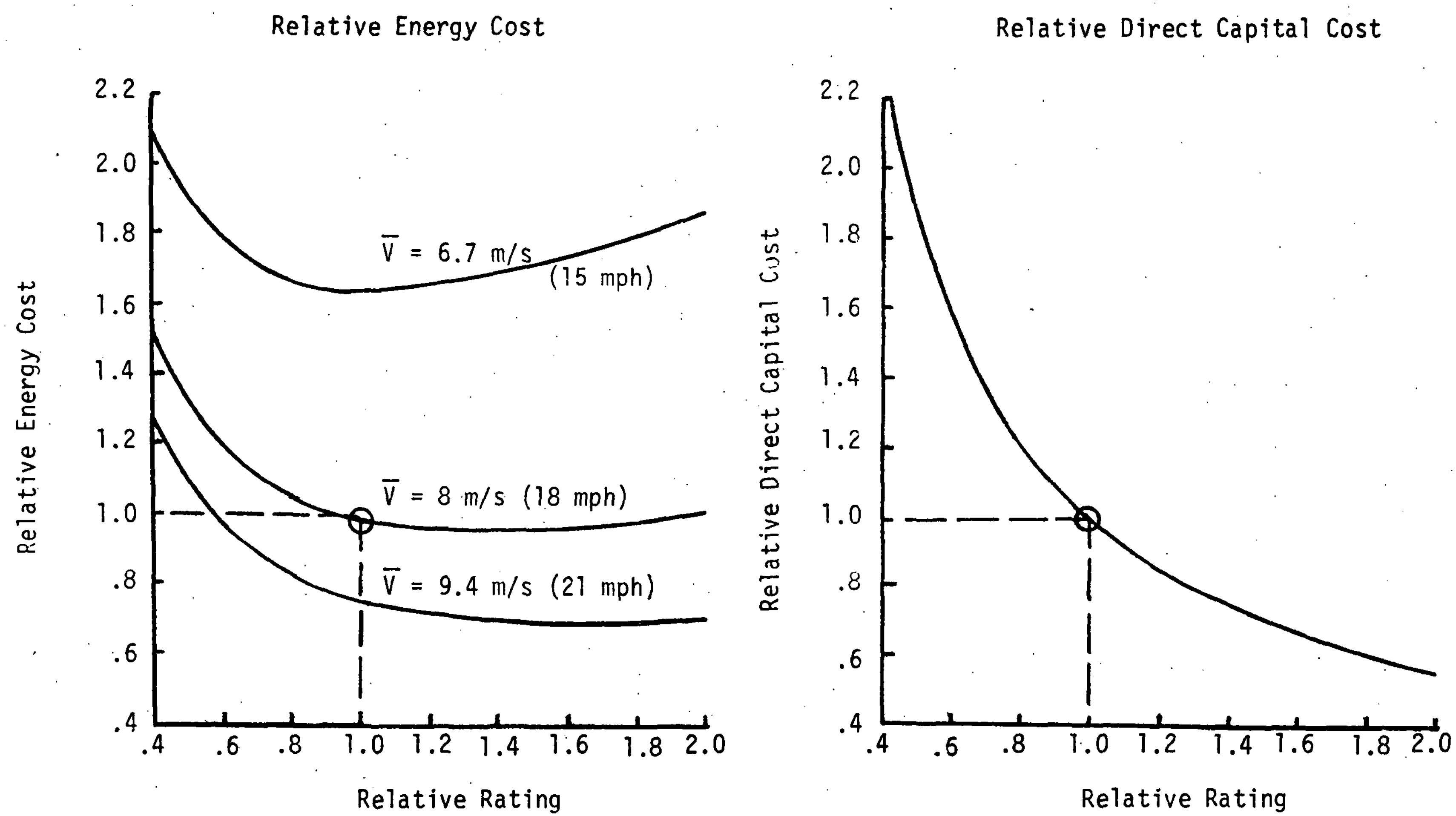


Figure 3-53. Effect of Gearbox, Generator/Transformer Rating Selection With All Other Components Held Fixed

etc., to carry the higher operating loads. If these component capabilities are not increased, then degraded component life may be the result, which will also increase effective energy cost.

The results of the analysis indicate that using component options to adapt specific WGS designs to different wind sites appear attractive only when the WGS is installed at a higher than design median wind speed site. It should be remembered, however, that moving a fixed machine to a higher wind site improves plant factor at the expense of a slight energy cost penalty (Figures 3-48, 3-49 and 3-50). And, the minimal energy cost benefit shown in Figure 3-53 may be partially or completely negated by upgrading other system components. The alternative, to accept degraded component strength and durability, will result in the same effect; a reduction in the apparent cost benefits shown in Figure 3-53.

Additional factors have to be considered if component options are offered for the WGS. These include the problems of interface matching of gearboxes and generators with other system components, and adjustments to the control system rates and software. Although gearboxes and generators are usually offered in a variety of ratings, and utilize the same basic hardware, there are often differences between machines as ratings are changed, the result of the specific applications for which the machines were developed in the past. This may require rewiring or rerouting of hydraulic lines and coolant paths, additional adapters to mount ancillary equipment and modifications to structural members for mounting provisions. These additional practical problems may also impact system cost, and would further argue against considering component options for adapting a fixed WGS machine to different wind speed sites.

3.6 WGS Preliminary Designs

Overall descriptions of the selected WGS preliminary designs are given below. These preliminary designs, for the 500 KW low power and 1500 KW high power WGS units, were based on the same configuration; the system concept selected at the end of the conceptual design phase. However, some changes and refinements to the component approaches were made and are also described below. Details on each subsystem are given in each subsystem section of this report.

3.6.1 System Descriptions

Overall layouts of the WGS preliminary designs are shown in Figures 7-1 and 7-2. Both WGS units utilize a two bladed rotor, where the blades are filament wound composite structures with hingeless attachments to a rigid hub. The hub is a rugged welded structural steel assembly. Blade pitch control is effected by a linkage system actuated by a hydraulic cylinder, which rotates the blades on pitch bearings mounted on the hub.

The hub is mounted on a fixed spindle supported by cross roller bearings. This allows the non-rotating spindle to carry the high bending moments produced by the rotor without the requirement for heavy bearings to support a rotating shaft. The rotor torque is transmitted to a triple mesh commercial gearbox by a quill shaft running through the center of the static spindle. The gearbox is connected to a commercial synchronous generator of the type used by utilities, with a parking brake/inching drive assembly located between the gearbox and the generator. This assembly is used to stop the rotor from low rpm levels to rest, and to rotate the rotor after the system is shutdown for maintenance operations.

The control system uses a microprocessor for all sequencing and data reporting functions, including WGS dispatching, startup, shutdown, operational monitoring and failure reporting. Conventional electro-mechanical controls are used for primary control functions (pitch control, orientation control, etc.). The rotor is provided with a hub mounted mechanical blade feathering capability so that if the control system fails, or is overpowered by environmental conditions (gusts), the blades are automatically feathered, preventing system overspeed and damage.

The rotating machinery assembly is mounted on a structural steel platform which, in turn, is mounted on top of the tower through a cross roller bearing assembly. Orientation is effected by use of a hydraulic motor driving a worm gear which engages a large ring gear on the top of the tower.

The tower can be either a structural steel truss type or a pre-cast, post-tensioned concrete shell type. Both mass concrete and pile type foundations can be utilized, depending upon the type and size of tower and the specific soil conditions at the WGS site.

Most of the electrical protective and power conditioning equipment is mounted on a pad at the base of the tower, along with the microprocessor and related control equipment and the data recording and transmission equipment.

3.6.2 Preliminary Design Characteristics

The high power and low power WGS preliminary design characteristics are summarized in Tables 3-20 through 3-24. Table 3-20 gives the major system and subsystem technical characteristics for both preliminary designs. Tables 3-21 and 3-22 give weight and cost breakdowns, respectively, for the 500 kW WGS preliminary design utilizing a steel truss tower. Tables 3-23 and 3-24 give similar information for the 1500 kW WGS, also utilizing a steel truss tower. The weights and costs presented in these tables are for both the preliminary designs prepared for the high power and low power systems and the corresponding values of weights and costs predicted by the parametric model. The reasons for differences between the design estimates and the model predictions are explained below.

Since the model uses the developed performance calculation process for the selected concept, the performance data predicted by the model is that predicted for the selected designs. Also, most of the major system dimensions (rotor diameters, blade chord, etc.) and operating parameters (rotor speed, generator speed, etc.) predicted by the model are those of the completed preliminary designs. The differences between the model and preliminary design results are due to refinements in estimated weights and costs developed through the detailed preliminary design effort.

Some of the weight and cost differences in Tables 3-21 to 3-24 are due to changes in component concept. These concept changes are summarized in Table 3-25 with notations for the reasons. As shown in the table, cost reduction was the impetus for most of the component concept changes. Each of these changes, as well as the original concepts, are discussed in detail in subsequent subsystem sections.

TABLE 3-20. SYSTEM TECHNICAL CHARACTERISTICS

	LOW POWER SYSTEM	HIGH POWER SYSTEM
SYSTEM		
Rated Power, kW	500	1500
Site Median Wind Speed, m/s (mph)	5.4 (12)	8 (18)
Rated Wind Speed, m/s (mph)	9.3 (21)	11.5 (26)
Yearly Energy Output, kWh	1.3×10^6	5.7×10^6
Plant Factor, %	29	43
Minimum Wind Speed, m/s (mph)	4.5 (10)	5.4 (12)
Cut-out Wind Speed, m/s (mph)	14 (30)	20 (45)
Design Maximum Wind Speed, m/s (mph)	54 (120)	54 (120)
ROTOR SUBSYSTEM		
Number of Rotors per Tower	1	1
Number of Blades per Rotor	2	2
Type of Rotor Control	Direct Control, Variable Pitch	
Blade Type	Multiple Cell, Filament Wound Composite	
Hub Type	Rigid	Rigid
Design Shaft Output Power, kW	560	1648
Rotor Diameter, m (ft)	45.7 (150)	54.9 (180)
Rotor Solidity, %	3	3
Rotor Precone, Deg.	8	10
Rotor Speed, rpm	32.3	34.4
Blade Root Chord, m (ft)	1.5 (5.1)	1.9 (6.1)
Blade Tip Chord, m (ft)	.75 (2.5)	.95 (3.1)
Airfoil Section	230xx	230xx
Blade Planform Taper	Modified 3:1, Mid-span to Tip	
Blade Thickness Taper	40% at Root to 12% at Tip	
Blade Twist	Optimum	Optimum
Root Cut Out	5%	5%

TABLE 3-20. SYSTEM TECHNICAL CHARACTERISTICS (continued)

	LOW POWER SYSTEM	HIGH POWER SYSTEM
DRIVE SUBSYSTEM		
Gearbox Type	Triple Mesh Gear	
Gearbox Input Torque, Nm (ft-lb)	166 (122) x 10^3	456 (357) x 10^3
Gearbox Input Speed, rpm	32.3	34.4
Gearbox Output Torque, Nm (ft-lb)	2870 (2120)	8400 (6240)
Gearbox Output Speed, rpm	1800	1800
Rotor Orientation Drive Type	Worm Gear Drive	
Maximum Orientation Drive Speed, rpm	1/3	1/3
Control System Power Supply	Hydraulic	
ELECTRICAL SUBSYSTEM		
Generator Type	Synchronous or Induction	
Generator Rating, kW	510	1522
Generator Output Voltage, KV	2.4	4.16
Generator Output Frequency, Hz	60	60
Generator Speed, rpm	1800	1800
Transformer Output Voltage, KV	As Required	
Emergency Power Supply	Battery	
CONTROL SUBSYSTEM		
Control System Type	Automated, Self-Monitored	
Primary Controls	Electro-Hydraulic	
Sequencing and Supervisory Controls	Microprocessor	
Rotor Pitch Rate, Deg/sec	5	5
Rotor Overspeed Limiter	Mechanical Pitch Drive	
Orientation System Yaw Rate, Deg/sec	2	2
STRUCTURE SUBSYSTEM		
Tower Type	Steel Truss (or Pre-stressed Concrete)	
Tower Height, m (ft)	33.5 (110)	38 (124)
Tower Base Span, m (ft)	9.5 (31)	10.7 (35)
Tower Cap Span, m (ft)	3.05 (10)	3.05 (10)
Foundation Type	Anchored Pile or Footing Foundation	

TABLE 3-21. SYSTEM WEIGHT COMPARISON

500 kW WGS

STEEL TRUSS TOWER

	WEIGHT*, kg (lbs)	
	ESTIMATED FROM PRELIMINARY DESIGN	PREDICTED BY PARAMETRIC MODEL
ROTOR SUBSYSTEM	8,020 (17,680)	6,440 (14,200)
BLADES	3,080 (6,800)	2,010 (4,430)
HUB	4,940 (10,880)	4,430 (9,770)
DRIVE SUBSYSTEM	16,780 (37,000)	12,520 (27,600)
GEARBOX	9,890 (21,800)	4,050 (8,920)
OTHER	6,890 (15,200)	8,470 (18,680)
ELECTRICAL SUBSYSTEM	2,820 (6,210)	2,990 (6,580)
CONTROL SUBSYSTEM	20 (50)	520 (1,150)
STRUCTURE SUBSYSTEM	44,950 (99,100)	28,830 (63,580)
STRUCTURAL STEEL	37,470 (82,600)	17,000 (37,500)
OTHER	7,480 (16,500)	11,830 (26,080)
TOTAL WEIGHT ON FOUNDATION	72,590 (160,040)	51,300 (113,110)

*WEIGHT ON FOUNDATION

TABLE 3-22. SYSTEM COST COMPARISON
500 kW WGS
STEEL TRUSS TOWER
1000 UNITS

	COST PER UNIT, $			
	ESTIMATED FROM PRELIMINARY DESIGN		PREDICTED BY PARAMETRIC MODEL	
SYSTEM INTEGRATION	14,400		---	
ROTOR SUBSYSTEM	110,000		142,140	
BLADES		(76,600)		(112,580)
HUB, including hub-mounted controls		(33,400)		(29,560)
DRIVE SUBSYSTEM, including gearbox, shafting and couplings, ring gear and yaw mechanism	78,000		83,770	
ELECTRICAL SUBSYSTEM, including generator, transformer, cable, utilities	43,800		54,410	
CONTROL SUBSYSTEM	31,400		11,350	
STRUCTURE SUBSYSTEM, including turntable, enclosure, ladders, tower (installed), foundation	97,400		89,990	
TOTAL WGS UNIT COST	375,000		381,660	
OTHER CAPITAL COSTS, including site acquisition and clearing, shed, installation of components	75,670		41,440	
DIRECT CAPITAL COST, $	450,670		423,100	
DIRECT CAPITAL COST, $/kW	901		846	
ANNUAL COSTS:				
15% Direct Capital Cost*	67,600		63,470	
Operation and Maintenance	23,280		25,780	
TOTAL YEARLY COST, $	90,880		89,250	
YEARLY ENERGY OUTPUT, kWh	1.28×10^6		1.28×10^6	
ESTIMATED ENERGY COST, ¢/kWh	7.1		7	

Subsystem components are described in the appropriate subsystem sections of this report. Subsystem cost trends are shown in Figures 3-54 through 3-58.

*Includes Income Taxes, Debt Service, Return on Equity and Depreciation (see Section 9).

TABLE 3-23. SYSTEM WEIGHT COMPARISON

1500 kW WGS

STEEL TRUSS TOWER

	WEIGHT*, kg (lbs)	
	ESTIMATED FROM PRELIMINARY DESIGN	PREDICTED BY PARAMETRIC MODEL
ROTOR SUBSYSTEM	17,430 (38,430)	12,590 (27,760)
BLADES	5,160 (11,390)	4,030 (8,890)
HUB	12,270 (27,040)	8,560 (18,870)
DRIVE SUBSYSTEM	35,180 (77,560)	28,160 (62,080)
GEARBOX	20,860 (46,000)	10,360 (22,840)
OTHER	14,320 (31,560)	17,800 (39,240)
ELECTRICAL SUBSYSTEM	6,950 (15,320)	6,730 (14,850)
CONTROL SUBSYSTEM	20 (50)	680 (1,500)
STRUCTURE SUBSYSTEM	69,360 (152,900)	58,130 (128,140)
STRUCTURAL STEEL	52,210 (115,100)	37,930 (83,610)
OTHER	17,150 (37,800)	20,200 (44,530)
TOTAL WEIGHT ON FOUNDATION	128,940 (284,260)	106,290 (234,330)

*WEIGHT ON FOUNDATION

TABLE 3-24. SYSTEM COST COMPARISON

1500 kW WGS

STEEL TRUSS TOWER

1000 UNITS

	COST PER UNIT, $			
	ESTIMATED FROM PRELIMINARY DESIGN		PREDICTED BY PARAMETRIC MODEL	
SYSTEM INTEGRATION	25,100		---	
ROTOR SUBSYSTEM	194,900		276,000	
BLADES		(122,000)		(218,900)
HUB, including hub-mounted controls		(72,900)		(57,100)
DRIVE SUBSYSTEM, including gearbox, shafting and couplings, ring gear and yaw mechanism	181,000		187,140	
ELECTRICAL SUBSYSTEM, including generator, transformer, cable, utilities	64,800		84,990	
CONTROL SUBSYSTEM	31,400		11,350	
STRUCTURE SUBSYSTEM, including turntable, enclosure, ladders, tower (installed), foundation	134,600		136,620	
TOTAL WGS UNIT COST	631,800		696,100	
OTHER CAPITAL COSTS, including site acquisition and clearing, shed, installation of components	89,000		52,850	
DIRECT CAPITAL COST, $	720,800		748,950	
DIRECT CAPITAL COST, $/kW	481		499	
ANNUAL COSTS:				
15% Direct Capital Cost*	108,120		112,340	
Operation and Maintenance	45,350		48,300	
TOTAL YEARLY COST, $	153,470		160,640	
YEARLY ENERGY OUTPUT, kWh	5.68×10^6		5.68×10^6	
ESTIMATED ENERGY COST, ¢/kWh	2.7		2.83	

Subsystem components are described in the appropriate subsystem sections of this report. Subsystem cost trends are shown in Figures 3-54 through 3-58.

*Includes Income Taxes, Debt Service, Return on Equity and Depreciation (see Section 9).

TABLE 3-25. COMPONENT CONCEPT CHANGE SUMMARY

COMPONENT	CONCEPT		REASONS
	IN PARAMETRIC MODEL	PRELIMINARY DESIGN	
ROTOR BLADE	Filament Wound Composite Spar Bonded Afterbody and Skin	All Filament Wound Composite Structure	Reduce Cost
ROTOR HUB	Flexplate	Rigid	High Stiffness Required
DRIVE SHAFT	Rotating Shaft with Support Bearings	Fixed Spindle with Quill Shaft	Reduce Loads and Cost
TURNTABLE BEARING	Shaft with Support Bearings	Cross Roller Bearing	Reduce Cost
CONTROL SYSTEM	All Electro-Mechanical	Electro-Mechanical Primary Microprocessor Sequencer	Increase Flexibility, Reliability, Reduce Cost

As shown in Tables 3-21 and 3-23, weight differences exist between the preliminary design estimates and those predicted by the parametric model. Rotor blade weights are heavier, due to the increased structural requirements discovered in the analysis portion of the preliminary design phase. Hub weight differences, which are as much as 50% for the 1500 kW system, are due to the change from the flex plate concept in the parametric model to the rigid welded structure concept employed in the preliminary design.

Weights for the drive subsystem are higher in the preliminary design estimates as compared to the parametric model prediction. The weight increases are due to the gearbox which, for the preliminary design, has been selected for higher cyclic loads than the original gearboxes used to formulate the parametric model. However, the "other" category, which includes the drive shaft and support bearings, is lower in weight due to the use of a static spindle and quill shaft for rotor support in the preliminary design, as opposed to the rotating shaft and pillow block/bearing approach used in the parametric model.

The difference in model prediction versus preliminary design estimate for the control subsystem is primarily a bookkeeping change, where some of the weight previously charged to the control system is included in the drive subsystem of the preliminary design. The drive subsystem for the preliminary design includes the hydraulic supply and actuators for pitch and yaw control.

Both the 500 and 1500 kW system structural subsystems have substantially higher weights for the preliminary designs than those predicted by the parametric model. Most of this weight increase is due to the increase in structural steel. The structural steel increase is due to the use of heavier members to achieve their recommended cross-sectional area to length ratios. The parametric model tended to utilize members which had optimistic cross-sectional area to length ratios when scaling the tower from the original baseline designs.

The net effect of these weight changes in each subsystem results in a 40% higher system weight (on the tower) for the 500 kW WGS, when compared to the parametric model prediction, and a 20% increase in system weight for the 1500 kW WGS. Some of these weight increases, particularly the hub weight increase, are due to a necessary technical change in the characteristics of the component (in this case, the need for a high stiffness hub), followed by a decision to utilize a fabrication approach which would minimize cost. Hence, weight was usually a second order consideration in the development of the preliminary design and, aside from minor cost increases for shipping the WGS components to the site and erecting the system, these weight increases are important only insofar as they affect costs.

The system cost comparisons, summarized in Tables 3-22 and 3-24, are the data most significant for comparing the results of the preliminary design to the model predictions. As shown in these tables, an additional cost element, system integration, has been added to the preliminary design cost estimate, which was not included in the parametric model. As normally defined, system integration costs cover all interconnecting wiring, interface adapters and connectors, subsystem interconnections for hydraulic lines, instrumentation cables, etc., and other hardware required to integrate the subsystems into an entire system. In addition, system integration includes the labor costs associated with the installation of the hardware.

A significant decrease in rotor cost from the parametric model to the preliminary design is shown in the tables. There is some increase in hub cost, for both the 500 kW and the 1500 kW systems, due to the more massive structure of the hub, even utilizing the lower cost fabrication approach. However, a significant decrease in cost is shown for the rotor blades. The parametric model was based on a composite blade utilizing a filament wound spar with a bonded Nomex afterbody and skin. As shown before, the high leverage that rotor cost exerts on the system optimization caused a re-evaluation of this fabrication approach during the preliminary design. At this point in the study, results from the Army blade development program mentioned previously (paragraph 3.3.3) were evolving, and the potential of fully automated, composite filament wound rotor blades were assessed for the WGS application. Extending the fabrication technology of the Army helicopter rotor blade to the WGS rotor scale was judged practical, within the state-of-the-art and, most importantly, offered substantially lower fabrication costs. The blade design was then reformulated to utilize an almost totally automated filament wound spar/skin blade and resulted in the reduced rotor subsystem cost.

The costs predicted for the drive, electrical, control and structural subsystems, taken as a whole, are approximately equal to that predicted by the parametric model for both the 500 kW and 1500 kW systems. Drive subsystem costs were approximately the same as predicted, even though the gearbox duty rating was increased. The original gearbox cost predictions assumed additional cost would be required to improve normal commercial quality gearboxes to best commercial quality gearboxes. In fact, this was not the case, and the preliminary design gearbox cost was close to predicted, even with the higher load capability. In addition, substantial cost savings were realized for the rotor shaft and bearing assembly through the use of the static spindle and cross roller bearing mounted hub design.

The electrical subsystem cost reduction is in part due to the fact that quantity discount prices were used for generators in the preliminary design phase of the study, and a reduction in cost of some of the protective equipment necessary for the electrical subsystem. A contributing factor was the decision to utilize a tower base mounted pad for housing the transformer and protective equipment as opposed to the original concept which utilized a pad approximately 200 ft from the base of the tower.

The control subsystem cost increase for the preliminary design is due, almost entirely, to the more complete analysis performed in the preliminary design phase and the additional control functions and provisions which have been included in the system.

Tables 3-22 and 3-24 show that the structure subsystem, which includes the tower, foundation, turntable and enclosure, costs approximately the same as predicted by the parametric model. Although the weight of the tower is substantially higher for both the 500 kW and the 1500 kW systems, the cost of the tower was revised in the preliminary design phase with more complete and accurate cost data. These data indicated a lower cost per unit weight of structural steel (erected on site), offsetting the cost of the additional steel in the preliminary design tower.

The total WGS unit (production) costs shown in Tables 3-22 and 3-24 are slightly lower than those predicted by the parametric model; almost 10% lower for the 1500 kW system and 2% lower for the 500 kW system. However, the other capital costs shown in the tables, which include the cost of the site, site preparation, shipping the WGS components to the site, erecting the system and checkout testing are somewhat higher than predicted by the parametric model. These higher costs are due to revised estimates for installing and checkout testing of the WGS units on site. The net result is a slight increase in direct capital cost for the 500 kW WGS, compared to the parametric model prediction, and a slight decrease in direct capital cost for the 1500 kW system. The net result is that both direct capital costs and energy costs are approximately the same for the preliminary design estimates as those calculated by the parametric model.

The above cost predictions have been based on quantity production in a fully-developed market for wind generators. The costs of first units, in a preproduction-prototype configuration with a limited market, have been estimated separately and are presented in Appendix E of this report.

3.7 Cost of Major Subsystems

The costs of major subsystems are presented simply as functions of rotor diameter or rated power, whichever is more appropriate for the particular subsystem under consideration. For example, rotor system cost is primarily a function of rotor diameter, whereas drive system cost is primarily a function of system rated power. These costs are shown in Figures 3-54 through 3-58 for the parametric model based on the conceptual design. Also shown in these figures are results of parametric cost analyses of the preliminary design, analyses which were performed after the main body of work in this WGS study was completed.

More specific definition of the WGS subsystem components in the preliminary design provided the basis for vendor cost quotes which were used in the preliminary design parametric model. By comparison, cost trends for the conceptual design were based on historical data related to component weights, complexity, power rating etc., and were, therefore, less precise than vendor quotes for the preliminary design.

Other factors which contributed to differences between cost trends of the conceptual design and the preliminary design are simplifications for ease of manufacture, particularly in the rotor subsystem, and reoptimization for minimum energy cost. The revised rotor cost model is shown in Figure B-1, Appendix B.

The WGS parametric model and baseline values of Table 3-11, which were used for the system optimization studies described previously, were also used for the major subsystem cost trends shown in Figures 3-54 through 3-58. These cost trends are optimized values based upon minimum energy cost. Note that optimum combinations of rotor tip speed and rated power were determined for each rotor diameter investigated. Results of this suboptimization task are shown in Figures 3-59 and 3-60.

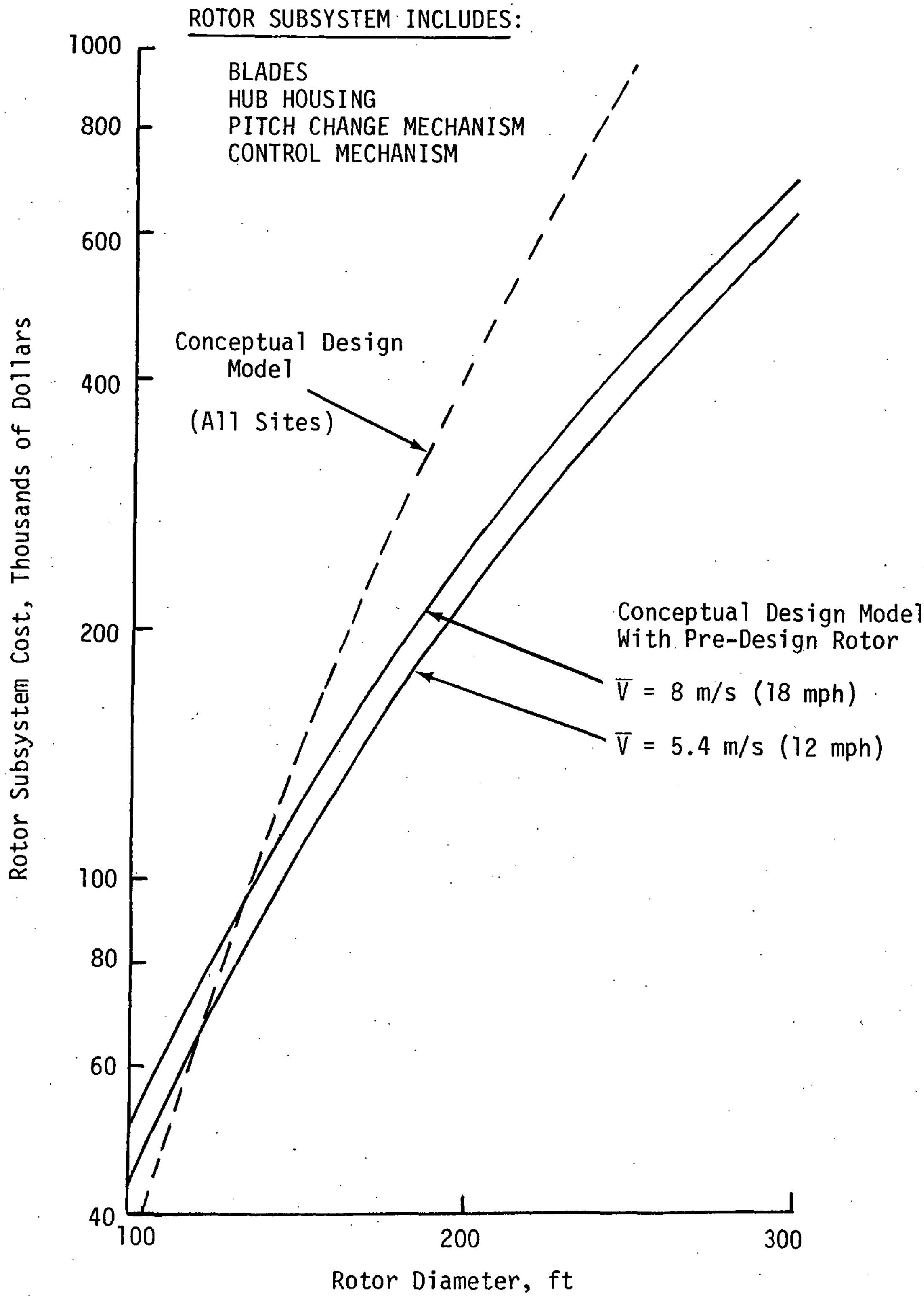


Figure 3-54. Rotor Subsystem Cost

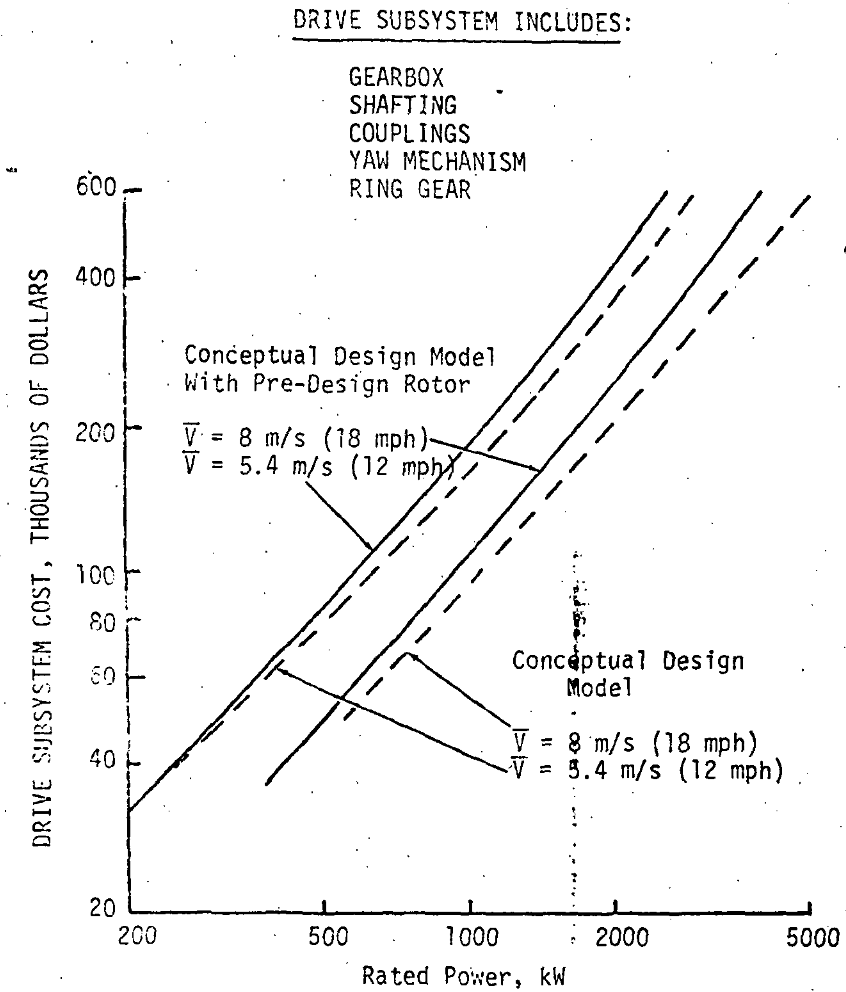


Figure 3-55. Drive Subsystem Cost

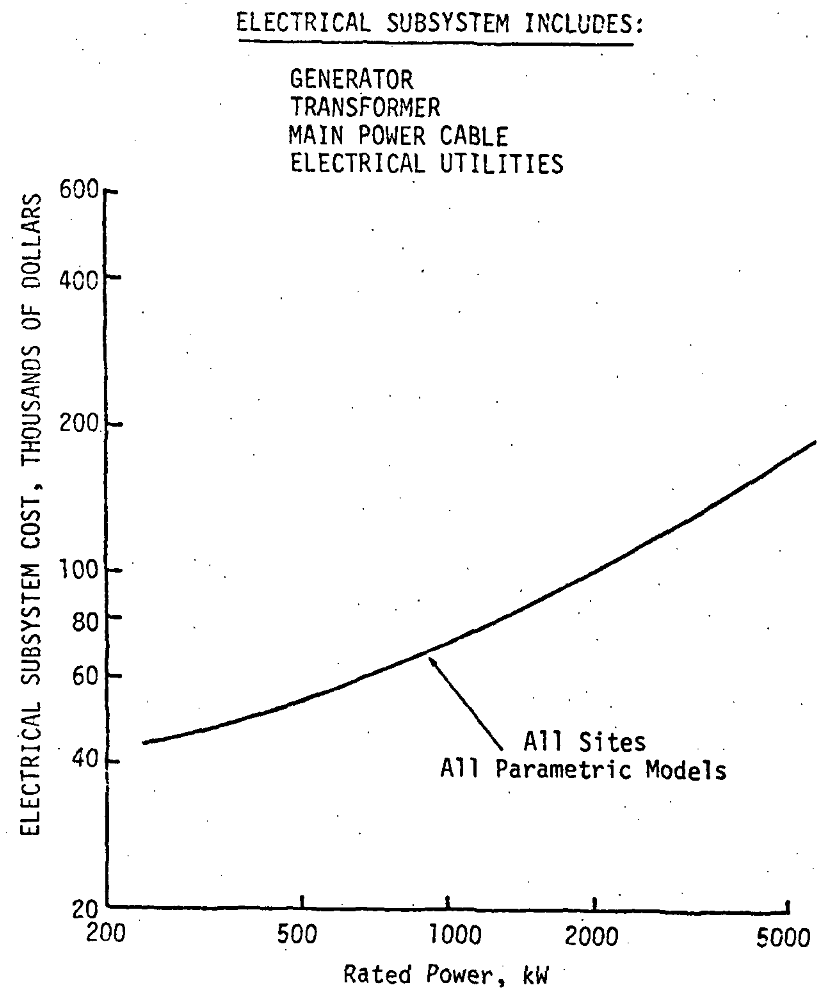


Figure 3-56. Electrical Subsystem Cost

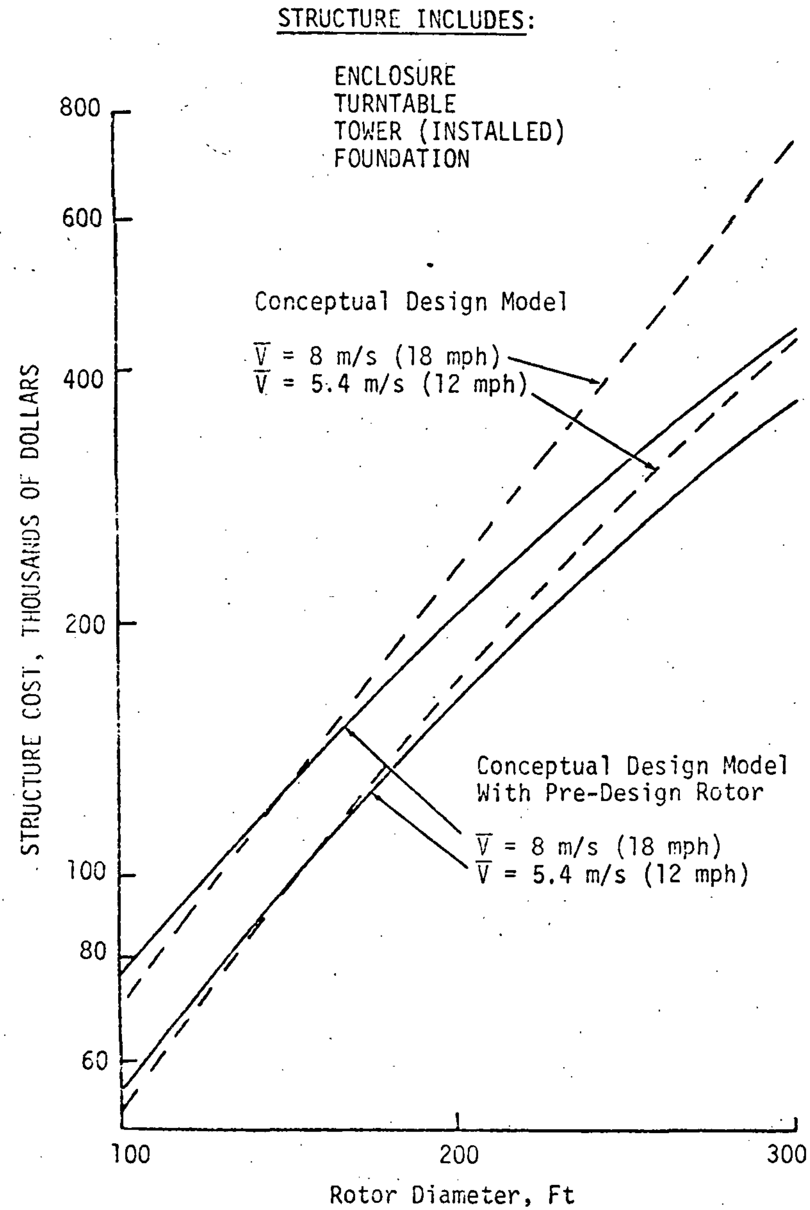


Figure 3-57. Structure Cost

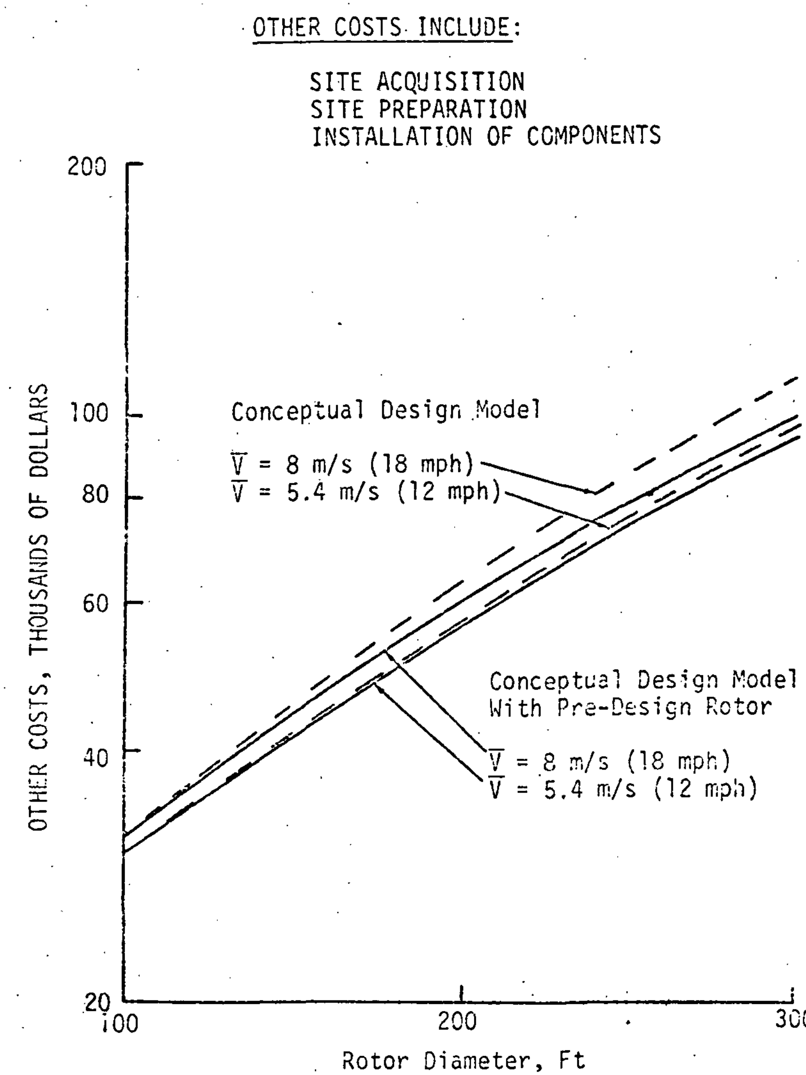


Figure 3-58. Other Costs

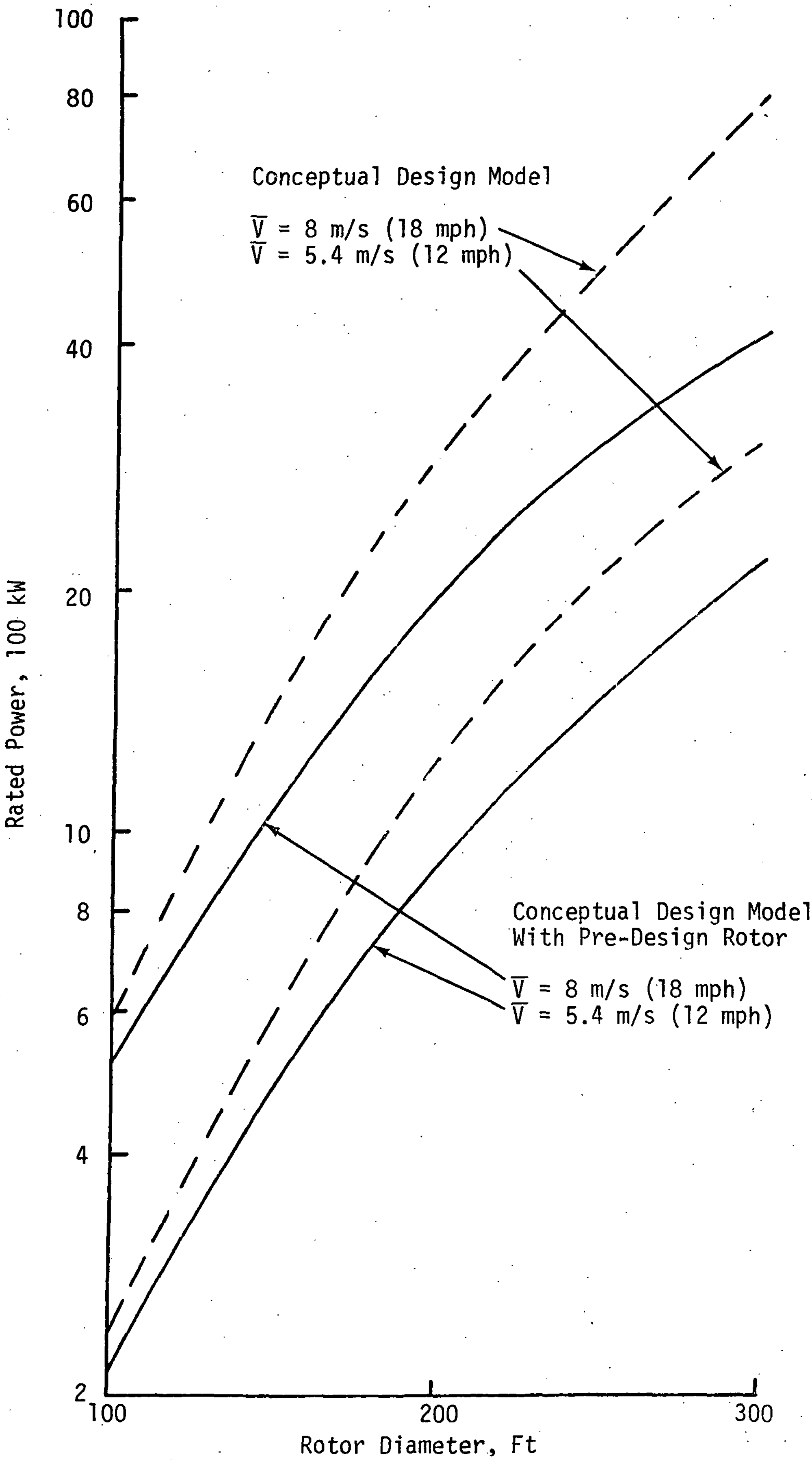


Figure 3-59. Rated Power for Minimum Energy Cost

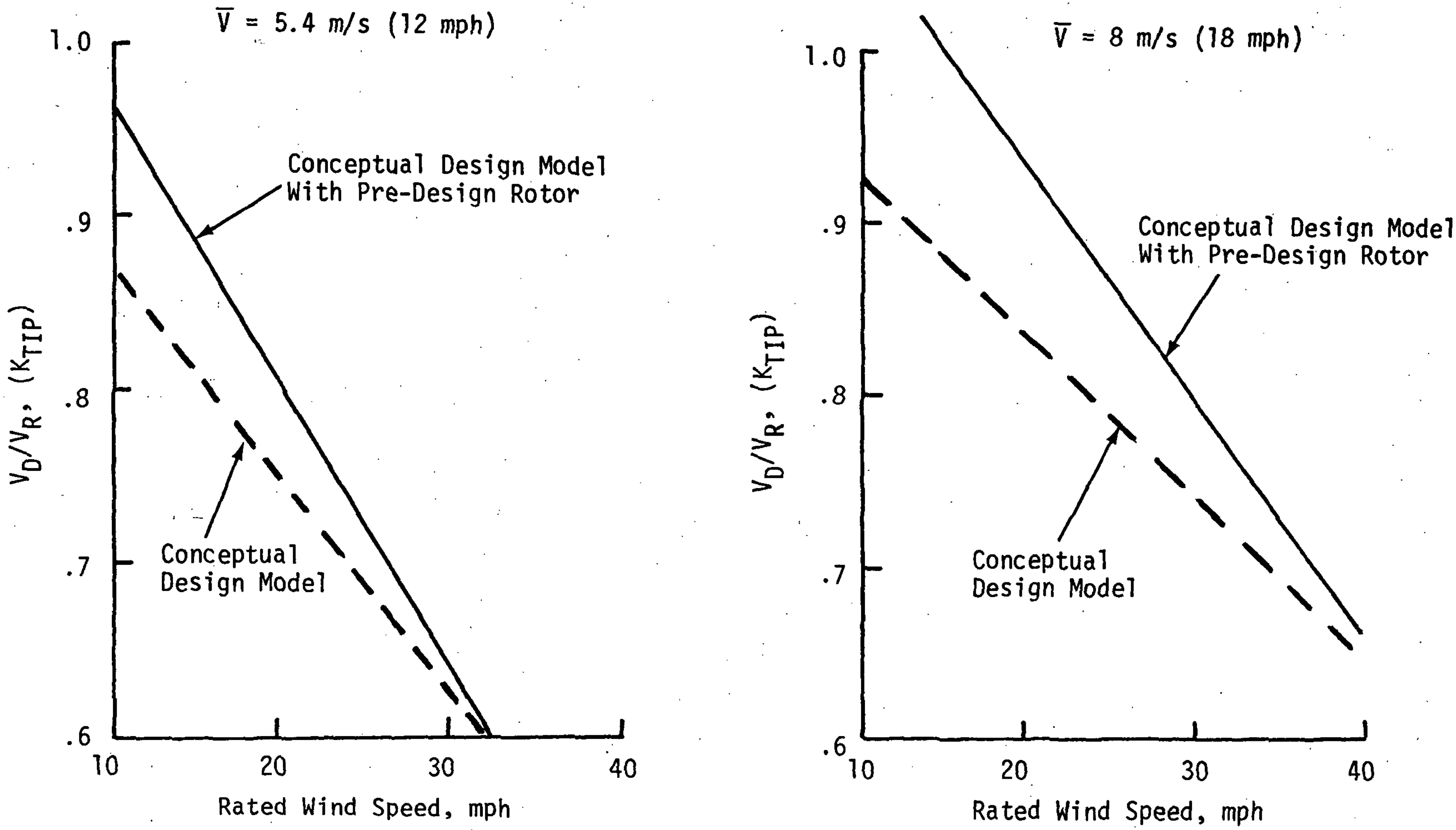


Figure 3-60. Rotor Tip Speed Selection Parameter (V_D/V_R) for Minimum Energy Cost

The cost of the control subsystem is assumed to be constant, independent of both rotor size and rated power. It is described in Section 5, and includes only the electrical and electronic devices that govern rotor and yaw mechanism operation, telemetry and supervisory functions, and fault monitoring.

Component costs associated with two median wind speeds (12 and 18 mph) are shown, representing the influence of wind frequency distribution on optimum WGS sizing. Drive system costs are considerably higher for low wind speed applications because optimum rotors are larger in diameter and slower turning, thereby producing higher drive system torques, the primary factor in drive system cost. For structure subsystem costs, it was assumed that rotor ground clearance is 50 feet for all cases.

3.7.1 WGS System Cost Trends

The preceding subsystems cost trends for the conceptual design and preliminary design parametric models were summed to obtain cost trends for the entire WGS system. Direct capital cost as a function of rated power is shown in Figure 3-61, and energy cost as a function of rotor diameter is shown in Figure 3-62. These trends show distinct shifts in the size of optimum systems toward larger diameter rotors and higher rated power levels. These shifts are primarily the result of the revised parametric model for the rotor (discussed in paragraph 3.8), and the associated reoptimization for minimum energy cost. Note that systems having minimum energy cost (¢/kWh) do not necessarily have the lowest direct capital cost ($/kWh).

3.8 Revision of the Rotor Parametric Model

The parametric analysis of the preceding sections was conducted using a WGS model based on the conceptual design. Subsequently, the rotor weight and cost equations were revised to reflect results of the preliminary design, which showed significantly lower costs and changed trends from the original parametric model.

3.8.1 Rotor Weight Equation

As a result of preliminary design, the hub design was changed from the conceptual configuration. Changes were made for structural requirements which became apparent during loads investigations, and for cost savings. The new configuration was subdivided into new groupings, according to their design function. The driving parameters were determined for each group, from which a functional relationship was determined. The coefficients and exponents were determined from the preliminary design estimates:

Blade Weight (WBLD). Blade weight is a function of blade length. It should be pointed out that the blade weight coefficients are based on preliminary design estimates, therefore, they are valid only for the preliminary design blade geometry.

Controls (WRCL). This group consists of hub-mounted blade pitch controls. The sizing of these items is strongly influenced by rotor maximum thrust and thus, is proportional to diameter squared.

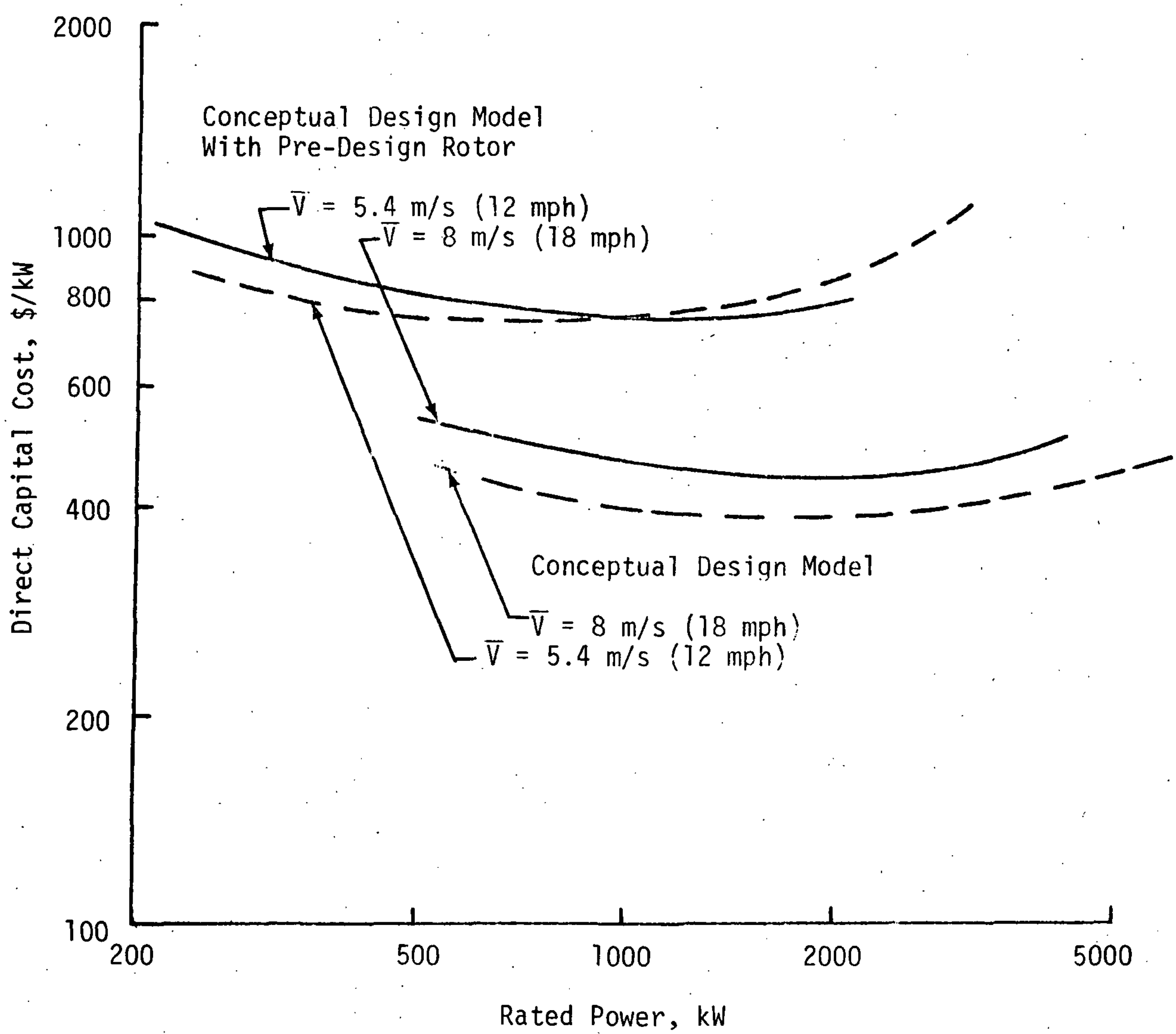


Figure 3-61. WGS Direct Capital Cost

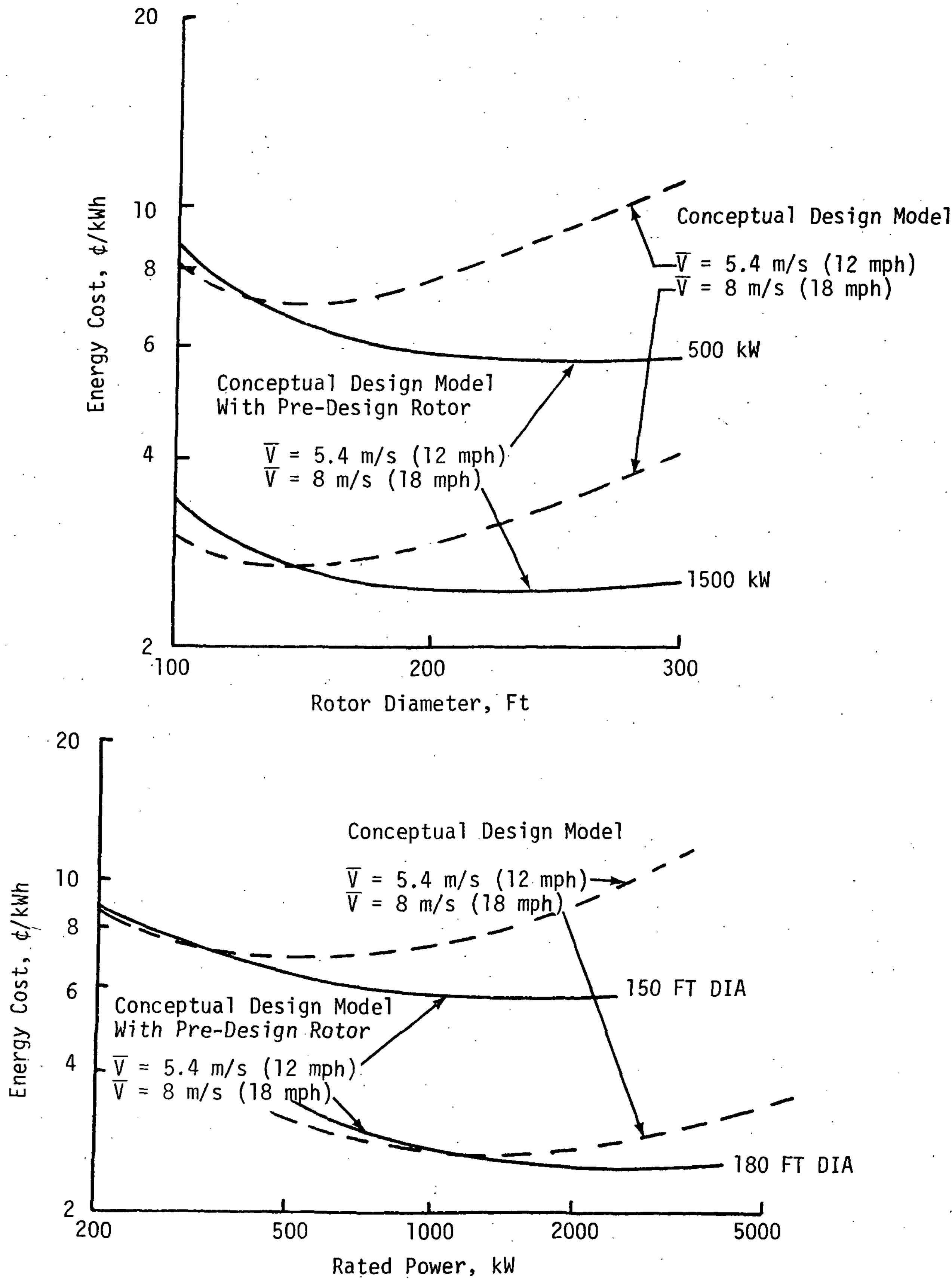


Figure 3-62. Effect of Rotor Diameter and Rated Power on Energy Cost

Barrel Assembly (WBRL). This assembly includes the hub weldment and the pitch and hub bearings. The barrel weight is related to maximum (rated) torque levels and hub size. The latter is assumed to scale with rotor diameter.

Grip Assembly (WGRP). The blade grips are a function of barrel weight and grip length. The latter is, of course, related to the distance from center line of rotation to blade attachments (RCUT). The equation is not valid for a RCUT less than 5%.

Hub Weight (WHUB). The total hub weight is the sum of these three components.

Rotor Weight (WROT). Rotor weight is the sum of total weight of blades and hub weight.

The equations are listed in Figure B-2, Appendix B. Note that the equations yield weight in kips; i.e., 1000 lb units.

3.8.2 Rotor Cost Equations

The ground rules for cost data in the parametric model were retained for the revised rotor cost equations. All costs are based on 1000 WGS production rate as expressed in constant 1975 dollars. Costs are those of manufacture and assembly, including the associated overhead and G & A rates and profit. In the case of subcontracted work, subcontractor profit is included.

The rotor cost is the sum of the costs of blades and hub. The hub is subdivided into three groups of components as determined for the weight estimates.

The blade cost (CBLD) is a function of both weight and length, the former is indicative of material needed. The latter parameter accounts for assembly and finishing operations. The coefficients of the blade cost equation were derived using preliminary design cost estimates for a 2000 blade production run.

The hub was assumed to be manufactured and assembled by the prime contractor. Cost estimates are based on manufacture of similar parts. The exceptions are the bearings, which are purchased at catalog prices. The coefficients for the barrel (CBRL), hub-mounted controls (CRCL), and grips (CGRP) represent the average cost per unit weight to manufacture or buy these components.

The total hub cost (CHUB) is then the total of the preceding three hub components, and the total rotor cost (CROT) is the total cost of two blades plus the hub cost. The rotor related cost equations are listed in Figure B-2, Appendix B.

3.9 Conclusions and Recommendations

The system analyses conducted as a part of this WGS study have led to several major conclusions and recommendations on system configuration, size, site adaptability and component ratings. The most significant of these conclusions and recommendations are summarized below.

3.9.1 Conclusions

1. Site wind speed has the greatest influence on WGS energy cost, more than any other site characteristic or WGS design parameter.

2. The WGS concept utilizing a direct rotor-gearbox-generator power train with the rotor operated at constant speed offers the lowest energy cost. System concepts which utilize variable speed rotors offer only slight improvements in total energy generated, but add substantial complexity and cost for the equipment necessary to tailor power output to conventional utility requirements.

3. For a given site, WGS energy cost is relatively insensitive to rated power and rated wind speed over a wide range of rated power. This permits selecting these parameters to meet user requirements. A major tradeoff exists between the capital cost of a WGS unit and its plant factor, within this insensitive power rating range, when rated wind speed is varied.

3.9.2 Recommendations

1. When installed at non-design sites, the selected WGS designs for the 5.4 m/s (12 mph) and 8 m/s (18 mph) sites offer energy costs close to systems optimized for the specific site. Hence, to minimize development program costs and risk, it is recommended that development effort for utility WGS units be limited to these two designs.

2. Component options to tailor the selected WGS designs for non-design sites do not improve system energy cost characteristics. Component options either require major system development effort or offer little benefit. Hence, it is recommended that no significant compromise of the system designs be made to accommodate component options.

4.0 ROTOR SUBSYSTEM

The rotor subsystem, which includes the rotor blades, hub and mechanical controls, is the largest single production cost element of the WGS. It is the most technically demanding subsystem and, unlike the other WGS subsystems, is not available as an off-the-shelf component. Therefore, the rotor subsystem will also be the most costly element of the WGS development program.

For these reasons, the rotor subsystem design and analysis received heavy emphasis in the study program. Particular attention was placed on evolving a rotor design with low cost mass-producibility, utilizing the latest state-of-the-art technology. However, equal emphasis was placed on defining the technical requirements of the rotor and translating these requirements into design characteristics. Both of these facets of the WGS rotor, low cost and technical adequacy, are critical to the successful employment of the WGS in electric utilities.

The rotor subsystem design was developed, as was the overall system design, through the conceptual design, optimization and preliminary design tasks of the study. Much of the design and analysis effort conducted in the early phases of the program was based on preliminary system definitions, and served as the basis for the system and rotor subsystem optimization and subsequent preliminary design. Therefore, results of these efforts are briefly described to illustrate the evolution of the final preliminary designs and analyses. A number of component option evaluations conducted on the blades, hub and controls are described in detail to illustrate the significant rotor design selections. As was the case for other subsystems and components, some of these option evaluations were conducted at points in the study when the final system configurations were not yet defined. However, the results of these evaluations are believed valid for the final selected systems and repeating the evaluations would not alter the rotor configuration selections.

4.1 Requirements

In order for the WGS to be a viable alternative for electrical energy production, the rotor must be capable of extracting maximum energy from the available winds at its site. It must survive a variety of weather conditions, ranging from extremes of temperature to all types of precipitation. It must survive lightning strikes and hurricane winds. And to be cost effective, the rotor must be comparatively economical to build, require minimum maintenance and have a long operating life.

Pursuant to these requirements, the following design objectives were established for the rotor subsystem:

1. Maximize aerodynamic efficiency of the rotor

2. Select the simplest design that will achieve the desired aerodynamic performance objectives

3. Ensure that the rotor will meet its requirements under the following environmental conditions:

Temperatures from - 51°C to + 49°C

Rain, sleet, hail, snow, sand, dust and sunlight

Salt spray

Lightning

Foreign object damage (birds, stones and bullets)

Winds up to the maximum anticipated gusts during operation, and hurricane winds up to 53.7 m/s (120 mph) while stowed.

4. Provide sufficient durability and fatigue strength to achieve a 30 year operating life for major dynamic components (hub, blades and grips).

5. Utilize available rotor technology to the maximum extent possible to minimize production cost and development risk and reduce development costs for new technology.

These design objectives were established at the outset of the program and expanded and refined as the rotor design was developed. They include both the stated requirements of the WGS rotor as given in Appendix A, and the objectives Kaman established to ensure that the technical and cost requirements of the rotor subsystem were met by the preliminary design.

4.2 Design Approach

The approach to the rotor subsystem design and analyses paralleled those for the other WGS subsystems. The conceptual design task examined different rotor configurations and selected the configuration which offered the lowest cost and risk characteristics while meeting the WGS requirements. This was followed by the optimization task which included evaluations of components in greater depth. Finally, the preliminary designs were prepared with detailed drawings, weight estimates, cost estimates, specifications and supporting analyses.

Throughout the study, the latest proven state-of-the-art technology was employed to minimize cost and development risk. Rotor configurations examined were proven concepts adopted from helicopter designs. When feasible, simplifications of these proven designs were made to eliminate unnecessary cost elements. Standard commercial components (such as pitch bearings) were selected where possible, and low cost fabrication processes were emphasized.

In the design process, particular attention was paid to those operating modes and conditions which have significant effects on the rotor design. Rotor starting operations, tower/rotor clearance during blade tip excursions and low rpm stability of the rotor, for example, were examined to determine what design restrictions were necessary to insure safe rotor operation. In addition, emphasis was placed on analyzing transients caused by wind gusts, load loss and other operationally or environmentally induced causes.

4.3 Available Technology Base

The helicopter industry has developed sophisticated techniques for the design and analysis of rotor systems for helicopters up to 34,000 kg (75,000 lb) gross weight, operating at speeds up to 90 m/sec (200 mph). These techniques, with modifications for the specifics of the WGS rotor operating regime, were applied throughout the design study. More important, however, is the large body of experience available in the helicopter industry. This experience has been gained through 30 years or more of technology development in rotor materials, fabrication processes and testing, as well as design and analysis. For this program, judgements based on this experience were applied extensively to ensure maximum attention was focused on the most critical technical and cost areas of the rotor.

Blade fabrication technologies available for the rotor include conventional metal spar extrusion/bonded afterbody, hot-formed metal spar/bonded afterbody and all composite material construction. Economic metal blade construction is limited by the maximum extrusion length of the spar, which is generally around 15 m (50 ft) for WGS rotors. Since WGS rotors optimize at considerably larger sizes, composite blades appear to be the most economic blade fabrication approach for this application.

Composite blades can be fabricated in lengths of 30 m (100 ft) or more, limited only by supporting mandrel deflections which can distort the blade airfoil or planform shape. The specific type of filament-sound composite construction selected for the preliminary design of the WGS rotor blade is currently being developed by Kaman and Hercules for the Army AH-1Q main rotor blade. This construction has also been used for numerous filament-wound rocket motor casings and offers the low cost, mass producibility required for the WGS application.

The lack of cyclic pitch control and rotor articulation requirements simplifies the blade root end and hub designs. Component weight is also a less important factor for the WGS rotor, allowing more conservative design of the inboard section of the blade and hub to assure attainment of 30 year life goals. These factors eliminate the need for sophisticated low weight designs and permit the use of low cost, readily available commercial materials, parts and fabrication processes. The larger hub and grip sizes also allow selection of high capacity commercial bearings, even for blade pitch bearings which are usually sophisticated high cost specialty items.

4.4 Concept Selections

A number of different rotor subsystem configurations and component approaches were evaluated prior to the selection of the configuration incorporated into the WGS preliminary designs. The options considered, and the results of the analyses to evaluate them, are summarized below.

4.4.1 Selection Criteria

Various selection criteria were employed in the rotor concept selection task, varying in scope and depth, depending upon the particular phase of the study. Initially, the rotor configurations examined in the conceptual design task were selected on the basis of minimum energy cost and lowest possible development risk. Energy costs were evaluated through the system analyses summarized in Section 3. Development risk assessments were made on the basis of mechanical simplicity, mass producibility and experience from similar rotor systems developed in the past.

The three rotor concepts designed in more detail in the conceptual design task were evaluated as part of the overall system analyses detailed in Section 3. The system analyses included the effects of rotor subsystem performance factors, such as number of blades, airfoil, blade twist and taper, etc. Each concept was designed to meet the 53.7 m/s (120 mph) maximum wind requirement and to have minimum structural weight (and hence, cost) consistent with acceptable natural frequencies to avoid operation at resonance points.

These same decision criteria, minimum weight and cost, were employed for the remainder of the study. For each option evaluation (for example, the choice between a hingeless and a teetering hub), a minimum weight design was evolved, meeting the maximum wind speed, blade fatigue life, blade natural frequency, stability, etc., and technical requirements of the rotor. The costs of the rotor subsystem options were then compared, along with other factors, such as reliability, maintainability and development risk, and a final selection was made. In the preliminary design task, after the rotor component concepts were established, the process of minimizing costs reduced to minimizing the structural weight required to meet these technical criteria.

4.4.2 Alternative Configurations

A number of variations for the rotor subsystem blades, controls and hub were examined during the course of the study. The options available for the WGS rotor are described below. Details of the specific analyses used to evaluate the more significant of these options are presented in the subsequent section.

4.4.2.1 Blades

Geometry and Number of Blades - Both two-bladed and three-bladed rotors were considered in the study. A three-bladed rotor offers lower vibratory load inputs to the rotor drive shaft, transmission, tower and other components. Its principal disadvantages include higher initial and energy costs, as discussed in paragraph 3.4.3.3. Blade shape, including airfoil section, planform and thickness taper, and twist can have a significant effect on rotor performance and, hence, overall system energy cost. If composite material fabrication processes are selected for the blades, little fabrication cost penalty is incurred for selecting twist, taper and airfoil sections to optimize blade performance and, hence, minimize energy cost. However, relatively simple airfoil shapes, moderate taper and linear blade twist yield near maximum efficiencies and reduce potential complexity in blade tooling and fabrication and assembly processes. Results of optimization studies on a number of blades and blade geometry were presented in Section 3. Additional analyses are presented in paragraph 4.4.3.2.

Materials - Several material and fabrication process combinations were considered for the WGS rotor blade.

Conventional metal rotor blades represent inexpensive and well-established technology. However, metal blades are not easily optimized for aerodynamic efficiency due to limitations on fabricability of twist, taper and thickness distributions. Also, for rotors of the scale necessary for the WGS, the rotor blade spar is envisioned as a heavy section extrusion, and current extrusion tooling and billet sizes preclude spar lengths in excess of 15 to 18 m (50 to 60 ft). If composite material fabrication processes are selected for the blades, little fabrication cost penalty is incurred for selecting twist, taper and airfoil sections to optimize blade performance and, hence, minimize energy cost. In particular, filament wound composite construction can be used to provide maximum structural efficiency for blade lengths well in excess of the optimum WGS rotor sizes by proper selection of filament winding sequences and tooling design. The lack of length limitations for composite blade fabrication processes is particularly important, since it obviates the need for blade joints which add to blade costs and require special joint/structure interface design to keep bending stresses well below levels acceptable in other portions of the blade. Blade joints also produce adverse dynamic effects from the added mass required for the joint structure which, for practical joint locations, lowers blade bending natural frequencies both in-plane and out-of-plane. Increased dynamic response of blade bending modes, particularly one-per-rev, causes higher vibratory blade bending moments and tower vibrations. Longer periods of operation at blade resonances during rotor start-up also occur, particularly for those modes excited by gravity forces.

In the conceptual design task, a constant section metal spar blade, as shown in Figure 4-1, was recommended for optimization. This conclusion was based on the conceptual design task sizing analysis for the high power system, which was rated at 1000 KW. In the optimization task, it became evident that the rotor should optimize at larger sizes. Since metal blade fabrication did not appear feasible above the 1000 KW level, emphasis was placed on the use of composite materials for the remainder of the study.

The initial composite material blade configuration was that shown in Figure 4-1, and consisted of a filament-wound spar with a bonded Nomex afterbody and fiberglass skin. It was subsequently determined that the cost of blades fabricated in this way was high, relative to other, more automated methods of fabrication. Hence, a fabrication technique utilizing an almost completely automated filament winding process for the blade spar and skin was adopted, with afterbody skin panel stiffening provided where required by blankets of minimum thickness honeycomb. This method of fabrication proved substantially lower in cost and was, therefore, adopted for the preliminary design configuration.

Blade Balance - Use of composite materials for the blade also permits achieving a close balance between the blade center of gravity (cg), feathering axis and aerodynamic center (ac) without the use of balance weights; this is the usual technique employed in helicopter blade construction. Blades with quarter chord ac/cg balance are traditional helicopter blade designs, evolved over years of experimentation in the development of helicopter rotor systems. This experience has shown that serious dynamic instabilities can occur with flexible blades and/or control systems if the blade center of gravity falls behind its aerodynamic center. For most airfoil sections, this is near the quarter chord. It is also desirable to locate the feathering axis near the cg/ac axis to minimize

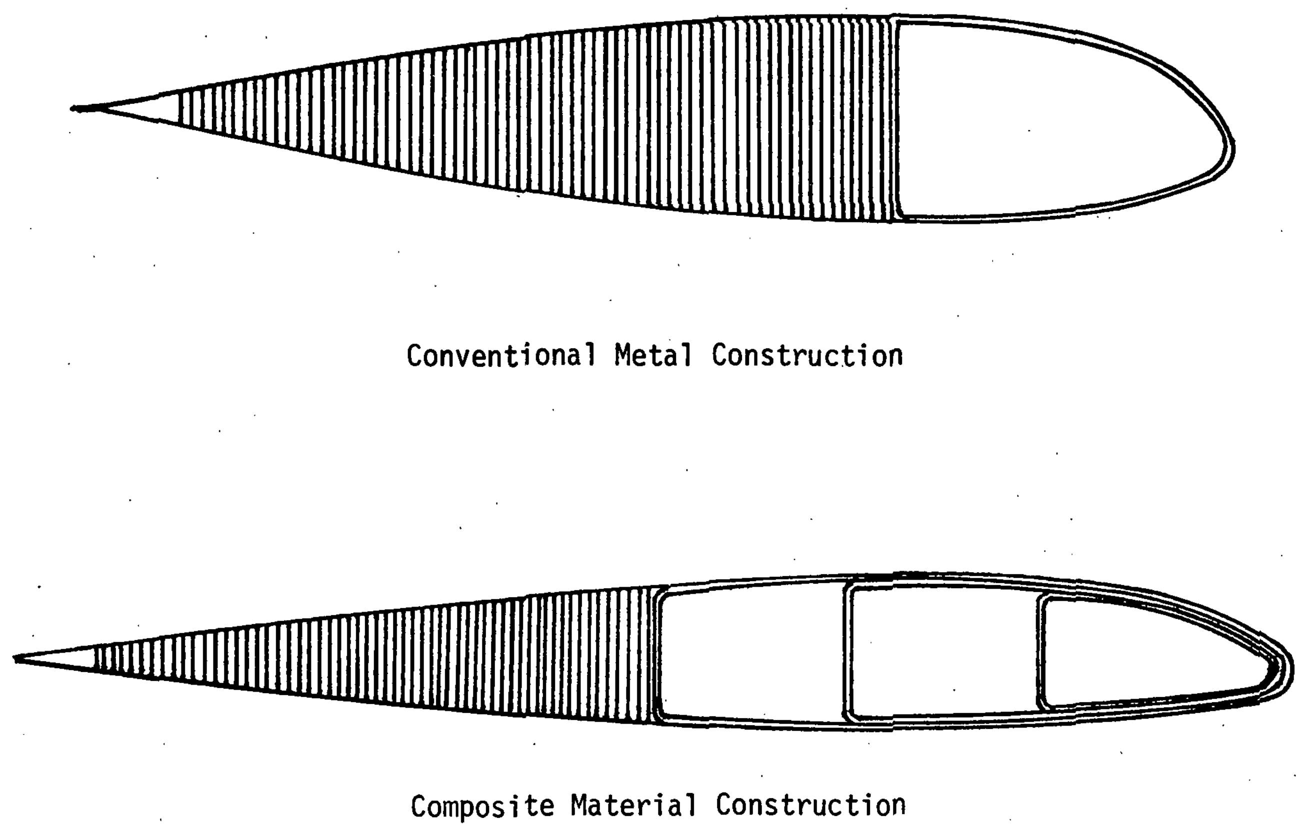


Figure 4-1. Rotor Blade Fabrication Options

eccentric feathering bearing loads, pitch control loads and intermodal dynamic coupling. The above considerations, when evaluated for the physical characteristics and operating conditions of WGS blades, using helicopter analysis methods, were found to apply directly; therefore, the ac/cg/feathering axis match is considered a design requirement for the WGS rotor. Automated filament wound fabrication with proper blade cross-sectional design permits quarter chord ac/cg balance, and this basic design feature for the rotor blade was carried throughout the study.

Blade Life - An analysis of energy cost vs blade life was conducted to determine if a cheaper blade having a shorter operating life is more cost effective than a more expensive blade having a long life. For WGS application, frequent replacement of large blades requires excessive maintenance man-hours and down time, costs of which are not offset by lower initial cost of shorter life blades. An analysis of optimum blade life is presented in paragraph 4.4.3.3 and shows that blade life should be maximized.

4.4.2.2 Controls

The selected WGS concept, which employs a synchronous AC generator, provides inherent rotor speed control when connected to the electric utility. The utility network provides, in effect, a synchronous speed governor for the rotor through the AC generator. To minimize excessive power fluctuations, however, some method of rotor torque control is necessary to regulate torque under varying steady and gust wind conditions, and to quickly change torque under various failure modes of both the WGS and the utility network.

As described in Section 3, both fixed pitch and variable pitch rotors were considered in the conceptual design task. Variable pitch rotors are controlled by rotating the blade into the wind as wind speed increases. The blade is mounted on bearings which permit the blade to rotate from the normal flat pitch operating position to the fully feathered high wind position (as in a propeller), thereby controlling torque. One possible advantage of the fixed pitch rotor lies in the elimination of pitch bearings and actuation systems to control blade pitch and, hence, rotor torque. Fixed pitch rotor systems also operate more efficiently than variable pitch, constant speed rotors and deliver power at lower wind speed conditions. However, some form of rotor torque control is necessary for fixed pitch rotors to limit torque and prevent overspeed, and the most feasible approach is to utilize a lift spoiler or drag flap on the rotor blade. In addition, blade loads are considerably higher for fixed pitch rotors at high wind speeds because of their flat pitch operating mode, as discussed later.

Results of the conceptual design task indicated that the complexity of torque and overspeed limiting of the fixed pitch rotor approaches, or exceeds, that of the variable pitch rotor, resulting in rotors of approximately the same cost.

In the start-up and shutdown modes, as discussed in detail later in this section, the problem of operating at variable rpm must be faced during each start-up and shutdown cycle. However, these are transient operations and the system controls can be designed to avoid sustained operation at critical resonance points. If, however, the rotor normally operates at variable speed in direct proportion to wind speed, significant time will be spent operating at potentially hazardous resonance conditions. This is illustrated in Figure 4-2, which presents a Campbell plot for the 1500 KW preliminary design WGS rotor. This plot shows the first flatwise (out-of-plane), first edgewise (in-plane) and second flatwise natural frequency lines for the rotor blade.

Typical of WGS rotors, and unlike most helicopter rotors, these natural frequency lines tend to be very flat. If a 20% margin from potential resonance lines at 2 per revolution and 3 per revolution for the first flatwise line is provided, the "safe" operating rpm range for the rotor is limited to the narrow region indicated in Figure 4-2 about the normal or design operating speed of the rotor. This also occurs for the other lines, becoming narrower at the higher harmonics. For a site with a wind speed distribution resulting in a substantial number of hours spanning a wide range of wind speeds, a fixed pitch variable speed rotor would have to operate near or at resonance points for the more critical lower harmonics. Therefore, some means of preventing resonance is required, either by adding damping provisions in the rotor or by operating the rotor in a reduced efficiency mode when the wind speed causes the rotor to approach a critical resonance point. It is possible to design a rotor with a sufficiently massive structure and high natural frequencies to insure operation at variable rpm does not produce significant response of the rotor. Variable speed rotors of this type which have been built and operated successfully in the past would be substantially more costly than the rotor configurations examined in this study and result in a higher system energy cost, even with their higher efficiency.

The technical and cost factors discussed above resulted in the selection of the constant speed, variable pitch rotor at the conclusion of the conceptual design task. The remaining program work then concentrated on more detailed design and analyses of this method of rotor control.

For the selected variable pitch rotor concept, blade pitch motion can be actuated either by directly driving the blade root mechanically, or through a servo flap near the tip of the blade using aerodynamic forces. Both of these techniques are well proven in the helicopter industry. In the conceptual design task, the servo flap approach for controlling blade pitch was adopted because it offers more stabilized control of the outboard portion of the blade. In particular, the servo flap provides rotor blade excursion stabilization under heavy gust conditions and when passing behind the WGS tower. Results of an analysis of these benefits are shown in paragraph 4.4.3.5. However, it was determined that the servo flap has insufficient blade pitch control forces during low wind conditions, particularly during starting. Secondary pitch positioning mechanisms would have to be provided at the blade root to orient the blades for starting, or an auxiliary starting motor could be provided to drive the rotor up to speed where the servo flap provides adequate control. Both of these approaches appeared to increase system cost and complexity and, as a result, direct pitch control was selected and kept through the preliminary design. A complete discussion of the starting problem is given in paragraph 4.5.4.

Direct mechanical control of blade pitch can be provided from the full feathered position to positive blade pitch angles. Pitch control to maximize starting torque at low wind conditions can also be readily programmed into the control

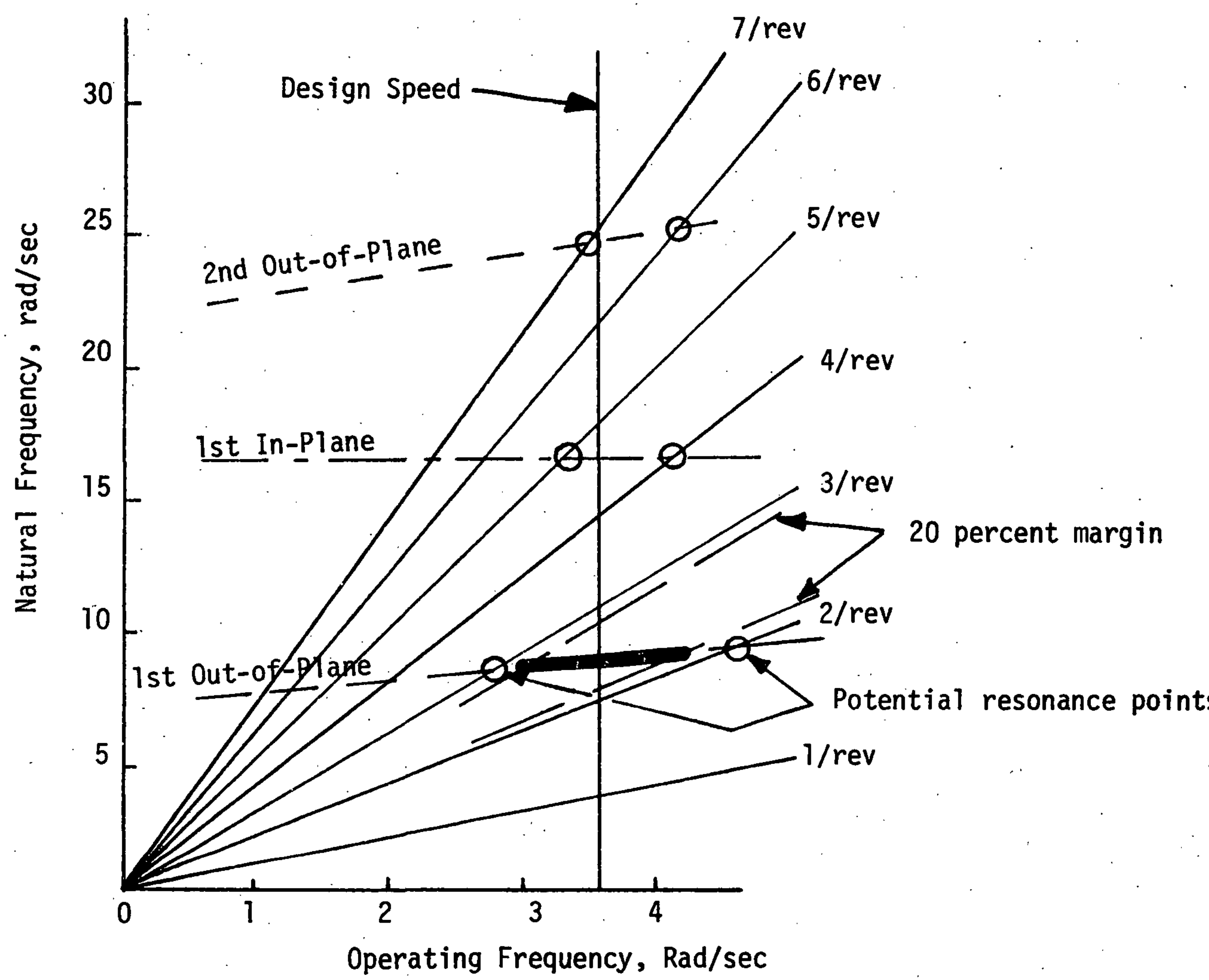


Figure 4-2. Variable Speed Rotor Operating Limits

system, and sufficient control power and rates can be selected to prevent excessive rotor overspeed for normal operating, wind gust and operational fault conditions. In addition, pitch control stiffness can be selected to achieve desired technical characteristics, such as de-coupling blade torsional responses from blade in-plane and out-of-plane responses.

4.4.2.3 Hub

A number of hub configurations are possible for the WGS rotor subsystem, from the simplest rigid, non-articulated concept to a fully articulated helicopter-type hub. Helicopter-type rotors with flapping or teetering hinges have lower bending moments out-of-plane at the blade root, but add considerable mechanical complexity to the hub. Large displacements of the blade tips during operation require greater offsets between the rotor head and tower to maintain adequate tower clearance when articulation out-of-plane is incorporated into the design.

Dynamic blade bending overshoot increases the risk of blade tip/tower intersection when the blades hit teetering or flapping stops during start-up and operations under gusty wind conditions. In addition, the lack of a strong centrifugal force field, due to the low operating rpm of WGS rotors, and low in-plane coriolis acceleration make the added complexity of lead-lag hinges often found in helicopters unnecessary. Because of these factors mentioned above, no articulation was included in the hub concept selected for the conceptual design task.

To minimize hub costs, a flex-plate type hingeless hub was adopted for both the conceptual design and parametric optimization tasks. The flex-plate type hub was envisioned as a slab of aluminum, machined to achieve moderate stiffness in the out-of-plane direction and high stiffness in the in-plane direction. Low out-of-plane stiffness provided bending moment relief under operating conditions, due to the natural coning effect of the rotor. The high in-plane stiffness provided the necessary characteristics to operate under relatively high in-plane loads caused by gravity. Figure 4-3 shows the concept used for the flex-plate hub.

As described in paragraph 4.4.3.4, a subsequent study examined the possibility of using a teetered hub to lower blade bending moments and possibly realize savings in blade weight and cost. It was determined that savings would be very small and not offset the additional cost and complexity incurred in the hub caused by the teetering provision. However, in the preliminary design task, it was found that the flex-plate out-of-plane blade natural frequencies were too close to the operating rotor speed range, thereby increasing vibratory response of the out-of-plane bending moments. This problem, similar to that previously described for the variable rotor speed configuration, led to the adoption of a rigid, hingeless rotor hub configuration to achieve the necessary root end stiffness characteristics. A detailed description of this hub concept and its characteristics is given in paragraph 4.5.

4.4.3 Configuration Analyses

The detailed rotor subsystem configuration analyses associated with concept selection are presented below. These analyses cover several aspects of the

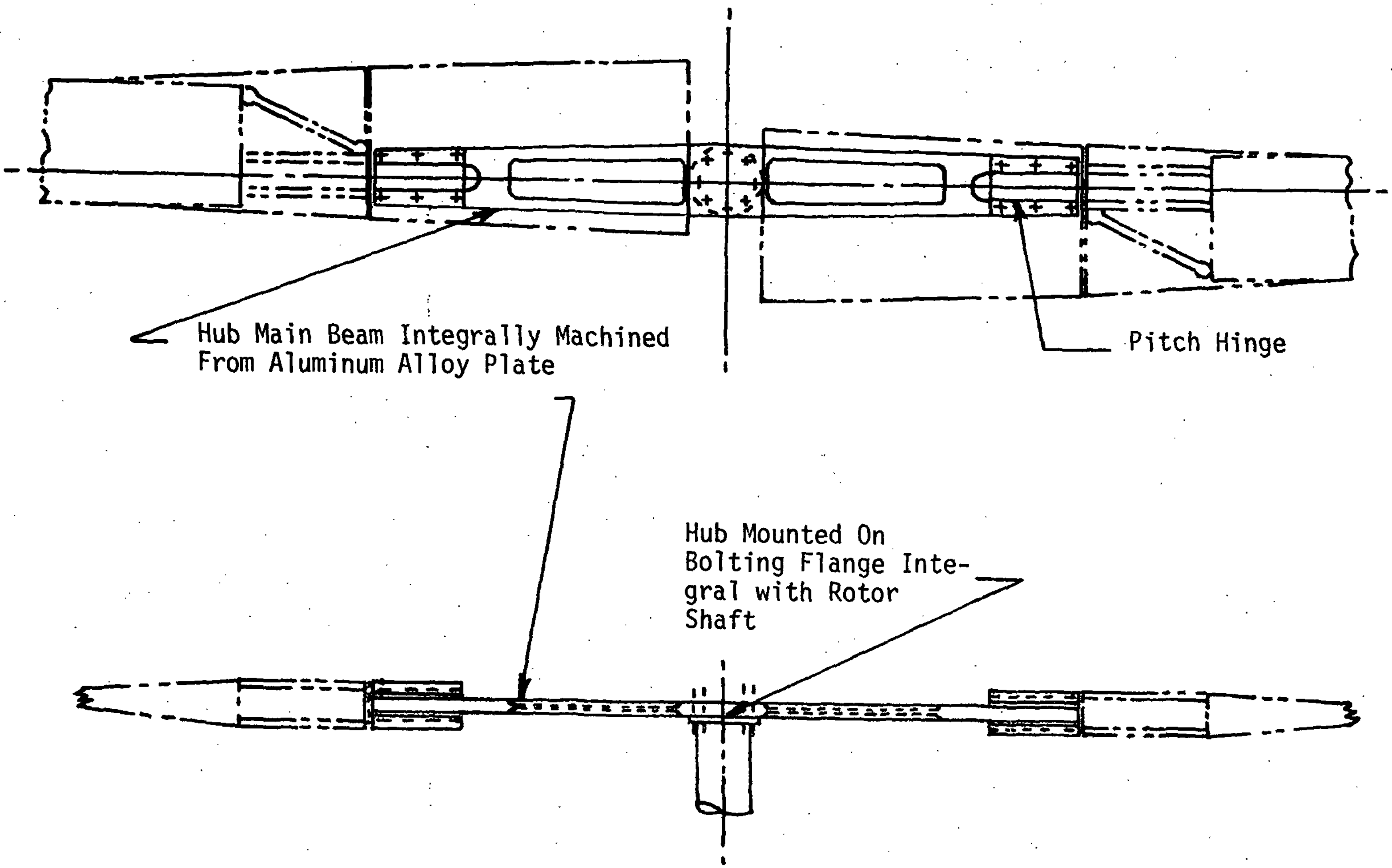


Figure 4-3. Flex-Plate Hub Concept.

rotor subsystem design and often have multiple design and operational facets which affected the final selections. The studies were conducted at various points in the program and, hence, the system concepts and quantitative results presented are often for slightly different systems than those finally selected for preliminary design. However, results of these conceptual design analyses are not limited to the specific system configurations and sizes used; the conclusions reached apply to concepts selected for preliminary design. The specific preliminary design presented in paragraph 4.5 was analyzed.

4.4.3.1 Rotor Size and Solidity Limits

The conceptual design task used a 1000 KW rated WGS as the reference high power system. This resulted in a rotor diameter less than 45 m (150 ft). The subsequent parametric optimization analysis indicated that minimizing rotor solidity (ratio of blade area to rotor disc area) minimized system energy cost and had a significant effect on the rotor diameter/solidity relationship for an optimum sized WGS. Larger rotor diameters and lower solidities appeared attractive and the question of limits on maximum diameter and minimum solidity was raised.

To determine the upper limits on rotor diameter and the lower limit on rotor solidity, a parametric blade design study was conducted. This study was for the composite rotor blade and flex-plate hub configurations used in the optimization task. The study utilized the optimum blade shape identified by the parametric analysis (airfoil section, thickness and planform taper, and twist). Blade designs for rotors with diameters of 45.7 m (150 ft), 61 m (200 ft) and 76.2 m (250 ft), were prepared for rotor solidities of 0.02, 0.035 and 0.0475. Mass and structural property distributions were then estimated for the blades, and blade moments and loads were calculated utilizing a standard helicopter rotor load program, modified to account for the effects of gravity, wind shear and tower shadow. Blade bending natural frequencies and mode shapes, and blade stability were also calculated using standard helicopter computer programs.

Results of these analyses are shown in Figures 4-4 through 4-11. Figures 4-4 and 4-5 present the vibratory out-of-plane and in-plane bending moment distributions, respectively, for the three rotor diameters and solidities. Although these bending moments appear high by helicopter standards, the stress levels in the blade structure are low and completely acceptable. Figure 4-6 through 4-8 show blade natural frequency plots for the rotors analyzed. The significant first and second flatwise (out-of-plane) and first edgewise (in-plane) natural frequency lines are all acceptable, either as is or, in a refined design, with minor adjustments of blade structure.

The torsional divergence boundaries for the rotors are shown in Figures 4-9 through 4-11. As indicated, the high torsional stiffness inherent in large WGS blades puts the rotor well into a stable operating regime.

After determining the technical acceptability of the rotors examined, the rotor blade weight and cost model in the parametric computer program were modified, as discussed in Section 3. This led to the subsequent optimization of both the high power and low power system rotors in the 45 to 55 m (150 to 180 ft) diameter range, well within the acceptable diameter limitations identified in this analysis. The optimization study also indicated that rotor solidity should be

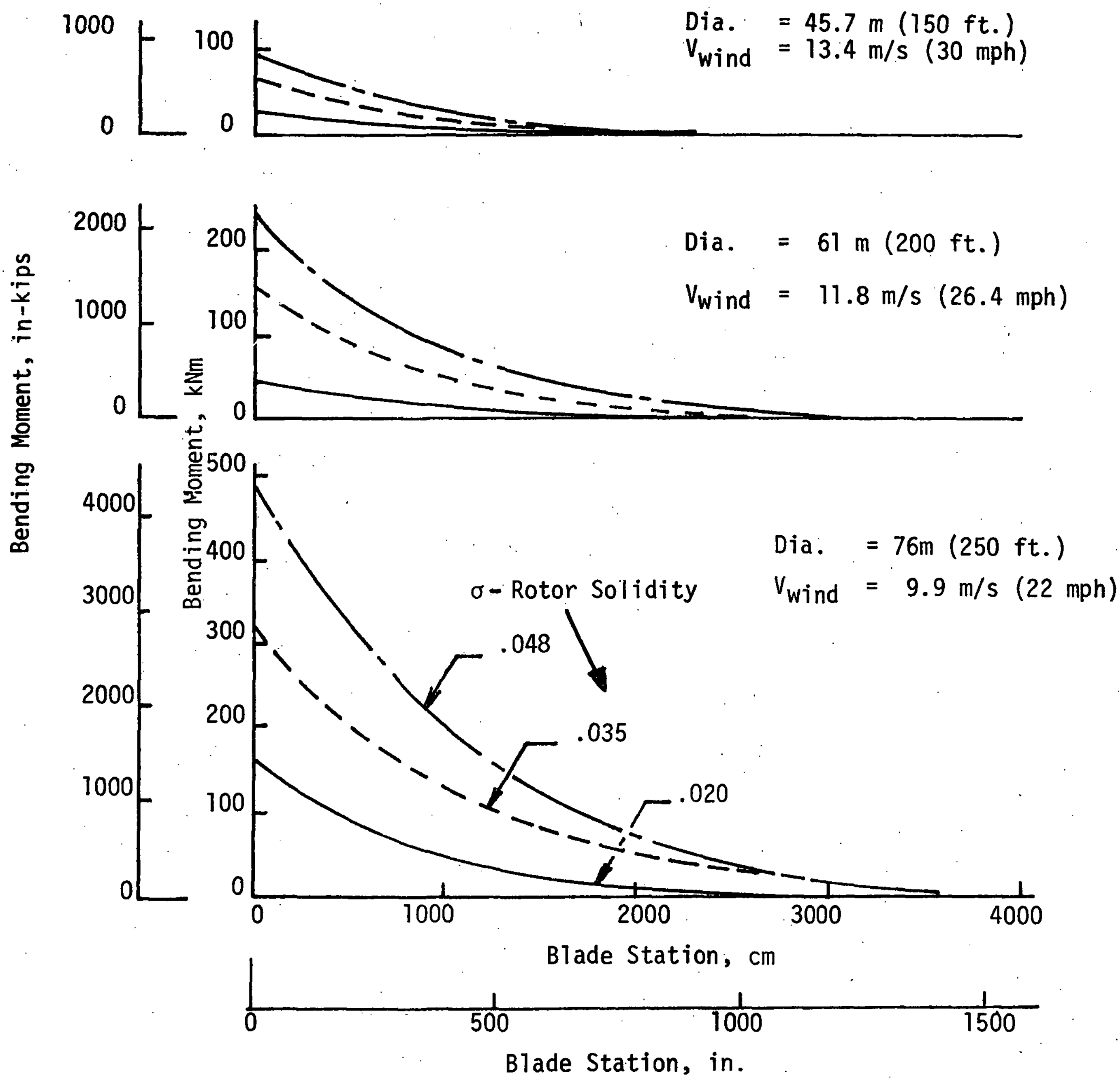


Figure 4-4. Vibratory Out-of-Plane Bending Moments

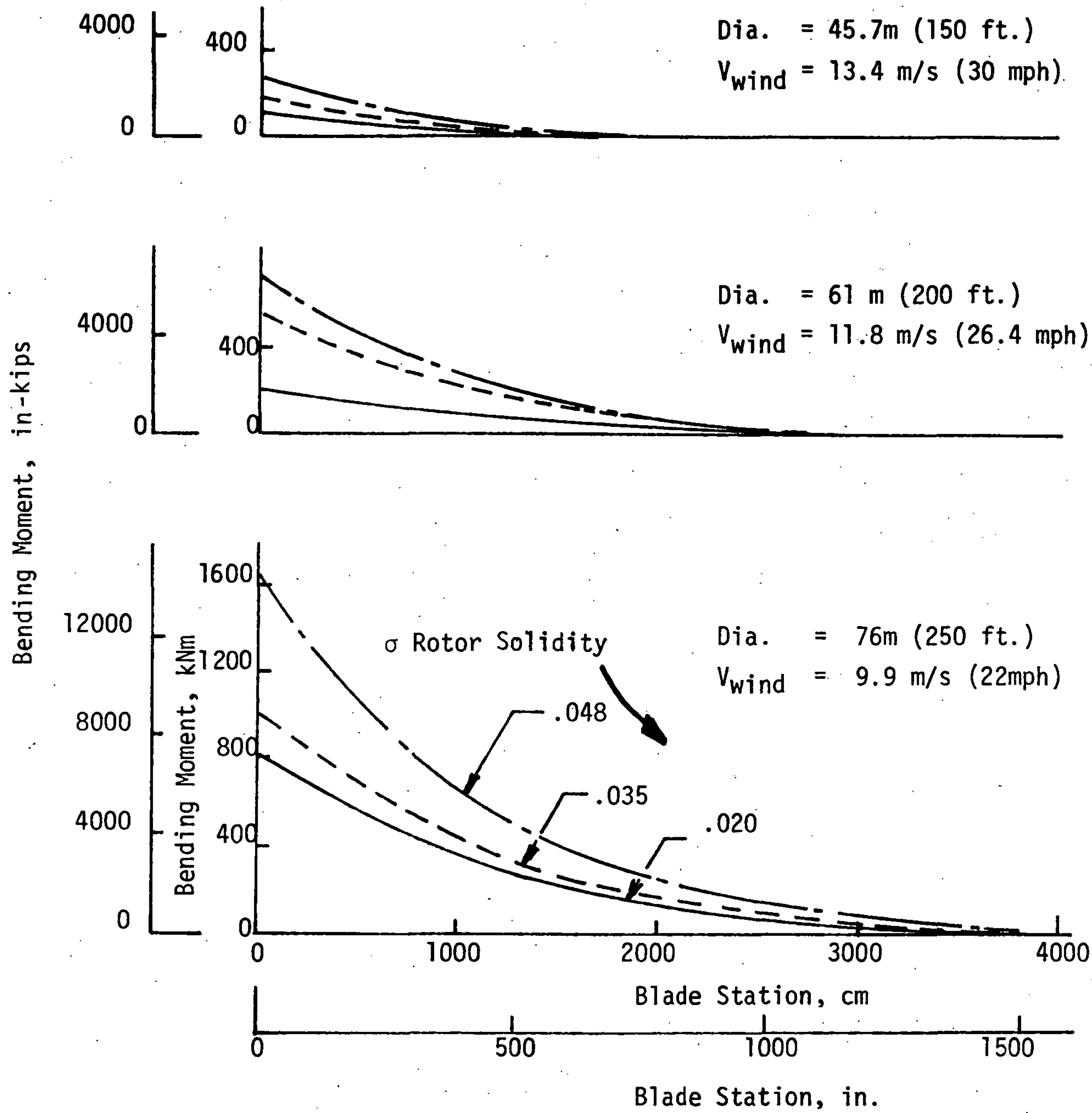


Figure 4-5. Vibratory In-plane Bending Moments

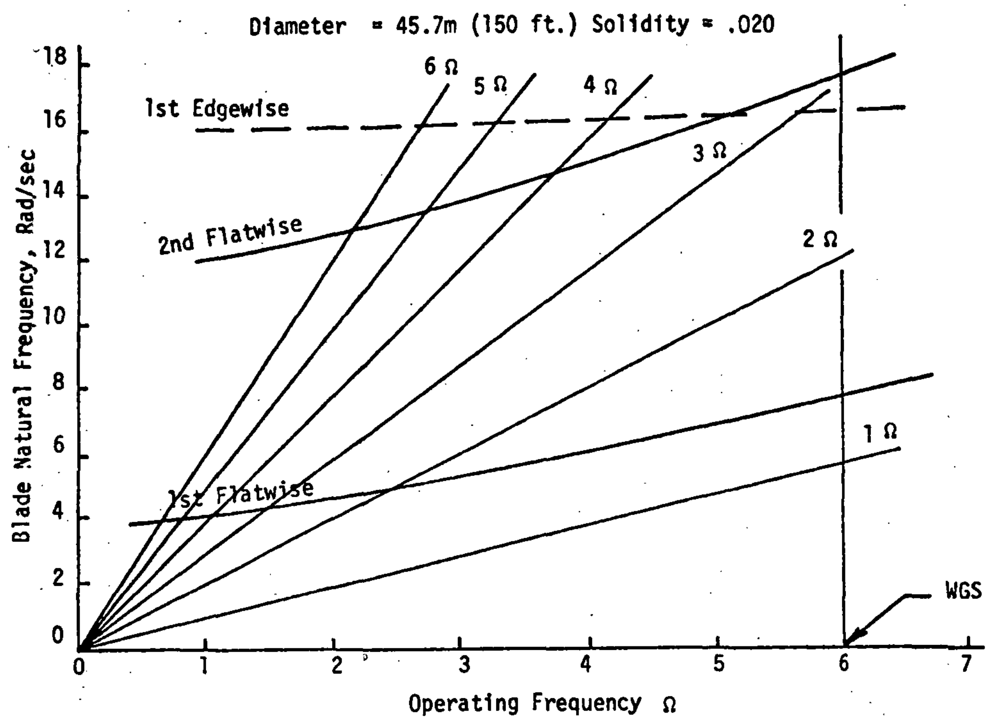


Figure 4-6a. Blade Natural Bending Frequencies

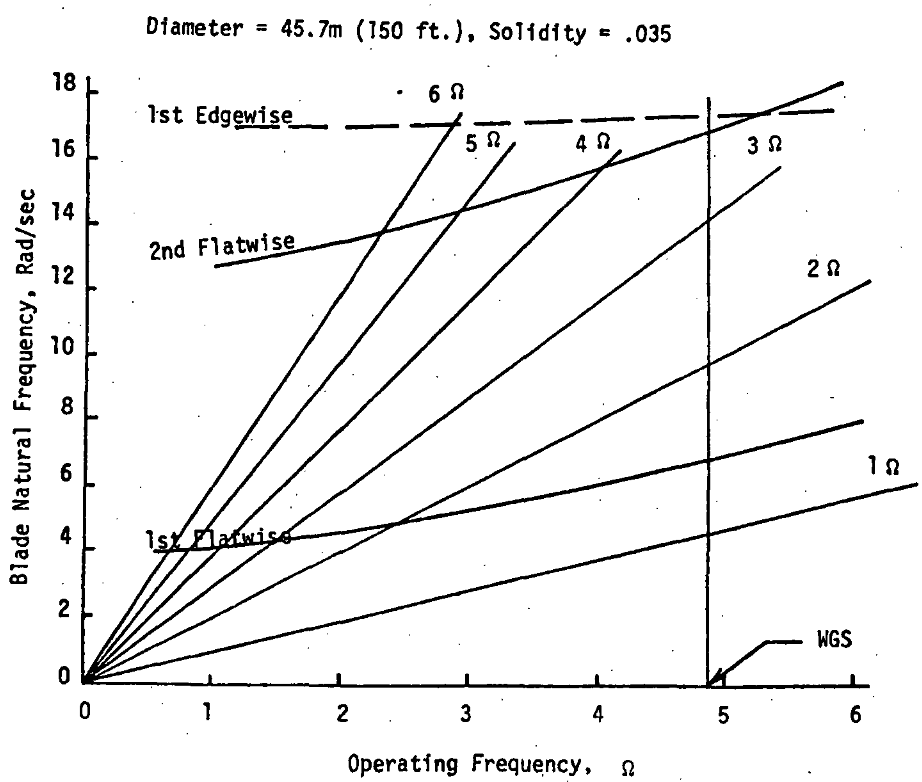


Figure 4-6b. Blade Natural Bending Frequencies

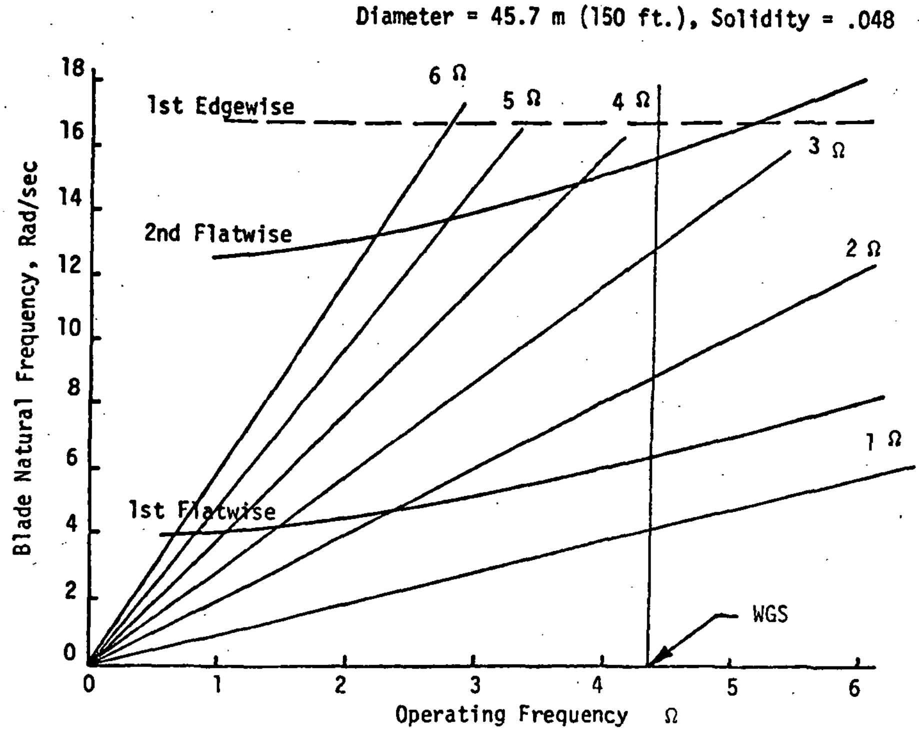


Figure 4-6c. Blade Natural Bending Frequencies

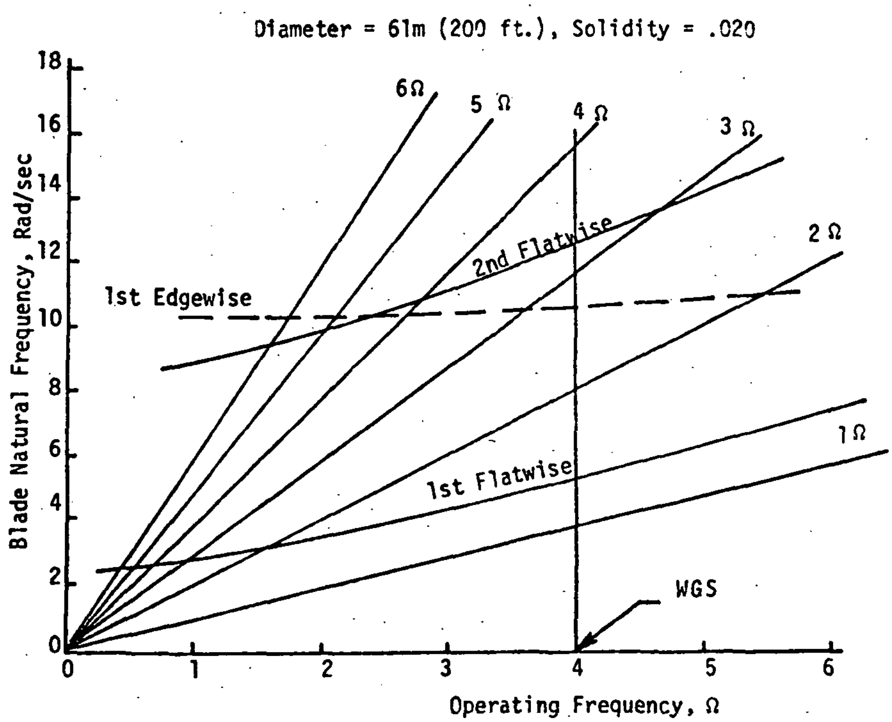


Figure 4-7a. Blade Natural Bending Frequencies

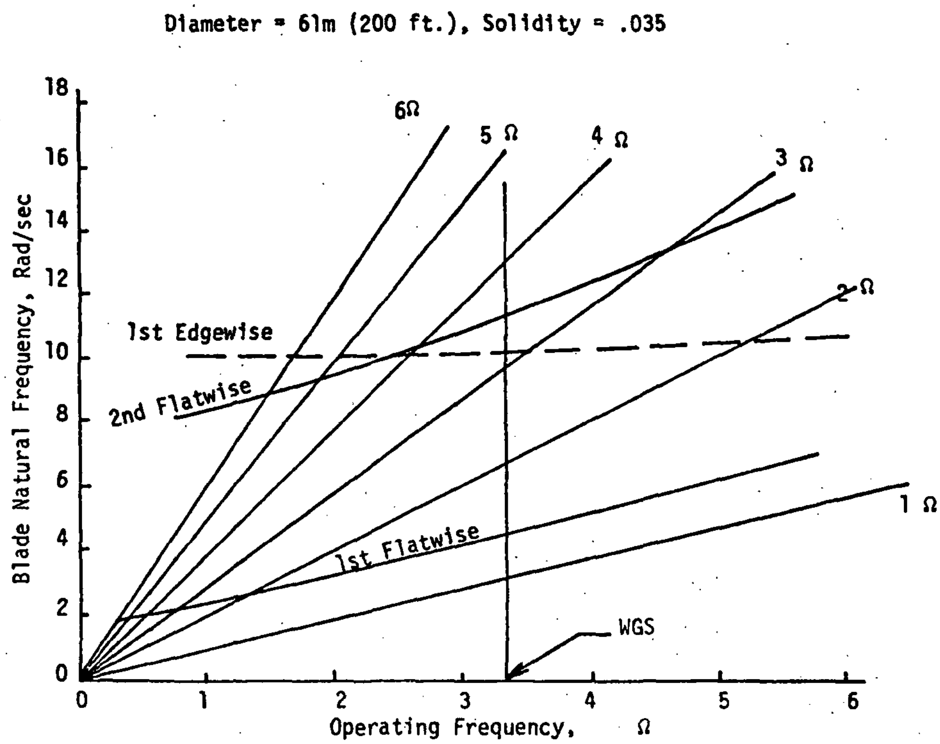


Figure 4-7b. Blade Natural Bending Frequencies

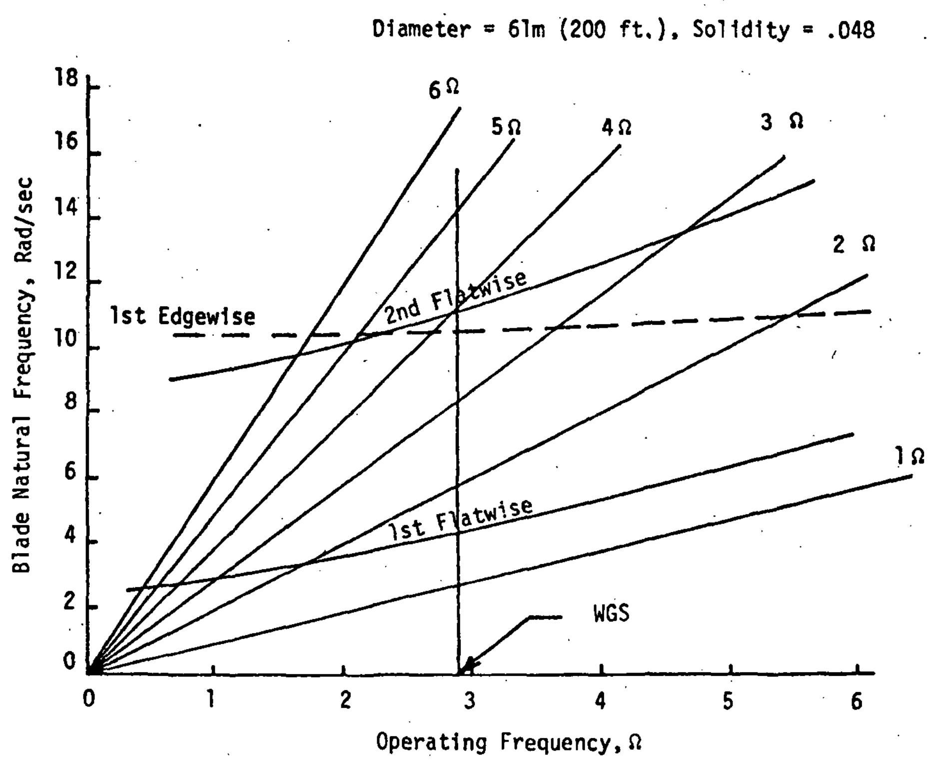


Figure 4-7c. Blade Natural Bending Frequencies

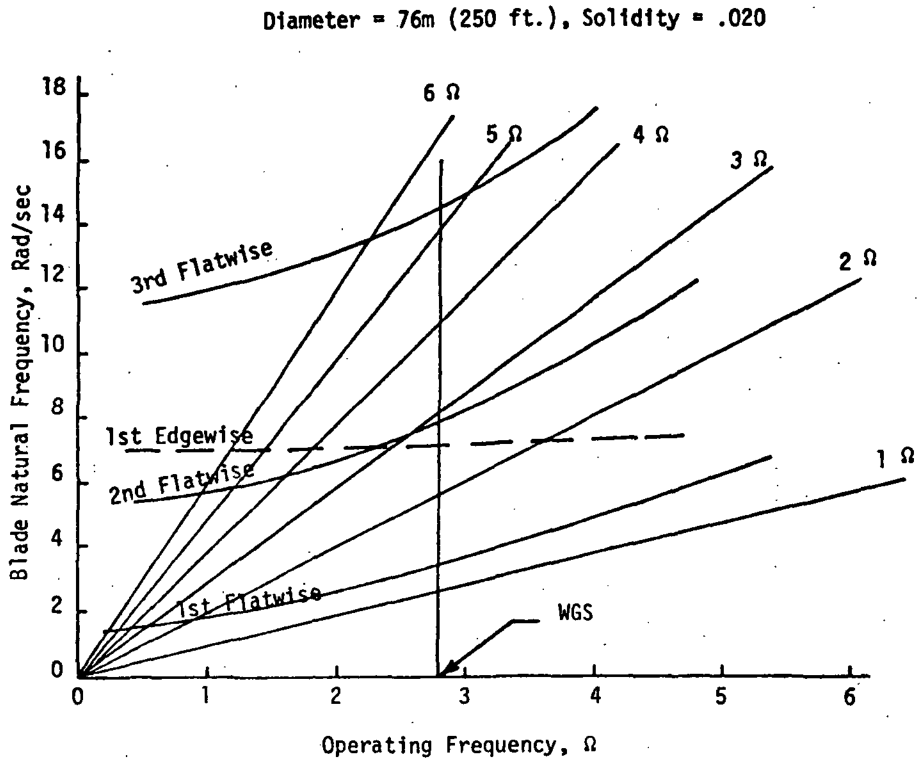


Figure 4-8. Blade Natural Bending Frequencies

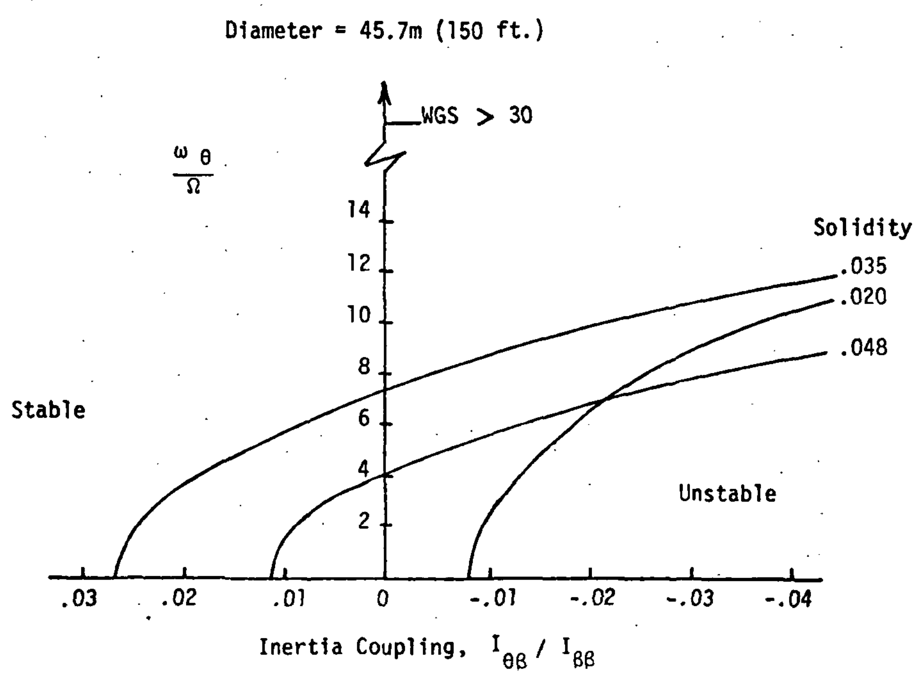


Figure 4-9. Torsional Divergence Boundaries

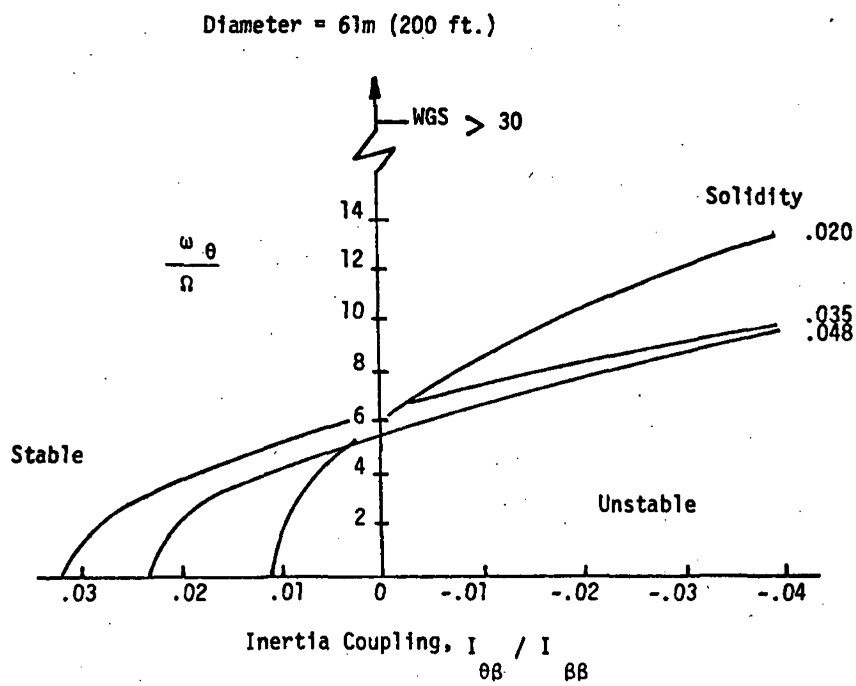


Figure 4-10. Torsional Divergence Boundaries

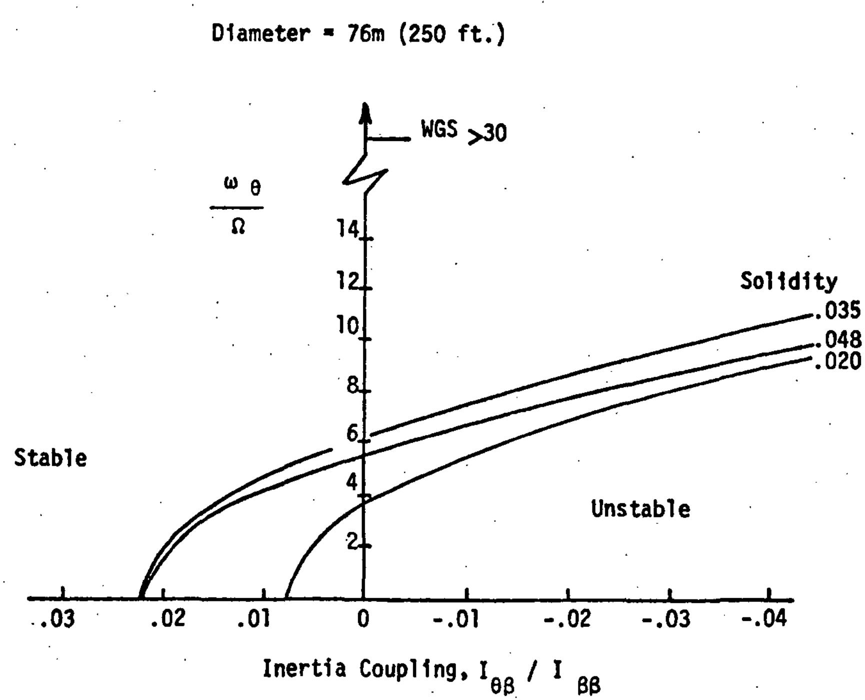


Figure 4-11. Torsional Divergence Boundaries

minimized, but that solidities below .03 did not offer any significant energy cost savings. Since the highest possible rotor solidity is desired for starting operations, a rotor solidity of 0.03 was selected, yielding near minimum rotor costs, with acceptable starting characteristics and rotor development risk. This rotor solidity was carried through the preliminary design, and is recommended for both the low power and high power WGS designs.

4.4.3.2 Blade Geometry

After the selection of composite fabrication technology for the rotor blades was made, the flexibility of this process permitted an optimization of the blade planform and the thickness taper, and the blade twist distribution, as well as the choice of airfoil section. Since most of the feasible options available for the blade shape can be accomplished with minimal blade cost impact, the principal blade geometry effect on overall energy cost is due to rotor performance. The rotor performance calculation procedure, an integral part of the parametric computer program, was used to investigate the various geometric factors affecting rotor performance. Results of these analyses were summarized in Table 3-12 in Section 3.

Two additional blade geometry studies were conducted after the final rotor solidity and diameter were selected for the preliminary designs. These studies covered both planform taper and blade twist distributions. Planform taper has a significant effect on blade weight and cost, since it influences both the aerodynamic performance of the blade and its mass and structural property distributions. To examine this effect, a number of blade planforms were examined on a rotor performance basis, with the results shown in Figure 4-12. This figure shows overall rotor aerodynamic efficiency as a function of blade taper ratio, for both full span taper and mid-span to tip taper. The mid-span to tip taper configuration, which employs a constant chord blade from the root end attachment at the hub to the mid-span of the blade and then the selected taper from the mid-span to the tip, has lower structural weight than full span taper blades because it eliminates excess material inboard where the aerodynamic benefit of a large chord is not significant. As shown in Figure 4-12, employing the mid-span to tip option only sacrifices a fraction of a percent in rotor aerodynamic efficiency.

Figure 4-12 also shows that taper ratios between 1:1 and 3:1 do not have a significant effect on aerodynamic efficiency. However, the effect of planform taper is much more significant for blade structural and tuning reasons, and was qualitatively judged to have a substantial impact on costs. As a result, the 3:1, mid-span to tip taper configuration was selected for the preliminary design as the most economic approach for the blade planform.

Although composite fabrication processes employing automated filament winding techniques can produce a rotor blade with almost any desired twist distribution, the complexity of the blade tooling and difficulties that might be encountered in removing the mandrels from the fabricated blade were recognized. The potential mandrel removal problem can be easily solved if linear or near linear twist distributions are used. The penalty of using a linear twist instead of the ideal twist recommended is shown in Figure 4-13. These data indicate that a linear blade twist, properly selected, can yield a blade design with performance

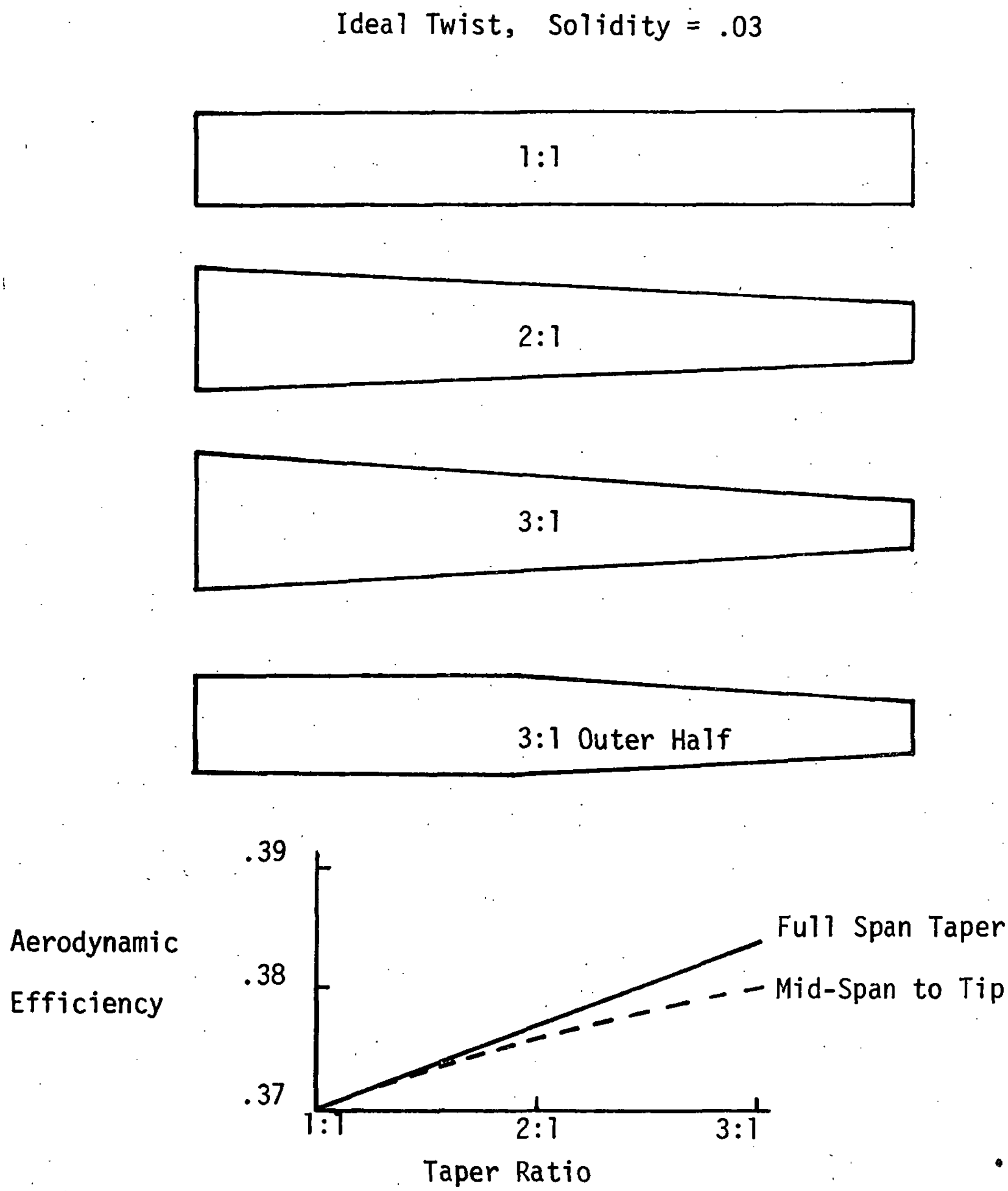


Figure 4-12. Planform Taper Investigation

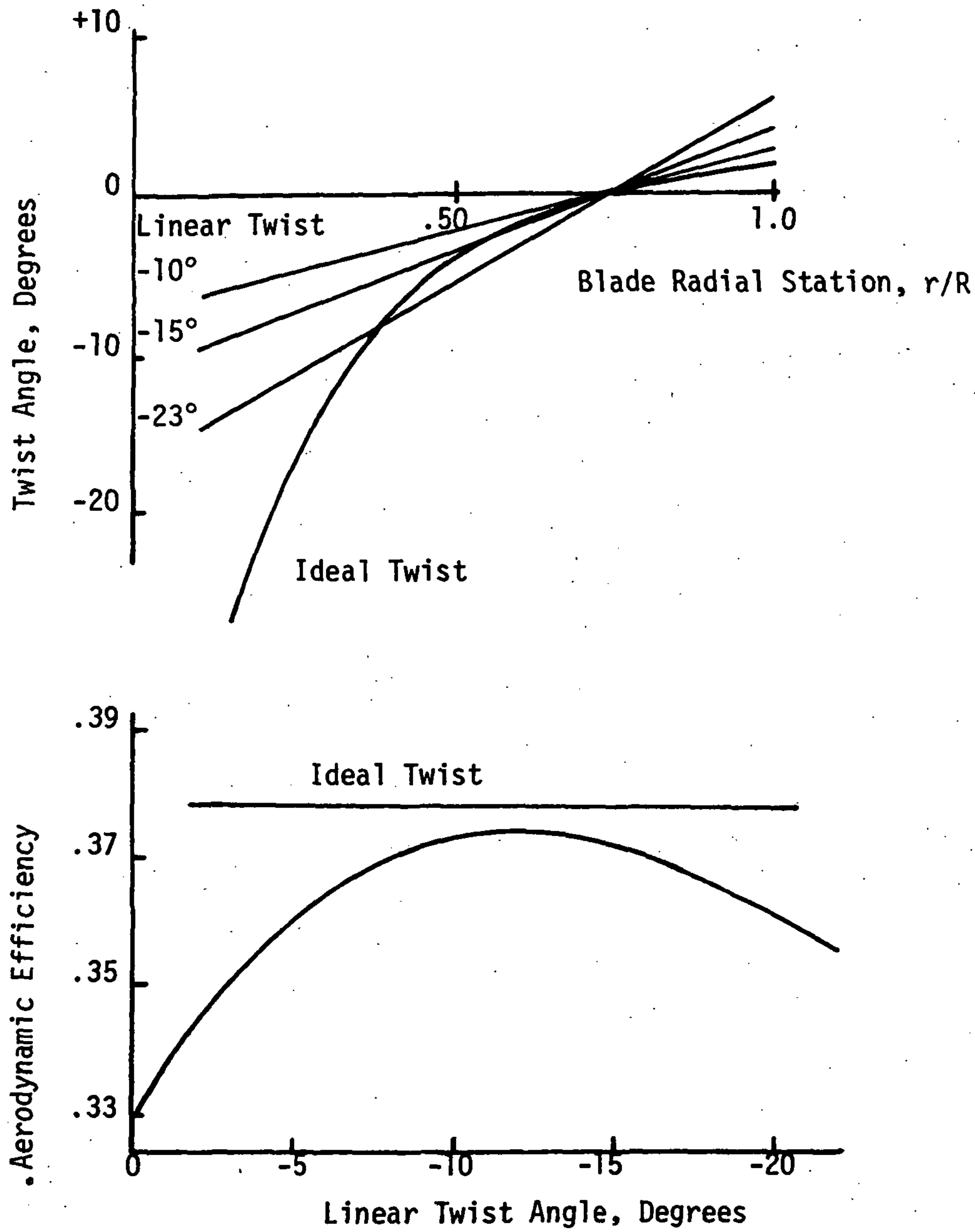


Figure 4-13. Blade Twist Distribution

approaching that of a blade with ideal twist. As shown in Figure 4-13, a linear twist of approximately 10 to 12 degrees results in a rotor efficiency less than one-half of one percent lower than that for an ideally twisted blade. Hence, if it is required to use a linear blade twist in lieu of the optimum twist distribution for tooling fabrication and mandrel withdrawal purposes, there will be no significant effect on overall system performance.

4.4.3.3 Blade Life

An important factor in rotor system design is the selection of the most economic blade life for the particular application. The tradeoff between blade life and initial cost is usually conducted on a total life cycle cost basis, where the cost of maintenance and replacement of the blade at periodic intervals is traded off against the unit cost for the blades. Although the Statement of Work for the study contract specified a 30 year life for all system dynamic components, blade life was a primary system parameter which warranted an investigation to determine the most economic blade life.

Due to the complexity of the analysis necessary to optimize blade life, this evaluation was conducted in parallel with the parametric computer program optimization studies. The analysis used the blade design evolved for the high power WGS in the conceptual design task. This blade design had been prepared to meet the 54 m/sec (120 mph) maximum wind condition and had a life slightly over 30 years under assumed loading and wind frequency distributions. With this design as a base, the blade was resized for several different maximum wind speeds. Using the representative blade loading profile and wind frequency distribution as functions of wind speed, the fatigue life of each blade was calculated. In this calculation, the number of start-up and shutdown cycles experienced each day of operation was included to take into account the high loading variations experienced by the blade during each cycle.

Results of the analysis are shown in Figure 4-14, which presents blade weight for the high power WGS conceptual design as a function of blade fatigue life and corresponding maximum wind speed structural limit. As shown, a blade design which can withstand the 54 m/s (120 mph) maximum wind condition has a life of approximately 30 years when the machine averages four start and stop cycles per day. Also illustrated is the relative rapid drop in blade fatigue life as its maximum wind speed capability is reduced, with relatively small savings in blade weight.

A system cost analysis based on these results was conducted to determine the effect of blade life on overall system capital and energy costs. This analysis examined changes in blade initial cost, hub cost, blade replacement cost, and operations and maintenance costs as blade life was varied. The analysis process started by selecting a blade life and calculating the blade initial cost, based on the blade cost parametric relationships given in Section 3. Since the hub cost is also a function of blade weight, hub cost also was calculated to account for lower blade weights as blade life was reduced. Assuming a system life of 30 years, the blade replacement cost was calculated on a linear pro-rated basis; that is, total blade cost for the life of the system would be 150% of the initial blade cost if blade life were only 20 years, rather than the 30 year system life. Operations and maintenance costs were also calculated to account for the lower cost of maintaining lighter, lower life blades.

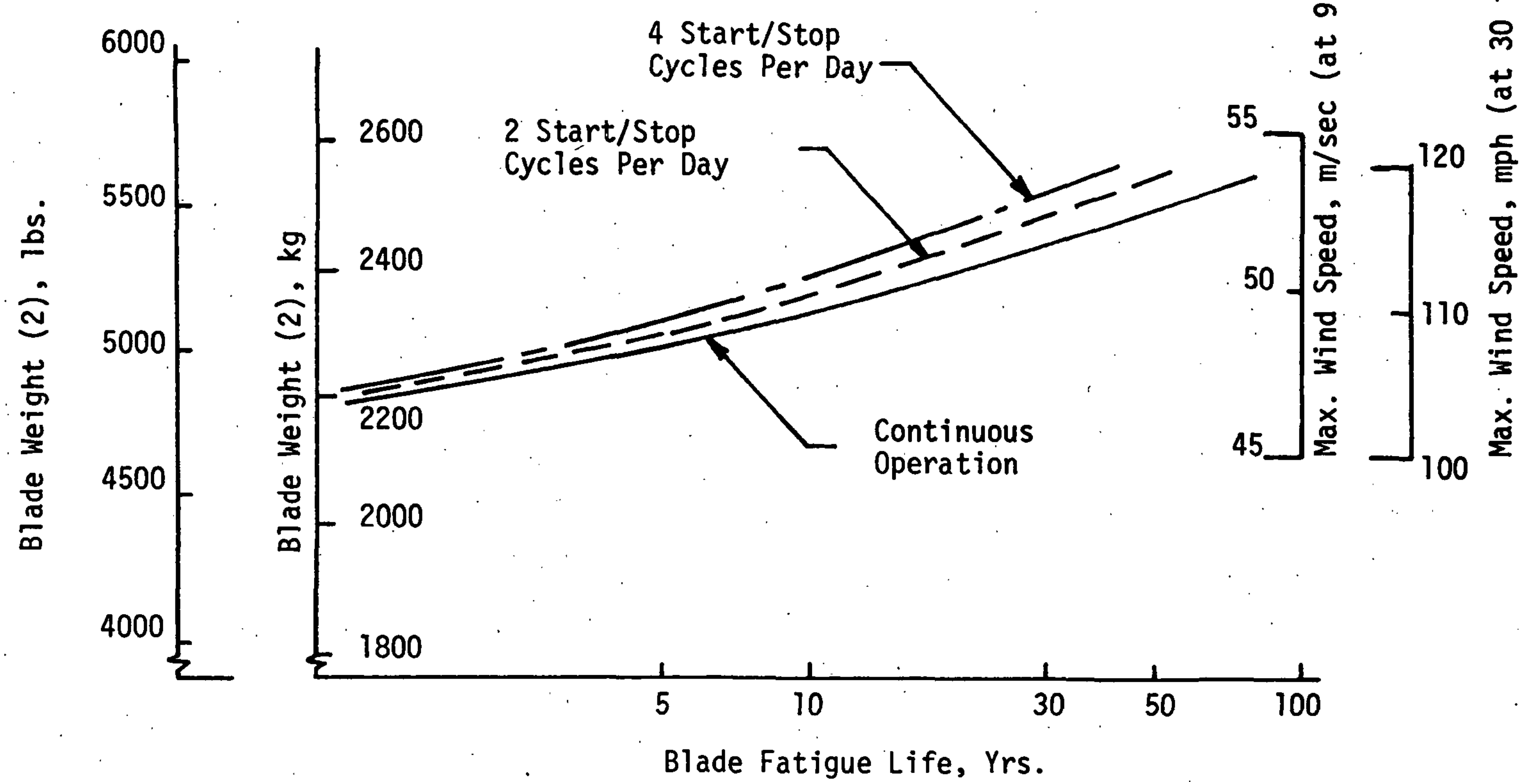


Figure 4-14 Blade Life/Maximum Wind Analysis

The cost analysis data are shown in Figure 4-15. These data show that there is no incentive in capital or energy costs to design blades for less than the system life of 30 years. It also appears that a blade life greater than 30 years would yield capital and energy cost savings. However, this saving is small and a more detailed study would be required to determine the exact blade life appropriate for the WGS unit.

4.4.3.4 Teetered vs Hingeless Hub

Although the flex-plate hub concept selected in the conceptual design task offers blade root bending moment relief, provision of some form of articulation to the rotor could further reduce the bending moments and lead to blade weight and cost savings. Therefore, a teetering hub configuration was examined to determine if this concept offered any cost advantages over the flex-plate hub.

The quantitative portion of the analysis utilized a single degree of freedom rotor model with teetering simulated by a zero offset articulated rotor. The odd harmonics represent the unique teetering response, while the even harmonics are identical for both the hingeless and teetered hubs. For the analysis, the same blade EI and mass distributions were used for the blades on both the teetering and hingeless hubs, and the computer program utilized to calculate blade loads included the effects of in-plane gravity loads, wind shear and tower shadow. The results of the analysis are presented in Figures 4-16 and 4-17, which show the total vibratory bending moments, both out-of-plane and in-plane for the hingeless and teetering hubs.

Tower shadow was represented by the method reported in Reference 4-1: the wind velocity in the plane of the rotor disc, downwind of the tower, was reduced by a cosine-squared function to a velocity 30% below free stream wind velocity, within a 30 degree rotor azimuth angle (15° on either side of the tower centerline). This representation is considered adequate for the single degree of freedom analysis of vibratory blade bending moments. It is unlikely that fatigue lives of the WGS blades would be affected by stronger tower shadow effects, because of the substantial margin between the calculated vibratory levels and the blade endurance limit (see paragraph 4.5.2.7). (See Tower Shadow note, bottom of page 4-111.)

The data in these figures show that the teetering hub substantially reduces out-of-plane vibratory bending moments below those for the hingeless hub. In-plane bending moments, which are generally higher than the out-of-plane moments, are not substantially affected by the use of the teetered hub. These results suggest that the teetered hub might offer some saving in blade weight and, hence, cost, if the blade design requirements are dominated by the out-of-plane moments. However, the out-of-plane static bending moments at the 54 m/s (120 mph) maximum wind condition are comparable for both hub configurations and are the critical bending moments driving the blade design. The blade structure which meets the static maximum wind requirement results in low, non-critical fatigue stress levels under normal operating conditions, even for the higher bending moments imposed by the hingeless hub configuration. Therefore, the lower out-of-plane bending moment advantage of the teetered hub is not a significant factor for the blade and no reduction in blade cost can be realized by employing a teetered

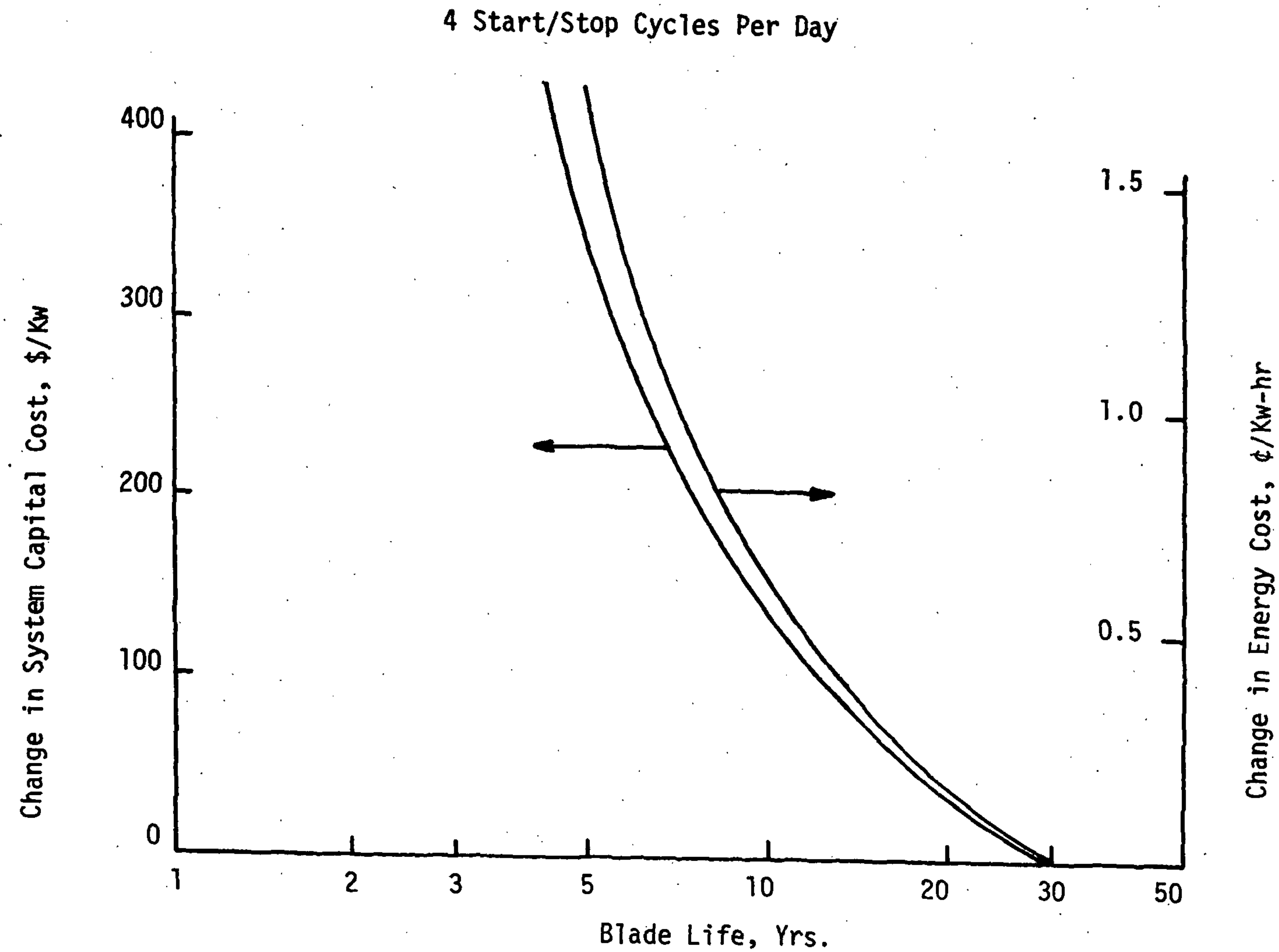


Figure 4-15. Blade Life/Cost Tradeoff Analysis

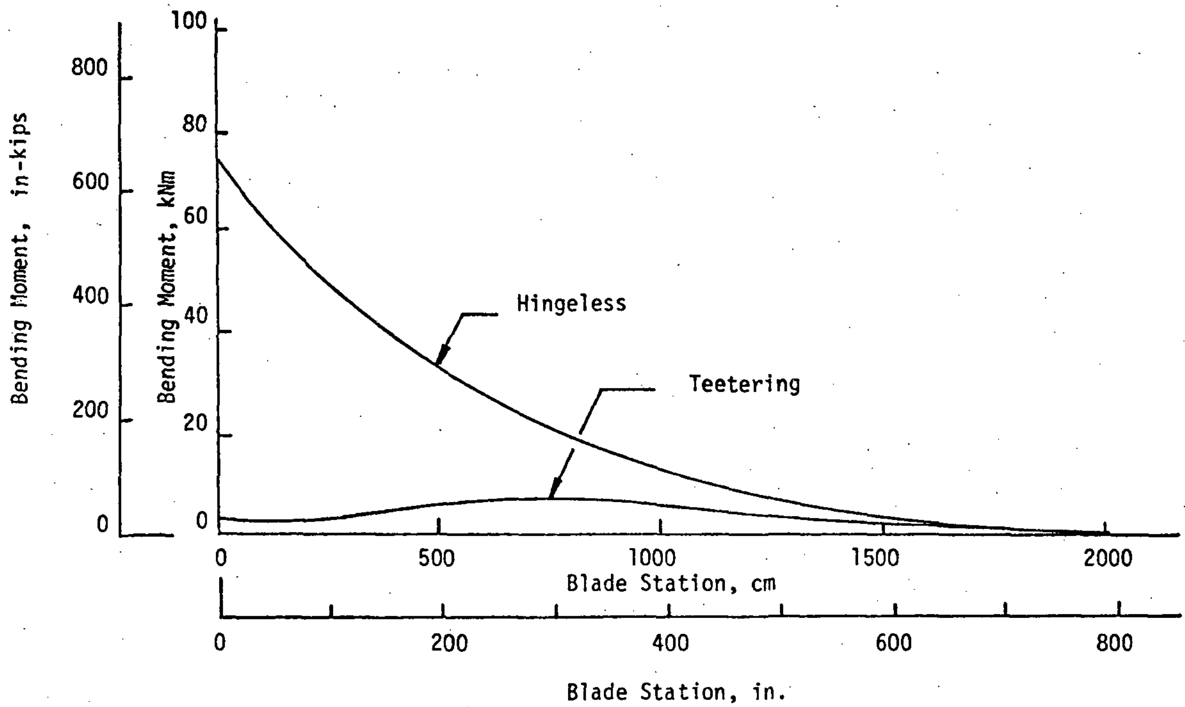


Figure 4-16. Total Vibratory Out of Plane Bending Moments

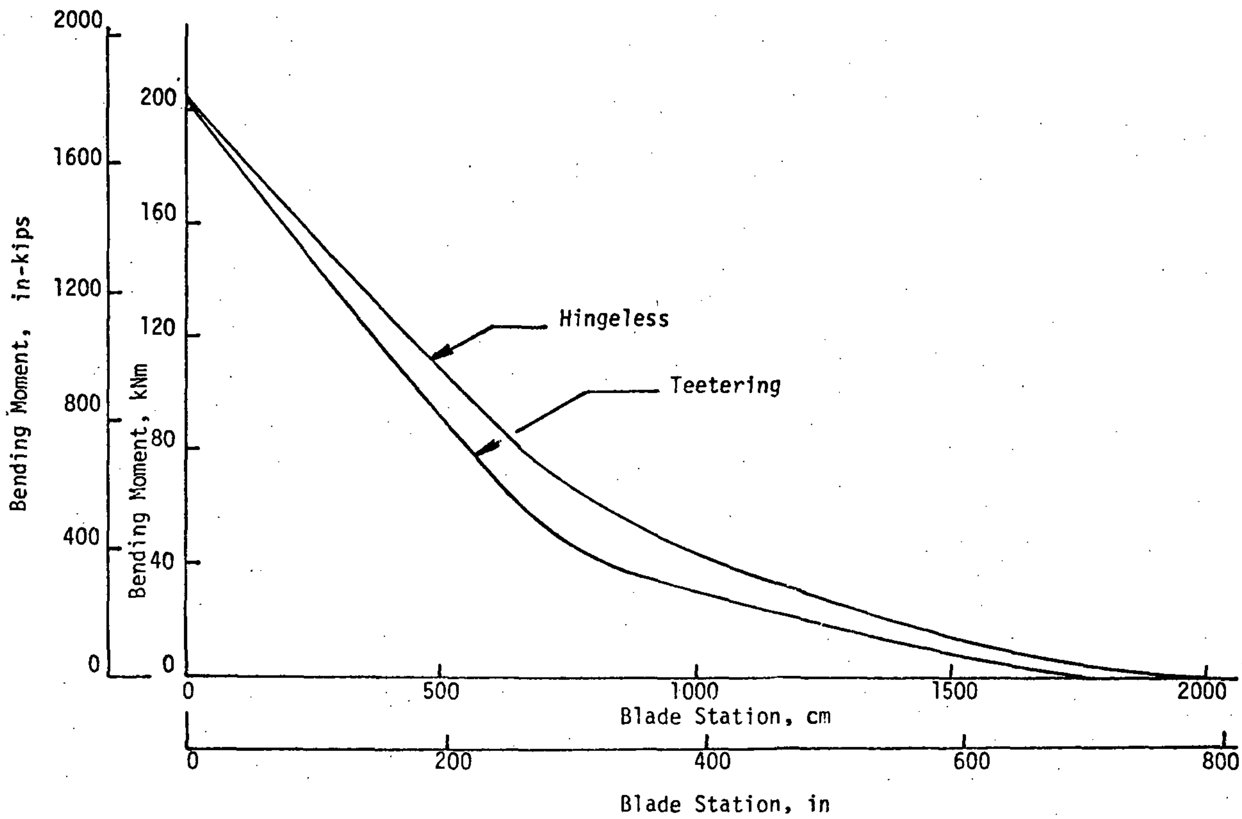


Figure 4-17. Total Vibratory In-Plane Bending Moments

hub configuration. When the additional complexity of the teetering hub was considered, this type of hub was dropped from further consideration.

Subsequent analyses conducted in the preliminary design task of the study strengthened this conclusion. In these analyses, blade stiffness distributions needed to tune blade natural frequencies imposed structural requirements on the blade which are more severe than those imposed by the hingeless rotor vibratory bending moments. With blade sections sized for stiffness or high wind conditions, fatigue stress levels on the hingeless rotor are sufficiently low to achieve the required 30 year life goal for the blades and associated hub components.

In addition to the lack of lower blade costs for a teetered WGS rotor hub, the greater cost and complexity of the teetered hub further argues against its utilization. The mechanical complexity of the teetering hinge itself, and the associated additional maintenance and potential failure modes, are further complicated by the need to articulate the blade pitch controls to accommodate the teetering motion, and the addition of gust locks and teeter stops to limit blade motion during operation. This increases tower to blade clearance requirements and may require extending the rotor shaft further from the top of the tower, thereby increasing moments and, hence, structural cost in the rotor shaft, bearings and bedplate support structure. In addition, impact loads experienced when the rotor hits the teeter stops during start-up operations may require provisions for impact load absorption mechanisms, further increasing costs.

4.4.3.5 Servo Flap vs Direct Pitch Control

A review of past wind generator systems, as well as prior experience at Kaman, suggested that the servo flap concept was a viable candidate for blade pitch control. Experimental systems built in the USSR as early as the 1930s had employed servo flap control, and continuing interest in the concept was evidenced in their contemporary reports on wind turbine technology. The principal attributes cited in wind turbine literature for the servo flap concept are also those which have been realized in helicopter applications: low control forces for pitch actuation, since the concept is an aerodynamic servo; and pitch stabilization of the rotor blades due to both the inherent static pitch stability contribution of the flap (analogous to an aircraft horizontal tail) and pitch damping which it also contributes during transient pitch oscillations.

An additional benefit of the servo flap is that of protecting the high value rotor system from damage by the bending loads developed by gale force winds. This possibility could be realized for a free feathering blade so that while parked, the pitch stabilization potential of the servo flap could be exploited for load alleviation, trimming the blade to a zero lift condition under any wind condition. This capability could be translated either into a higher maximum wind speed capability for a given structural strength blade (and associated weight and cost) or into less stringent design criteria and a reduced blade cost.

The evaluation of the servo flap concept was organized into an examination of four areas of rotor operational requirements:

1. Start-up of the WGS at low wind speeds, including synchronization with the utility network

2. Operation at rated rpm over the design wind speed range

3. Shutdown of the WGS, from rated to zero rpm, including emergency (load dump), as well as normal shutdown cycles

4. Load limiting (blade bending load alleviation) during gale force winds while stationary.

It was determined that a servo flap of reasonable area and moment arm to the blade pitch axis does not have adequate authority to control the blade to the required start-up pitch schedule. The very low dynamic pressure associated with low wind starts (e.g., 12 N/m^2 at 4.5 m/s) produces flap loads too low to overcome feathering friction and gravity induced pitching moments. Not until rotor speed increases to approximately 50% of rated speed was control estimated to become adequate. Potential solutions for this problem include a light duty pitch positioning device to preset the starting pitch angle or auxiliary starting torque provided by combination motor/generator (to compensate for the low torque of a fixed pitch start-up mode).

Evaluation of operation at rated rpm indicated that control requirements for the WGS servo flap are comparable to that of well proven helicopter applications. The influence of feathering bearing friction on flap response to control inputs can be a problem, particularly in view of the long component life required for the WGS. An attractive alternative to feathering bearings, which would serve to enhance flap control response, would be an all elastic hinging system for the rotor blade. However, this concept has not been developed to the point where it was judged feasible for the first generation WGS.

For normal shutdown cycles of the WGS, the control demands on the servo flap are analogous to those of the start-up cycle. Controllability becomes marginal at about 50% of rated speed and inadequate to track and hold a pitch profile that would insure attaining a complete (zero rpm) shutdown. In this mode, as in the start-up mode, a pitch lock could be provided to supplement the flap, with the blade "flown" to the feathered pitch angle before the rpm decayed to the critical point. Of greater concern, from the operational viewpoint, is the emergency shutdown capability required following a load dump, such as that resulting from a gust induced over-torque or a network fault initiation of the WGS circuit breakers. Under these conditions, a tight, high resolution pitch control system is required to prevent severe rotor overspeed, as discussed in Section 5. Again, an elastic hinging concept would enhance servo flap response and could be expected to contribute to adequate control capability.

An analysis of the servo flap's ability to limit loads during high winds produced very promising results for the servo flap controlled free-feathering blade concept. Preliminary evaluations had indicated that the critical situation would occur for gusts causing sudden changes in blade angle of attack, superimposed on the gale force (maximum) steady state wind. The critical concern was that the time rate of blade lift build-up in a blade angle of attack changing gust might be fast enough to substantially lead the blade pitching response generated by the servo flap. This could result in peak airloads comparable to those of a conventional fixed pitch rotor exposed to the same wind environment.

An analysis of the response of the conceptual design task high power (1000 KW) WGS blade design (with servo flap) was performed, utilizing a single degree of freedom (feathering) model of the blade and flap. The results of this analysis, shown in Figure 4-18, indicate that a fixed pitch servo flap of feasible proportions can provide a high degree of load alleviation, up to very high wind speeds and angle of attack changing gust rates. This feature of the servo flap concept can be implemented with a direct pitch controlled blade using a fixed flap to reduce gust induced loads without the complexity and limitations of the servo flap only controlled blade.

The results discussed above indicated that the servo flap, if employed as the only pitch control device for the WGS, would be very effective in alleviating wind loads, and adequate for pitch regulation during normal operation. In both the start-up and shutdown modes, however, the (unaided) servo flap was found to lack adequate controllability and pitch resolution. Auxiliary devices and control elements, and special operational procedures would be required to compensate for these deficiencies and to preserve the load alleviation and stabilization potential of the servo flap concept.

Due to the complexity of the servo flap control system, and the attendant increased development risks and cost, a direct blade root pitch control configuration was recommended for the WGS preliminary designs. This decision was based on the requirement to utilize current state-of-the-art, proven technology for the first generation WGS units. However, it is recommended that the potential of the servo flap concept in a WGS application be examined in a technology development program, particularly if blade load alleviation becomes an important factor for cost or operational feasibility reasons. The first task in this examination would be to evaluate the use of a fixed flap, essentially a modified blade tip configuration, to provide the required blade load alleviation and stabilization. Subsequent steps would examine a servo flap controlled WGS rotor, with and without root control provisions. This program would draw heavily on the results of other NASA sponsored servo flap/direct control rotor programs now being conducted by Kaman, and utilize the solid technical base provided by these projects to examine the unique requirements of the WGS rotor control system.

4.4.4 Selected Concept for Preliminary Design

The rotor subsystem design studies and analyses described above led to the selection of filament wound composite blades mounted on a hingeless hub. Blade pitch is controlled by direct mechanical controls at the blade root. As described in the next section, refinements of this concept through the preliminary design process resulted in modifications to this configuration. However, the basic selections of filament wound composite blades, hingeless hub and direct pitch controls were retained throughout the evolution of the preliminary designs and are recommended for subsequent development, test and demonstration.

Rotor Radius - 21.6m (71 ft.)

Blade Chord - 1.37m (4.5 ft.)

Blade Weight - 1263 kg (2781 lbs.)

$I_{\theta\theta}$ - 104 kgm^2 (76.83 slug-ft^2)

Flap Area x Arm - 2.55m^3 (90 ft^3)

$C_{L_{max}}$ - 1.5 Blade, 1.2 Flap

Gust Profile - Ramp α change

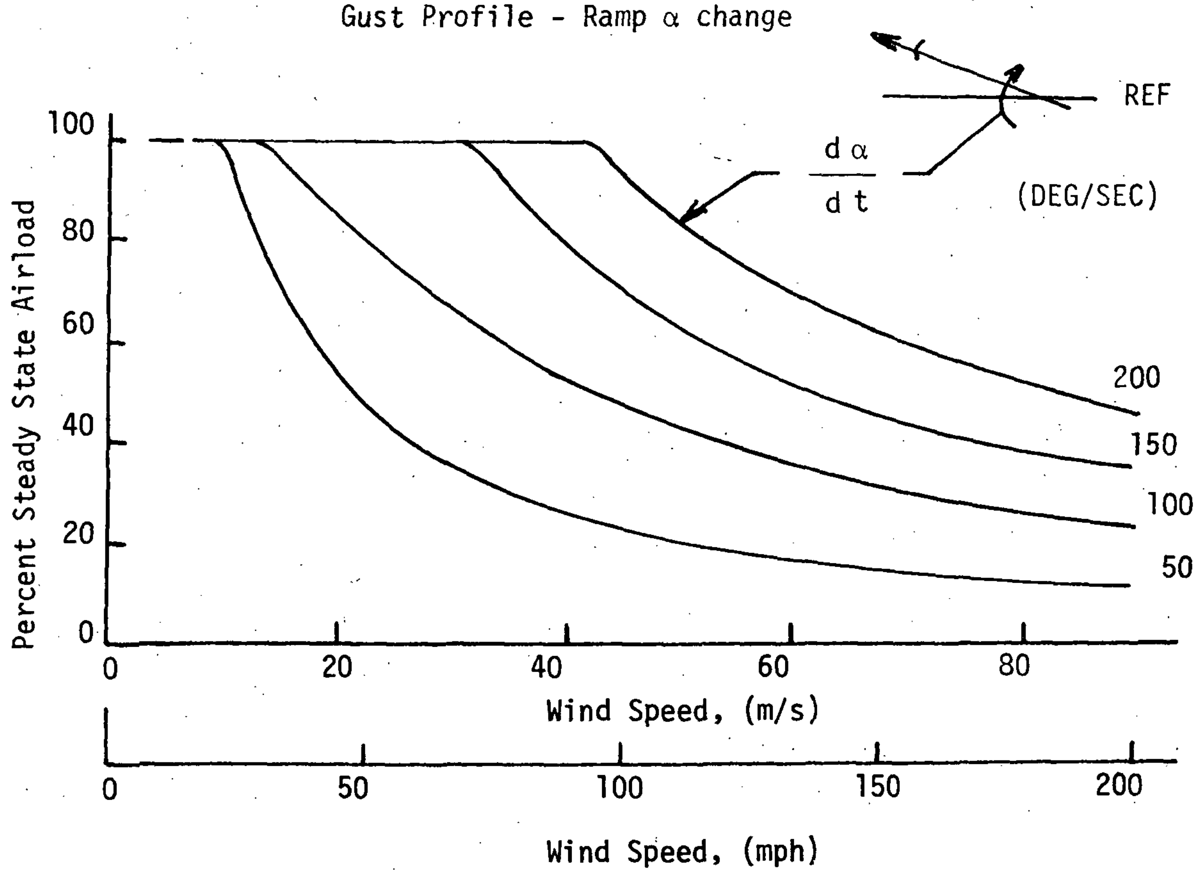


Figure 4-18. Wind Load Alleviation Characteristics of a Fixed Incidence Servo Flap.

4.5 Selected Concept Preliminary Design and Analysis

4.5.1 Design Description

The rotor designs for both the 500 KW and 1500 KW WGS units are similar in concept. The subsystem consists of the rotor head assembly, hub, blade grips and blades. These are shown in Figures 4-19 through 4-28, and the principal design parameters are listed in Table 3-20.

4.5.1.1 Rotor Head Assembly

The rotor head is a two-bladed hingeless hub assembly of rigid construction supported by tapered roller bearings on a non-rotating hollow spindle shaft. A central quill shaft transmits rotor torque to the generator drive system. A linear-travel blade pitch control mechanism is supported by the rotating quill shaft within the non-rotating hub spindle shaft. Multiple lug blade grips and pitch control bearings complete the rotor head assembly.

4.5.1.2 Rotor Hub

The hub is constructed as a weldment of large rolled plate steel cylinders shown in Figure 4-19. A cutaway view is shown in Figure 4-20. The larger two cylinders have axes coincident with the rotor blade pitching axes, and furnish the mounting for the two blade pitch bearings. These cylinders are pierced by a smaller cylinder containing the hub bearings. A welded structure was selected for buildup of this cruciform assembly to reduce fabrication cost. Low stress levels are achieved in all parts of this hub design by providing generous cross-sections throughout.

Hub bearings are a tapered roller pair, in which all thrust is carried by the aft bearing. Since these bearings effectively straddle the hub, overhung loads are minimized; the modest loads which remain are well within the capacity of the selected bearings for the 30 year life of the WGS.

4.5.1.3 Blade Grips

Forged aluminum grip fittings are provided to connect the blades to the hub. These fittings serve to redistribute blade loads into the pitch bearings and the pitch control linkage. Connection to the hub is through a pitch bearing (one for each blade), which is of crossed roller configuration. The crossed roller bearing is capable of reacting large moments, as well as thrust in any direction. Connection of each blade to its grip is by means of a double pinned multi-lug joint which picks up the main spar fitting. A truss to each blade trailing edge fitting carries trailing edge spline loads. The pitch control system described in Section 7 is linked directly to a clevis in each grip fitting. Linear movement of the pitch control beam causes each blade to rotate about its pitch axis. Sufficient travel is provided for blade pitch control throughout the operating range, and for full feathering.

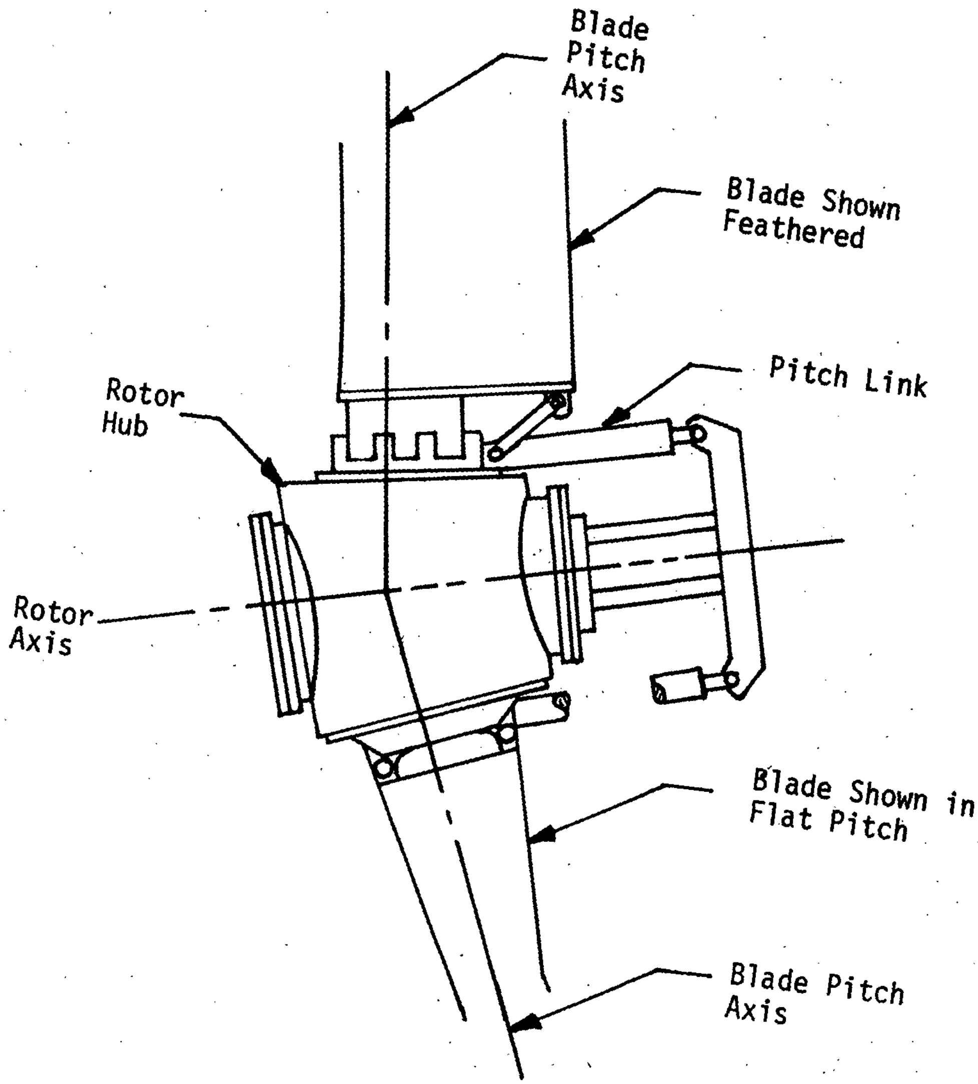


Figure 4-19. WGS Rotor Head Assembly.

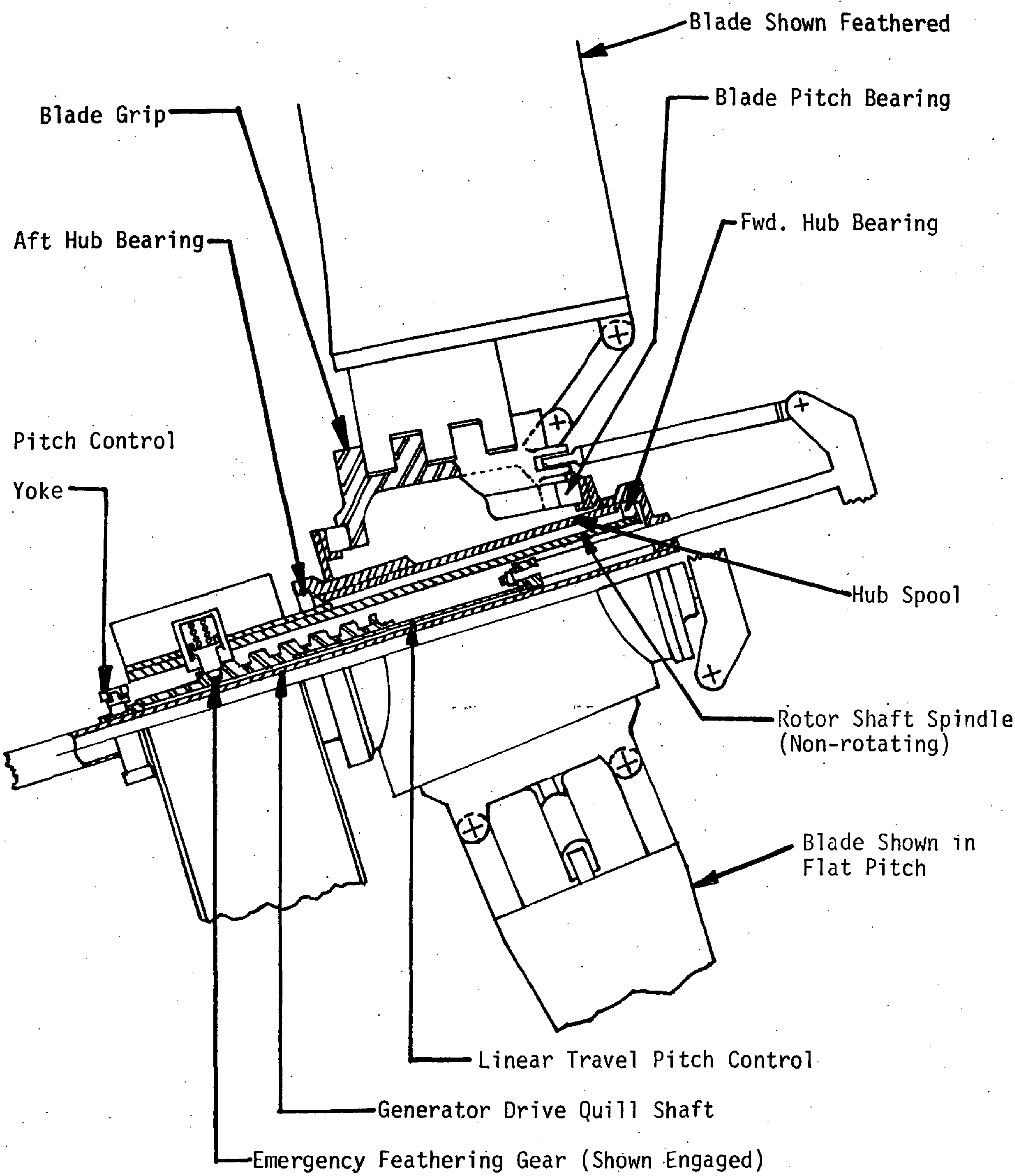
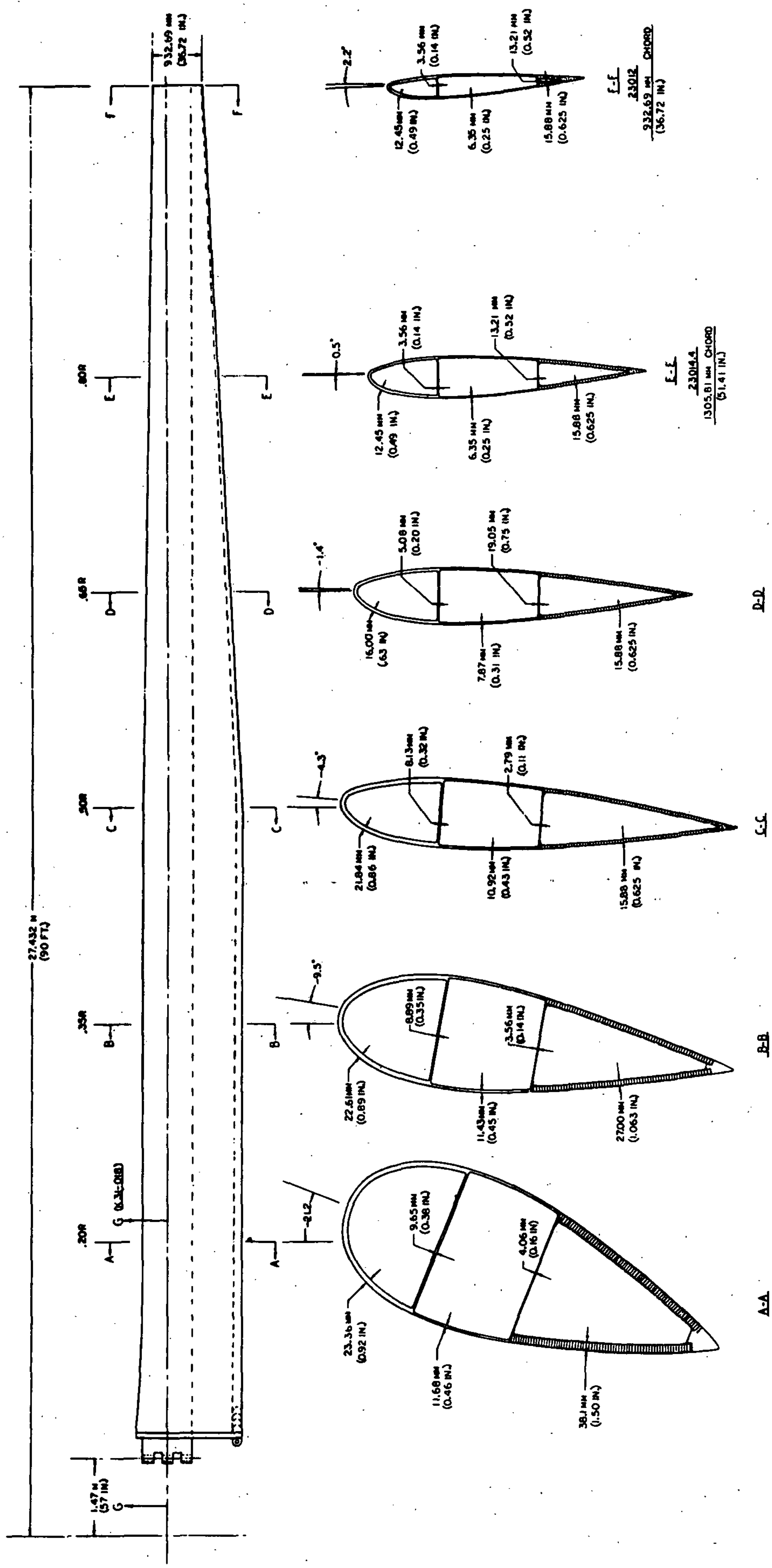


Figure 4-20. Rotor Head Assembly, Cutaway View.



27.432 m
(90 FT)
932.69 mm
(36.72 IN.)
1.47 m
(57 IN)
.20R
.35R
.50R
.65R
.80R
A-A
0040
1865.38 mm CHORD
(73.44 IN.)
23.36 mm
(0.92 IN.)
11.68 mm
(0.46 IN.)
38.1 mm
(1.50 IN.)
9.65 mm
(0.38 IN.)
4.06 mm
(0.16 IN.)
B-B
23029
1865.38 mm CHORD
(73.44 IN.)
22.61 mm
(0.89 IN.)
11.43 mm
(0.45 IN.)
27.00 mm
(1.063 IN.)
8.89 mm
(0.35 IN.)
3.56 mm
(0.14 IN.)
C-C
23018
1865.38 mm CHORD
(73.44 IN.)
21.84 mm
(0.86 IN.)
10.92 mm
(0.43 IN.)
15.88 mm
(0.625 IN.)
8.13 mm
(0.32 IN.)
2.79 mm
(0.11 IN.)
D-D
23016.2
1585.47 mm CHORD
(62.42 IN.)
16.00 mm
(.63 IN)
7.87 mm
(0.31 IN.)
15.88 mm
(0.625 IN.)
5.08 mm
(0.20 IN.)
19.05 mm
(0.75 IN.)
E-E
23014.4
1305.81 mm CHORD
(51.41 IN.)
12.45 mm
(0.49 IN.)
6.35 mm
(0.25 IN.)
15.88 mm
(0.625 IN.)
3.56 mm
(0.14 IN.)
13.21 mm
(0.52 IN.)
F-F
23012
932.69 mm CHORD
(36.72 IN.)
BLADE - PLANFORM/SECTIONS
1500 KW
SCALE 1/20 + 1/5
K31-017
KAMAN AEROSPACE CORPORATION

Figure 4-21.

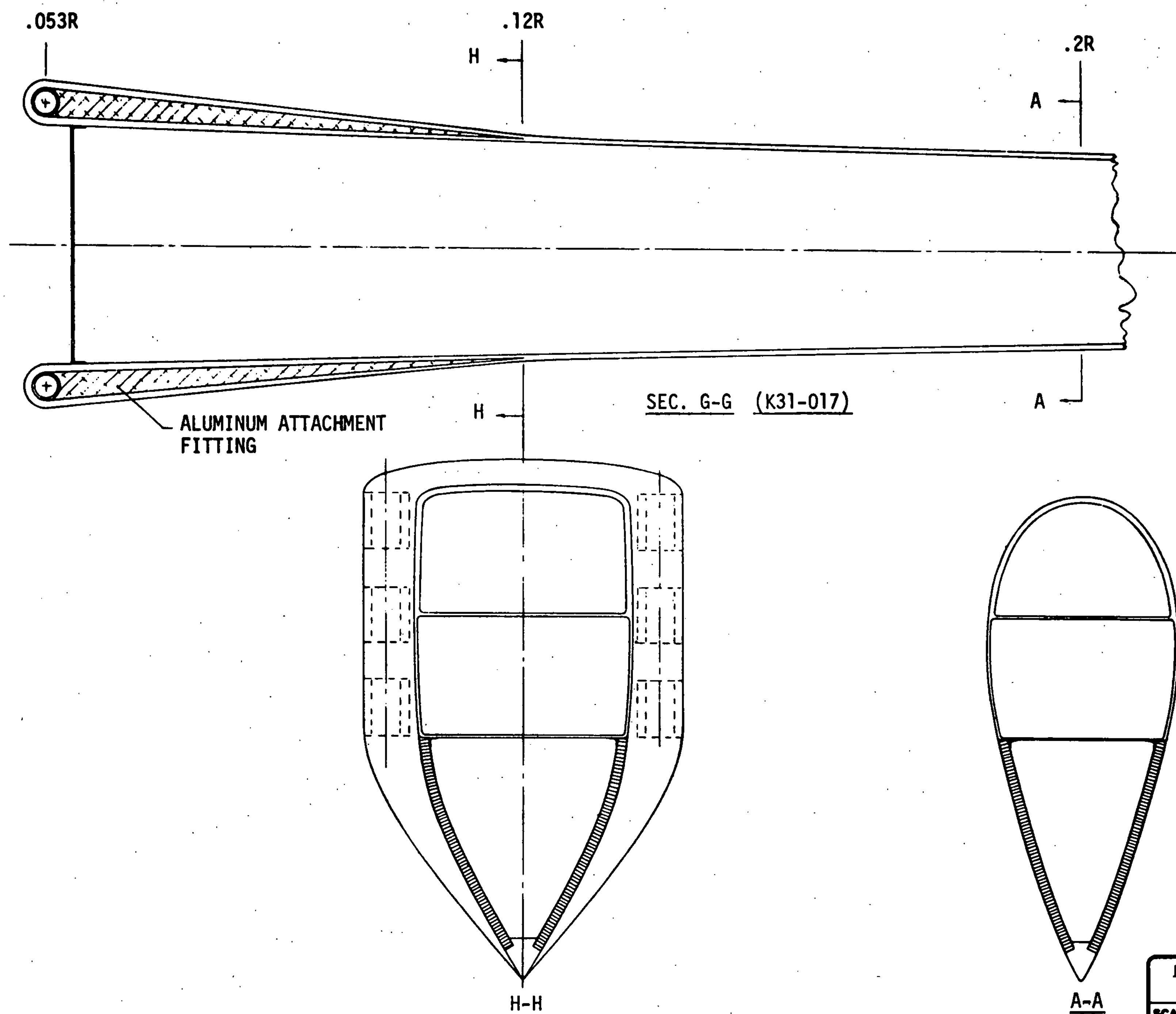


Figure 4-22.

BLADE - RETENTION
1500 KW
SCALE 1/5 | K31-018
KAMAN AEROSPACE CORPORATION

BLADE CONTOUR BUILD-UP (STARTING FROM OUTSIDE SURFACE)			
LOCATION	DESIGNATION-MATERIAL	THICKNESS INCHES	THICKNESS MM
A	EROSION GUARD-NEOPRENE	.062	1.57
	+60° S-2 GLASS	.041	1.04
	-60° S-2 GLASS	.041	1.04
	0° S-2 GLASS	.326	8.28
	-60° S-2 GLASS	.041	1.04
	+60° S-2 GLASS	.041	1.04
B	ALUMINUM MESH	.010	0.25
	+60° S-2 GLASS	.021	0.53
	-60° S-2 GLASS	.021	0.53
	0° S-2 GLASS	.167	4.24
	-60° S-2 GLASS	.021	0.53
	+60° S-2 GLASS	.021	0.53
C	ALUMINUM MESH	.010	0.25
	+60° S-2 GLASS	.015	0.38
	-60° S-2 GLASS	.015	0.38
	CORE ALUMINUM HONEYCOMB	4 LB. DEN.	
	120 WOVEN GLASS	.010	0.25
D	+60° S-2 GLASS	.070	1.78
	-60° S-2 GLASS	.070	1.78
E	+60° S-2 GLASS	.025	0.64
	-60° S-2 GLASS	.025	0.64
F	ALUMINUM MESH	.010	0.25
	+60 S-2 GLASS	.015	0.38
	-60 S-2 GLASS	.015	0.38
	120 WOVEN GLASS	.030	0.76
	CORE URETHANE FOAM	6 LB. DEN.	
G	+45° S-2 GLASS	.077	1.96
	-45° S-2 GLASS	.077	1.96
	0° S-2 GLASS	.613	15.57
	-45° S-2 GLASS	.077	1.96
	+45° S-2 GLASS	.077	1.96
H	ALUMINUM MESH	.010	0.25
	+45° S-2 GLASS	.038	0.97
	-45° S-2 GLASS	.038	0.97
	0° S-2 GLASS	.307	7.80
	-45° S-2 GLASS	.038	0.97
	+45° S-2 GLASS	.038	0.97
I	ALUMINUM MESH	.010	0.25
	+45 S-2 GLASS	.015	0.38
	-45 S-2 GLASS	.015	0.38
	CORE ALUMINUM HONEYCOMB	4 LB. DEN.	
	120 WOVEN GLASS	.010	0.25
J	+45 S-2 GLASS	.190	4.83
	-45 S-2 GLASS	.190	4.83
K	+45 S-2 GLASS	.080	2.03
	-45 S-2 GLASS	.080	2.03
L	ALUMINUM MESH	.010	0.25
	+45 S-2 GLASS	.015	0.38
	-45 S-2 GLASS	.015	0.38
	COMPRESSION MOULDED T.E. SPLINE		

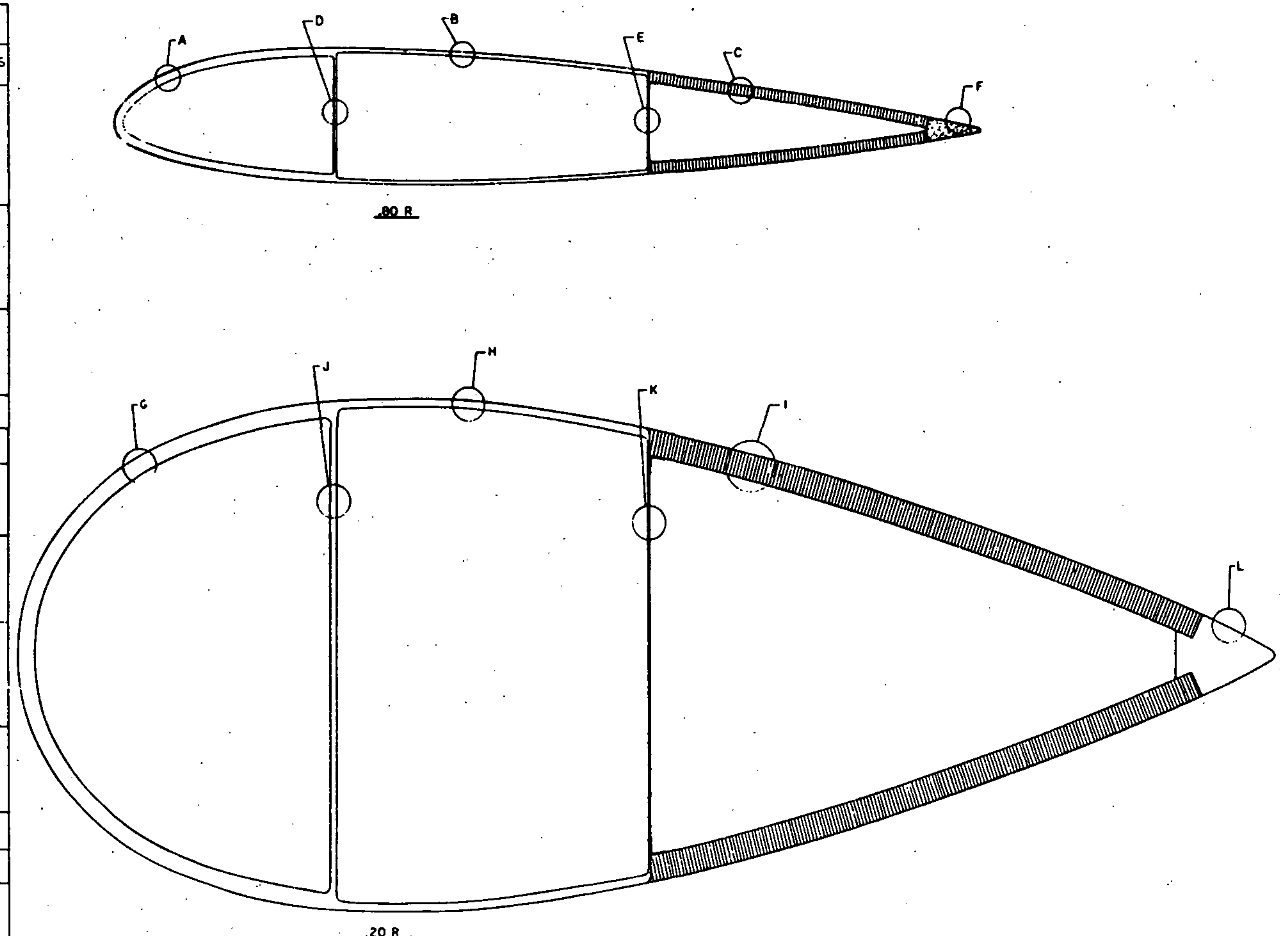


Figure 4-23.

BLADE-SECTION COMPONENTS
1500 KW
SCALE 1/2 K 31-020
KAMAN AEROSPACE CORPORATION
OLD WINDSOR ROAD, BLOOMFIELD, CONNECTICUT

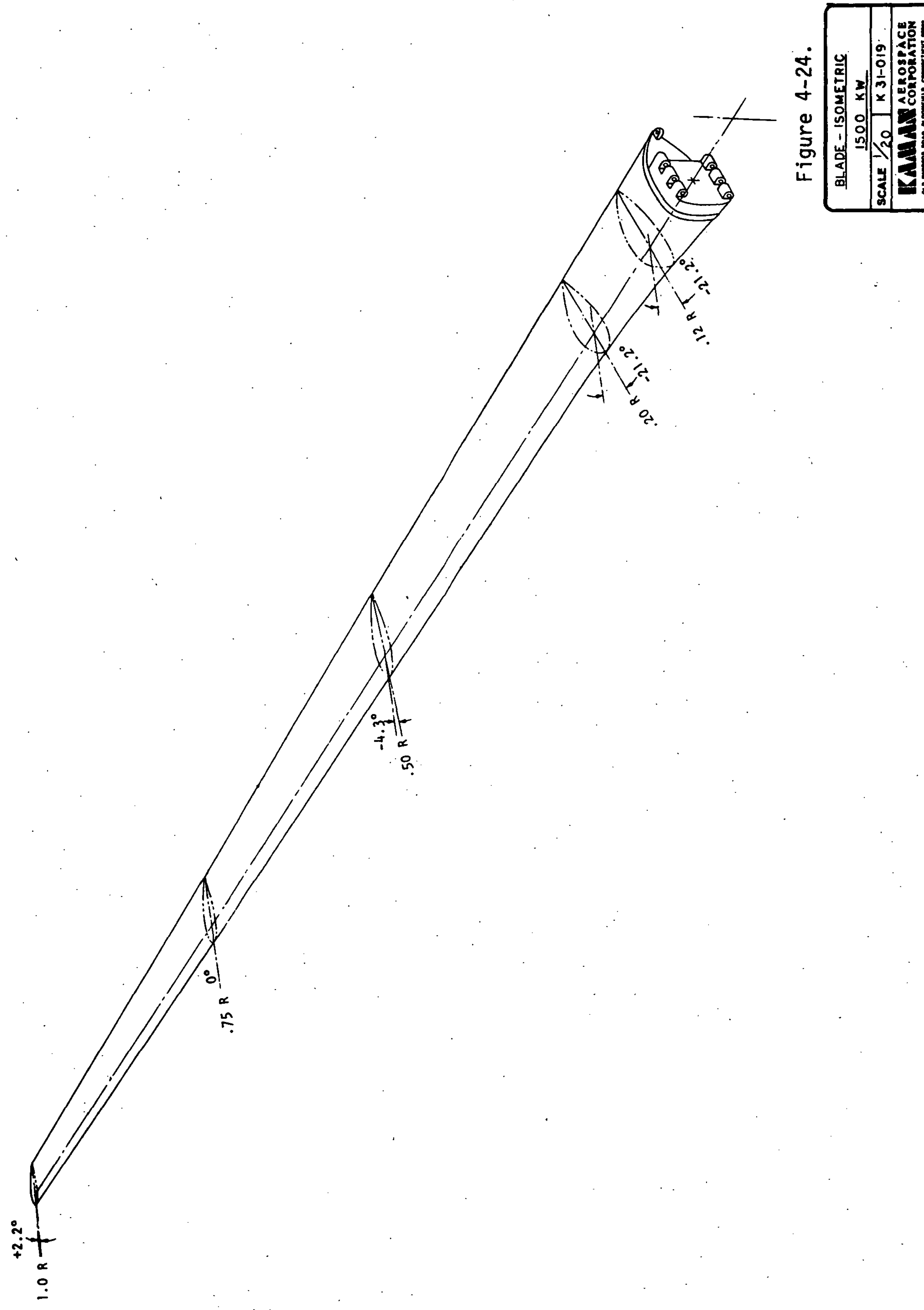

BLADE - ISOMETRIC
1500 KW
SCALE 1/20
K 31-019
KAMAN AEROSPACE CORPORATION
.12 R
-21.2°
.20 R
-21.2°
.50 R
-4.3°
.75 R
0°
1.0 R
+2.2°

Figure 4-24.

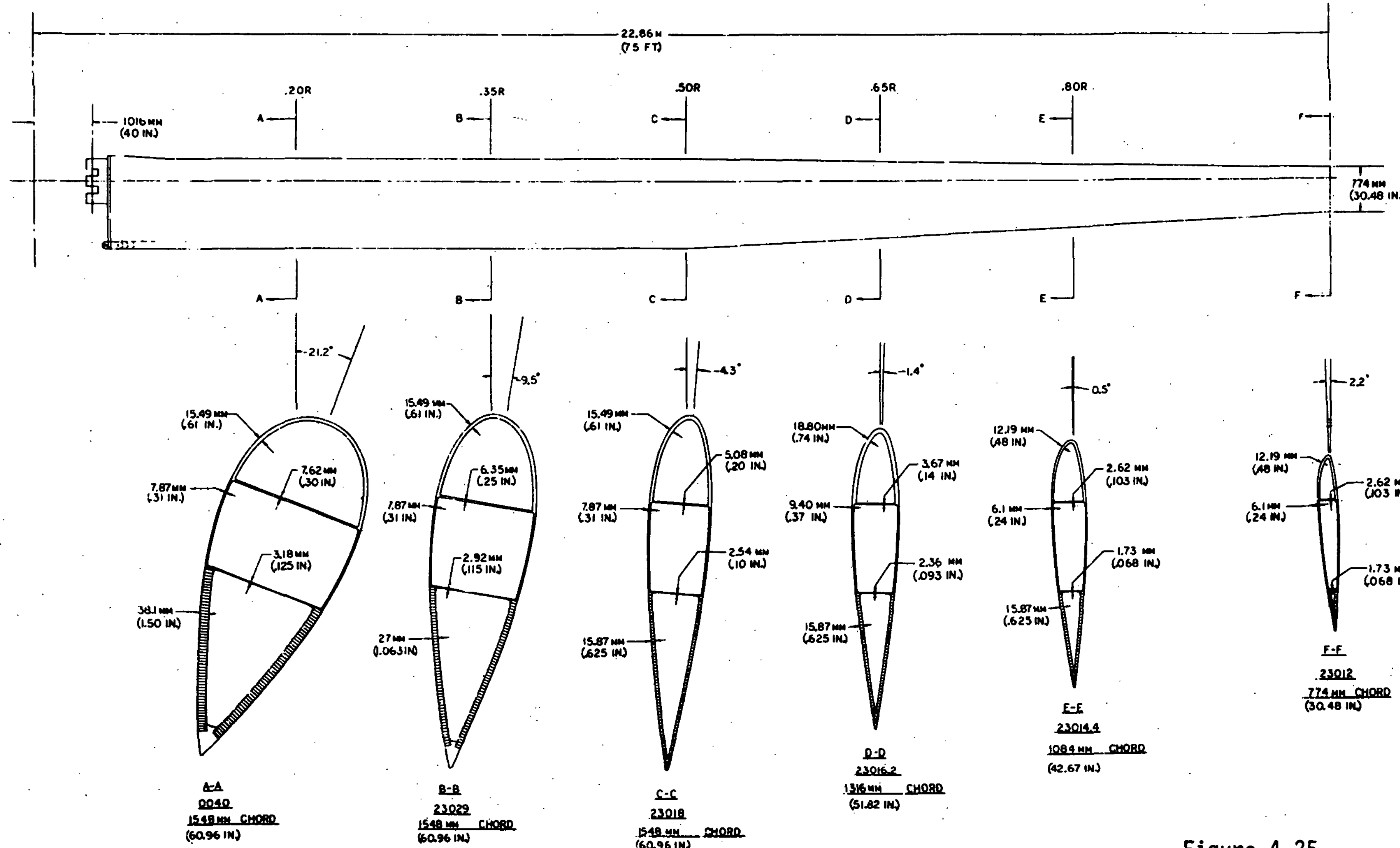


Figure 4-25.

BLADE PLANFORM/SECTIONS
500 KW
SCALE 1/20 + 1/5 | K31-026
KAMAN AEROSPACE CORPORATION

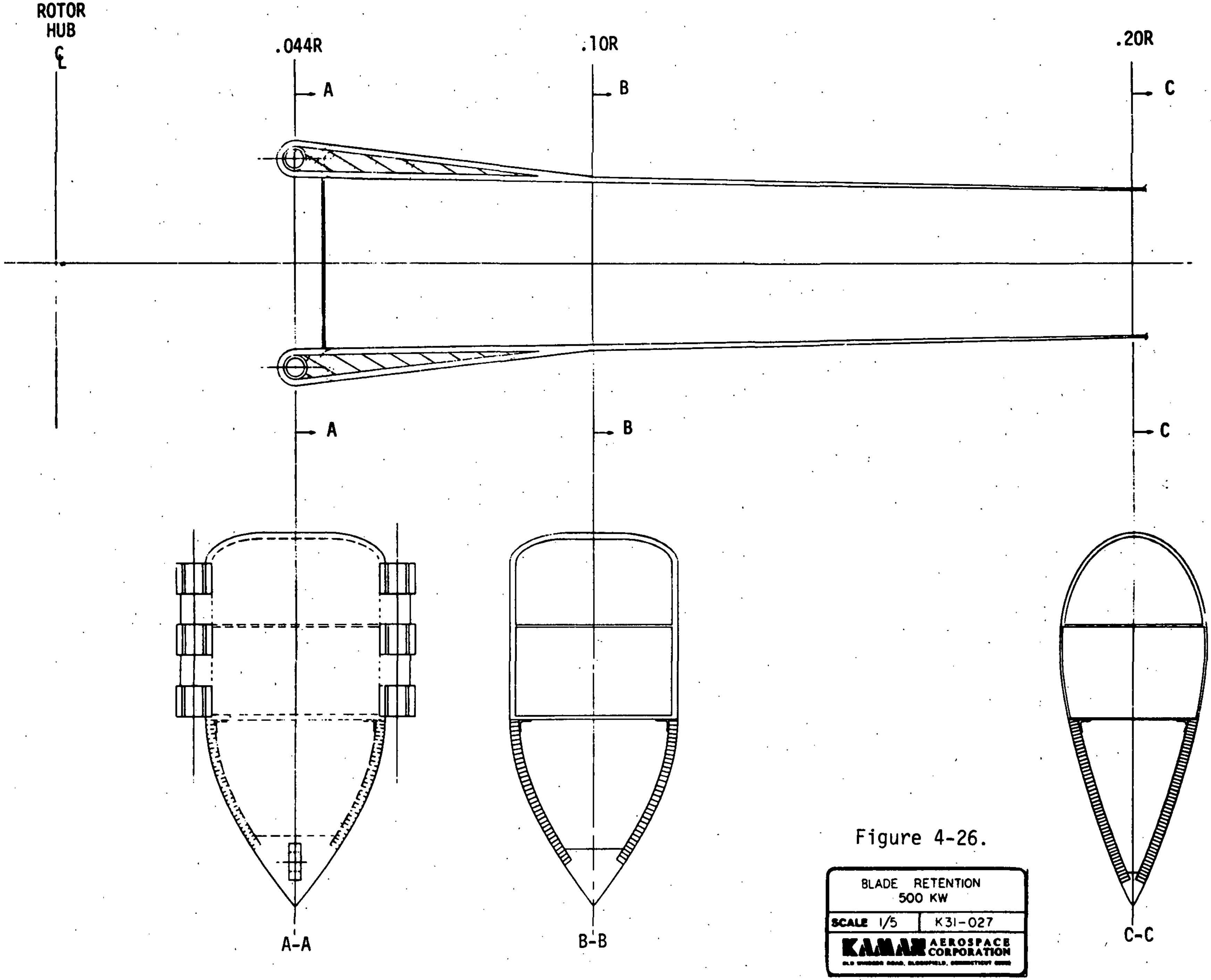

ROTOR
HUB
℄
.044R
.10R
.20R
A
B
C
A-A
B-B
C-C
BLADE RETENTION
500 KW
SCALE 1/5
K31-027
KAMAN AEROSPACE CORPORATION

Figure 4-26.

BLADE CONTOUR BUILD-UP (STARTING FROM OUTSIDE SURFACE)			
LOCATION	DESIGNATION-MATERIAL	THICKNESS INCHES	THICKNESS MM
A	EROSION GUARD NEOPRENE	.062	1.58
	+60° S-2 GLASS	.040	1.02
	-60° S-2 GLASS	.040	1.02
	0° S-2 GLASS	.320	8.13
	-60° S-2 GLASS	.040	1.58
	+60° S-2 GLASS	.040	1.58
B	ALUMINUM MESH	.010	0.25
	+60° S-2 GLASS	.020	0.51
	-60° S-2 GLASS	.020	0.51
	0° S-2 GLASS	.160	4.06
	-60° S-2 GLASS	.020	0.51
	+60° S-2 GLASS	.020	0.51
C	ALUMINUM MESH	.010	0.25
	+60° S-2 GLASS	.015	0.38
	-60° S-2 GLASS	.015	0.38
	CORE ALUMINUM HONEYCOMB	4 LB. DEN.	
	120 WOVEN GLASS	.010	0.25
D	+60° S-2 GLASS	.051	1.30
	-60° S-2 GLASS	.051	1.30
E	+60° S-2 GLASS	.034	0.86
	-60° S-2 GLASS	.034	0.86
F	ALUMINUM MESH	.010	0.25
	+60° S-2 GLASS	.015	0.38
	-60° S 2 GLASS	.015	0.38
	120 WOVEN GLASS	.030	0.76
	CORE URETHANE FOAM	6 LB. DEN.	
G	+45° S-2 GLASS	.050	1.27
	-45° S-2 GLASS	.050	1.27
	0° S-2 GLASS	.610	15.49
	-45° S-2 GLASS	.050	1.27
	+45° S-2 GLASS	.050	1.27
H	ALUMINUM MESH	.010	0.25
	+45° S-2 GLASS	.025	0.64
	-45° S-2 GLASS	.025	0.64
	0° S 2 GLASS	.210	5.33
	-45° S-2 GLASS	.025	0.64
	+45° S 2 GLASS	.025	0.64
I	ALUMINUM MESH	.010	0.25
	+45° S-2 GLASS	.015	0.38
	-45° S-2 GLASS	.015	0.38
	CORE ALUMINUM HONEYCOMB	4 LB. DEN.	
	120 WOVEN GLASS	.010	0.25
J	+45° S-2 GLASS	.150	3.81
	-45° S-2 GLASS	.150	3.81
K	+45° S-2 GLASS	.062	1.57
	-45° S-2 GLASS	.062	1.57
L	ALUMINUM MESH	.010	0.25
	+45° S-2 GLASS	.015	0.38
	-45° S-2 GLASS	.015	0.38
	COMPRESSION MOULDED T.E. SPLINE		

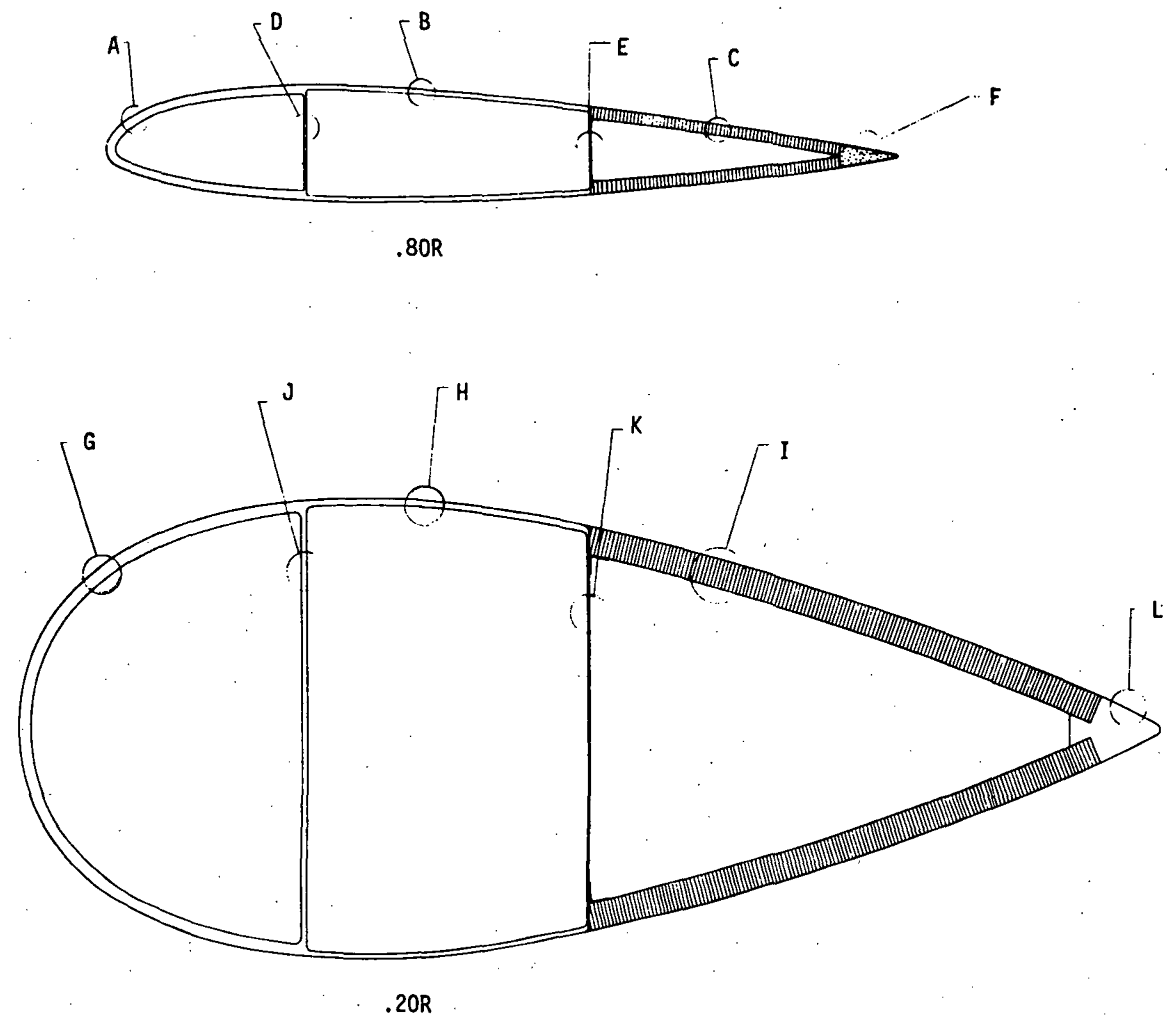


Figure 4-27.

BLADE-SECTION COMPONENTS
500 KW

SCALE 1/2 | K 31-028

KAMAN AEROSPACE CORPORATION
OLD WINDSOR ROAD, BLOOMFIELD, CONNECTICUT 06002

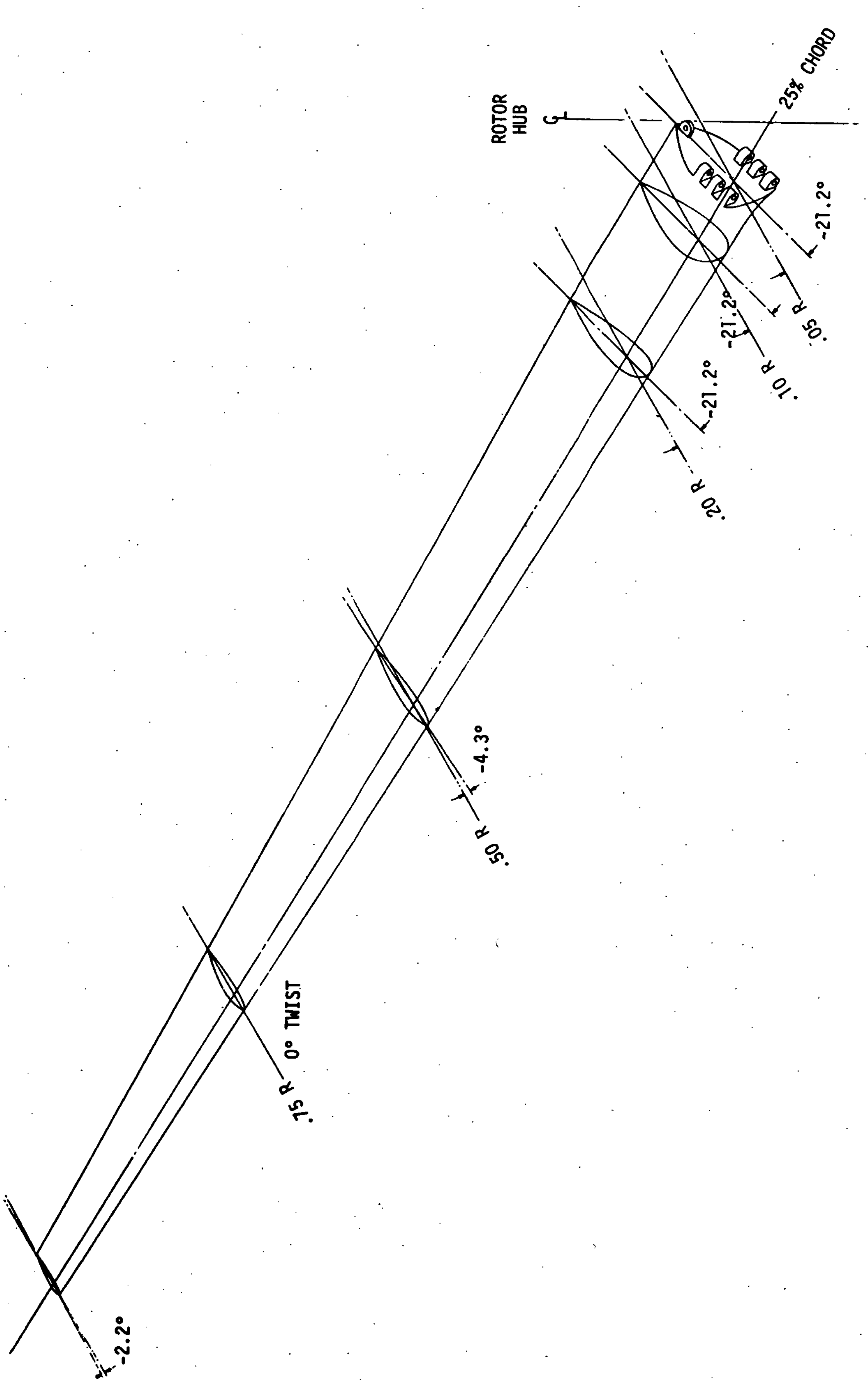


Figure 4-28.

BLADE ISOMETRIC 500 KW
SCALE 1/20
K31-025
KAMAN AEROSPACE CORPORATION

4.5.1.4 Rotor Blades

Construction - The WGS rotor blades are fiberglass composite, filament-wound construction, shown in Figures 4-21 through 4-24 for the 1500 KW system, and in Figures 4-25 through 4-28 for the 500 KW system. Blade stiffness and mass distributions are shown in Figures 4-29 through 4-32. Blade twist distributions are shown in Figure 4-33.

Unidirectional S-2 type fiberglass filaments are wound over a leading edge spar mandrel at ± 45° angles with respect to the blade centerline in the inboard stations of the blade, and at ± 60° angles in the outboard stations.

A second spar mandrel is positioned behind the partially wrapped leading edge spar, and additional layers of filament-wound S-2 glass are applied to both spar cells as noted in the blade contour build-up table of Figures 4-23 and 4-27.

Unidirectional layers of S-2 glass are interleaved spanwise during filament winding to increase out-of-plane bending stiffness, and to provide structural continuity and load transfer around the aluminum attachment fittings imbedded in the blade root, illustrated in Figures 4-22 and 4-26.

A third mandrel is positioned behind the two spar cells to position the aluminum honeycomb core blankets and trailing edge splines. A compression molded trailing edge spline extends from the blade root to mid-span for in-plane blade stiffness. The entire blade is filament wound with S-2 glass to complete the structure.

The outboard section of the blade leading edge is protected from sand and rain erosion by a neoprene guard. Lightning protection is provided by an aluminum mesh screen imbedded in the blade surface aft of the quarter chord.

Geometry - The blade planform is rectangular from the root to mid-span, and tapered (3:1 ratio) from mid-span to the tip. Thickness tapers from 40 percent chord at the blade root to 18 percent chord at mid-span and 12 percent at the blade tip. Blade twist is helical, from - 21.2° at 20 percent span to + 2.2° at the blade tip. Blade twist from root to tip is "wash-in" rather than "wash-out" as in typical helicopter rotors and fixed-wing aircraft. Geometry details are shown in Figures 4-21 and 4-25 for the 1500 KW and 500 KW systems, respectively.

4.5.2 Design Analysis

Rotor preliminary designs were supported by configuration, operational and structural analyses. Configuration analyses included blade geometry optimization which was accomplished as part of the system analyses described previously. These optimization studies led to the selection of blade thickness, planform and twist distributions. The major operational analyses included pitch control mode selection, overspeed limit calculations, blade tuning analyses, determination of blade flutter and divergence boundaries, and whirl resonance analyses. The major structural analyses covered the blade design, fatigue analysis and hub design. These are described below.

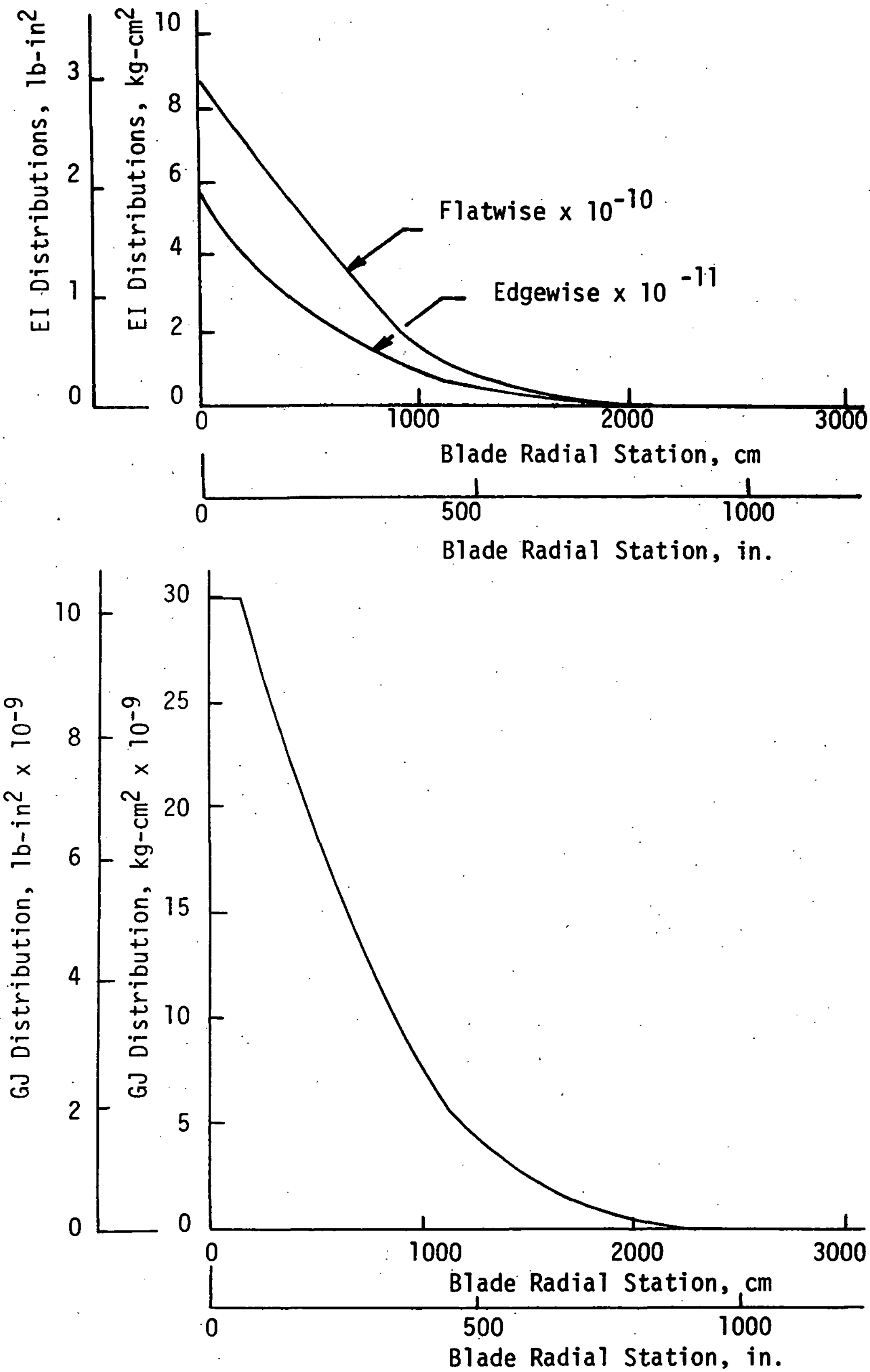


Figure 4-29. Blade Stiffness Distributions
500 kW System

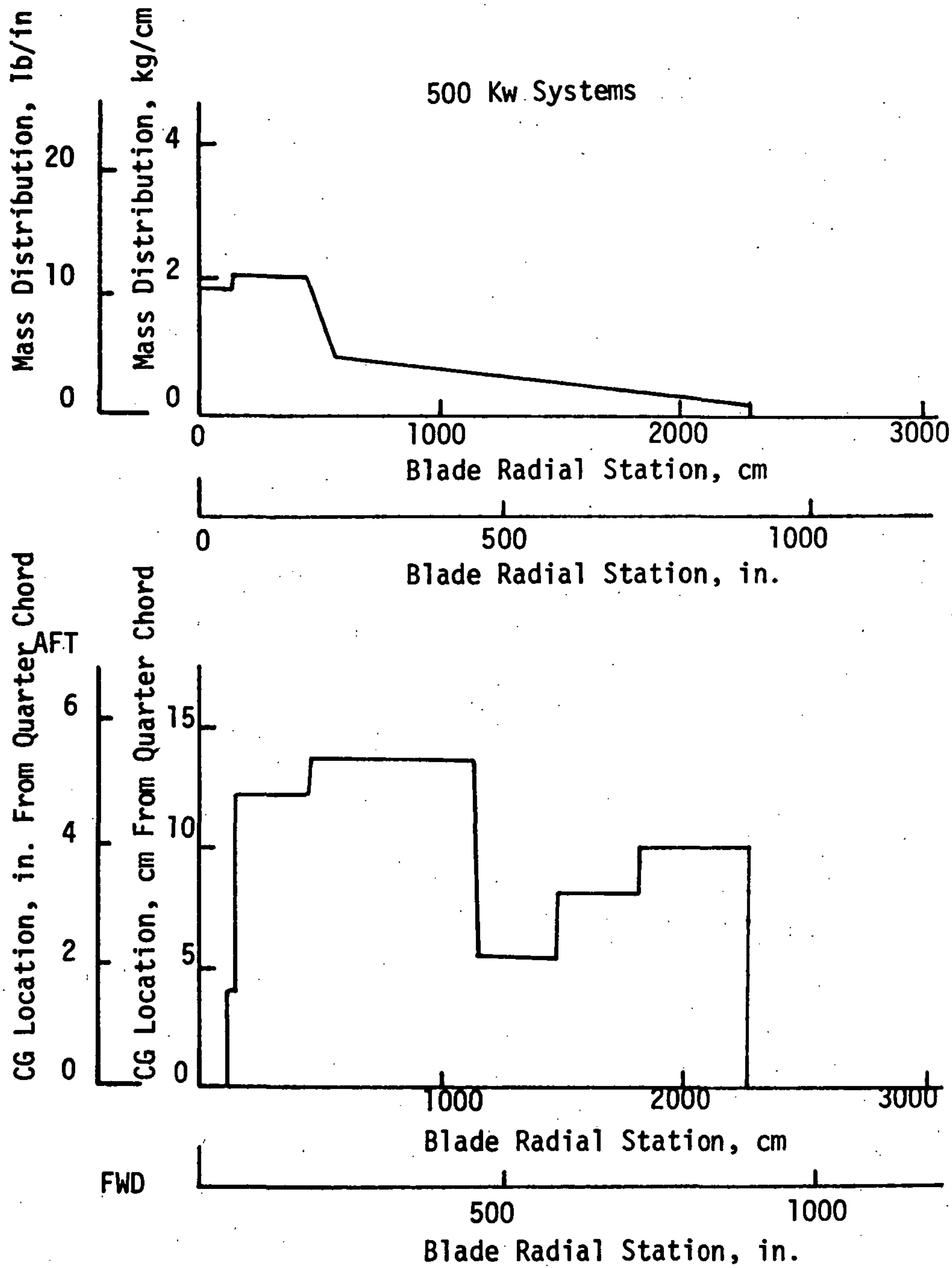


Figure 4-30 Blade Mass and CG Distributions

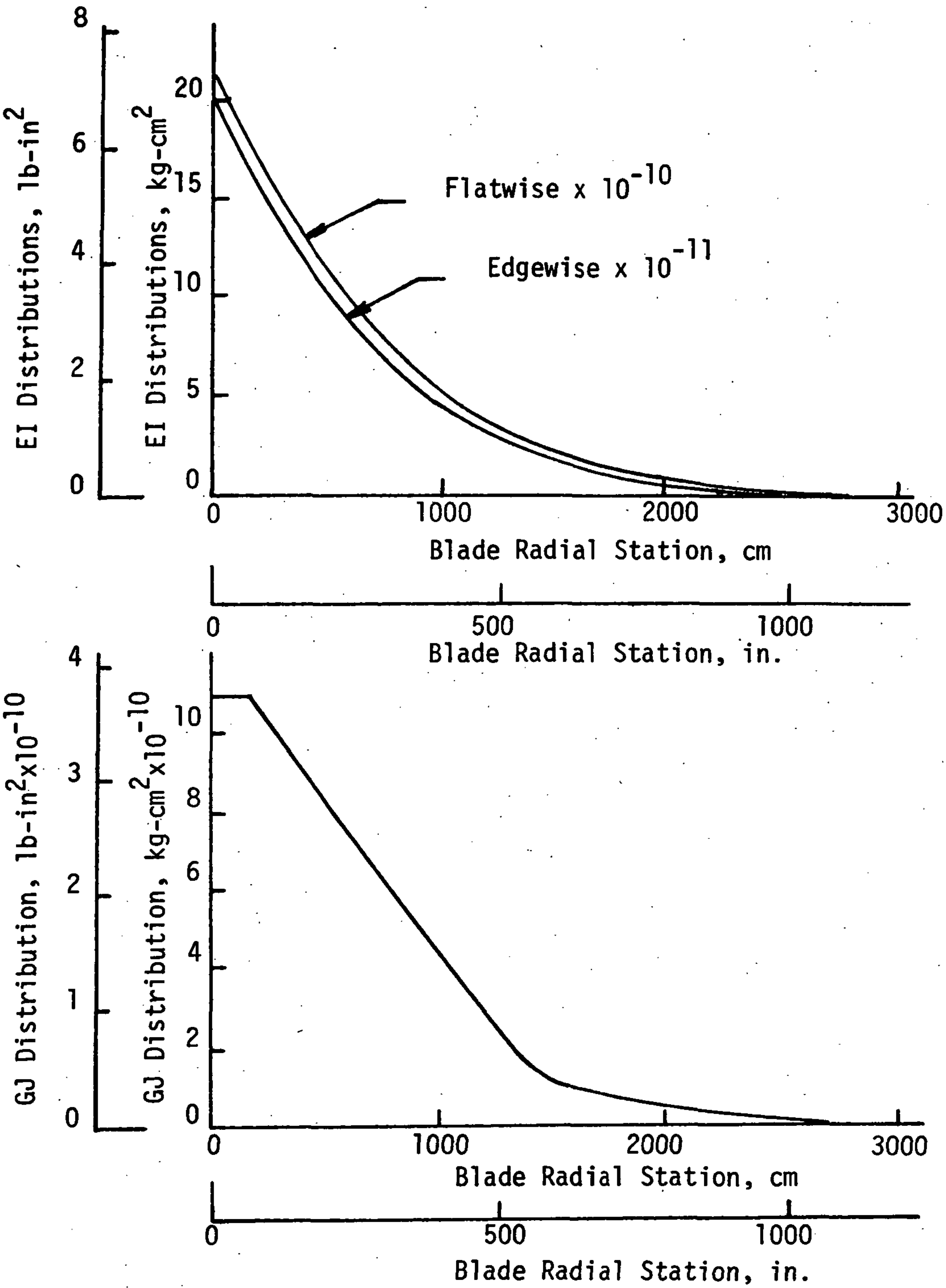


Figure 4-31. Blade Stiffness Distributions
1500 Kw System

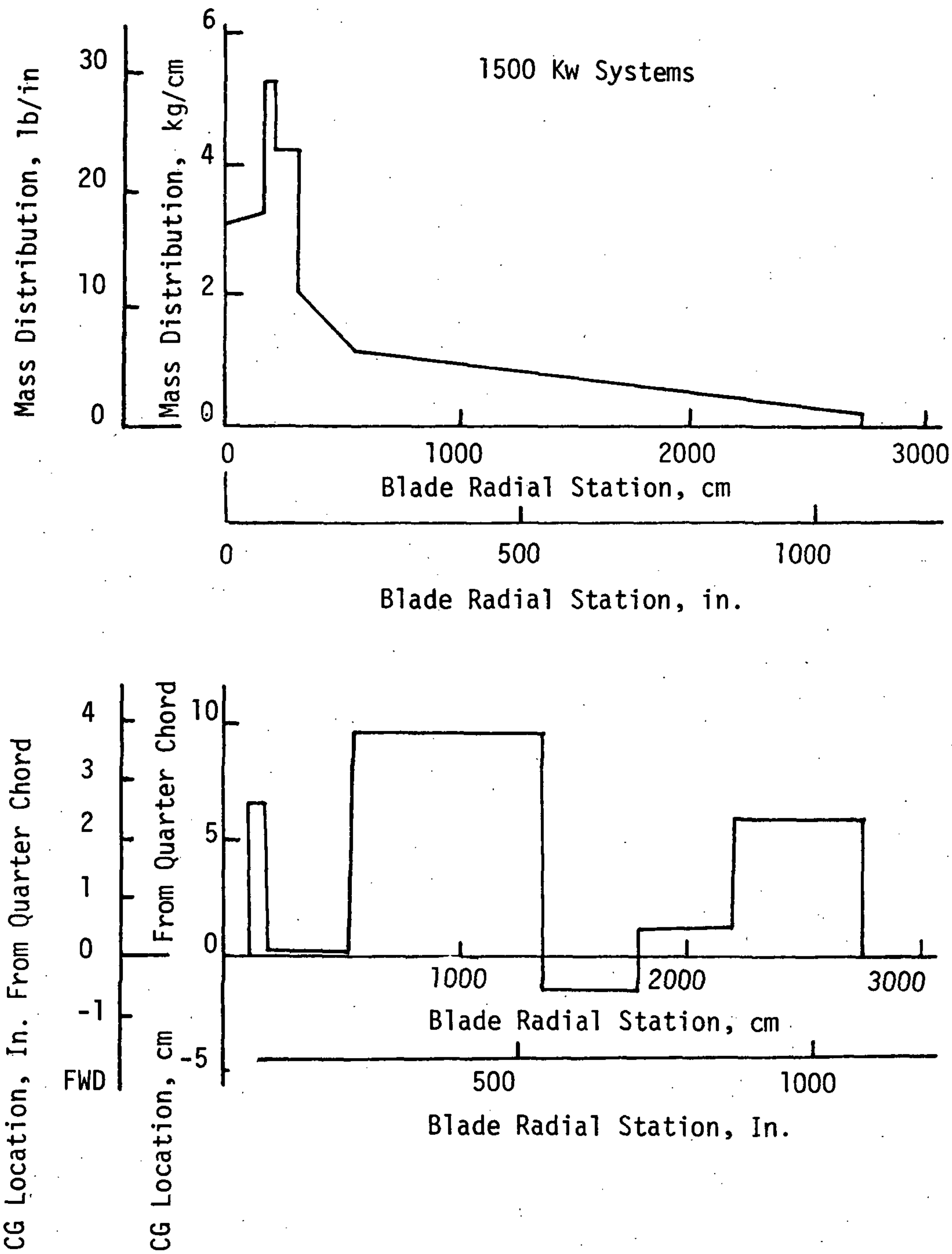


Figure 4-32. Blade Mass and CG Distributions

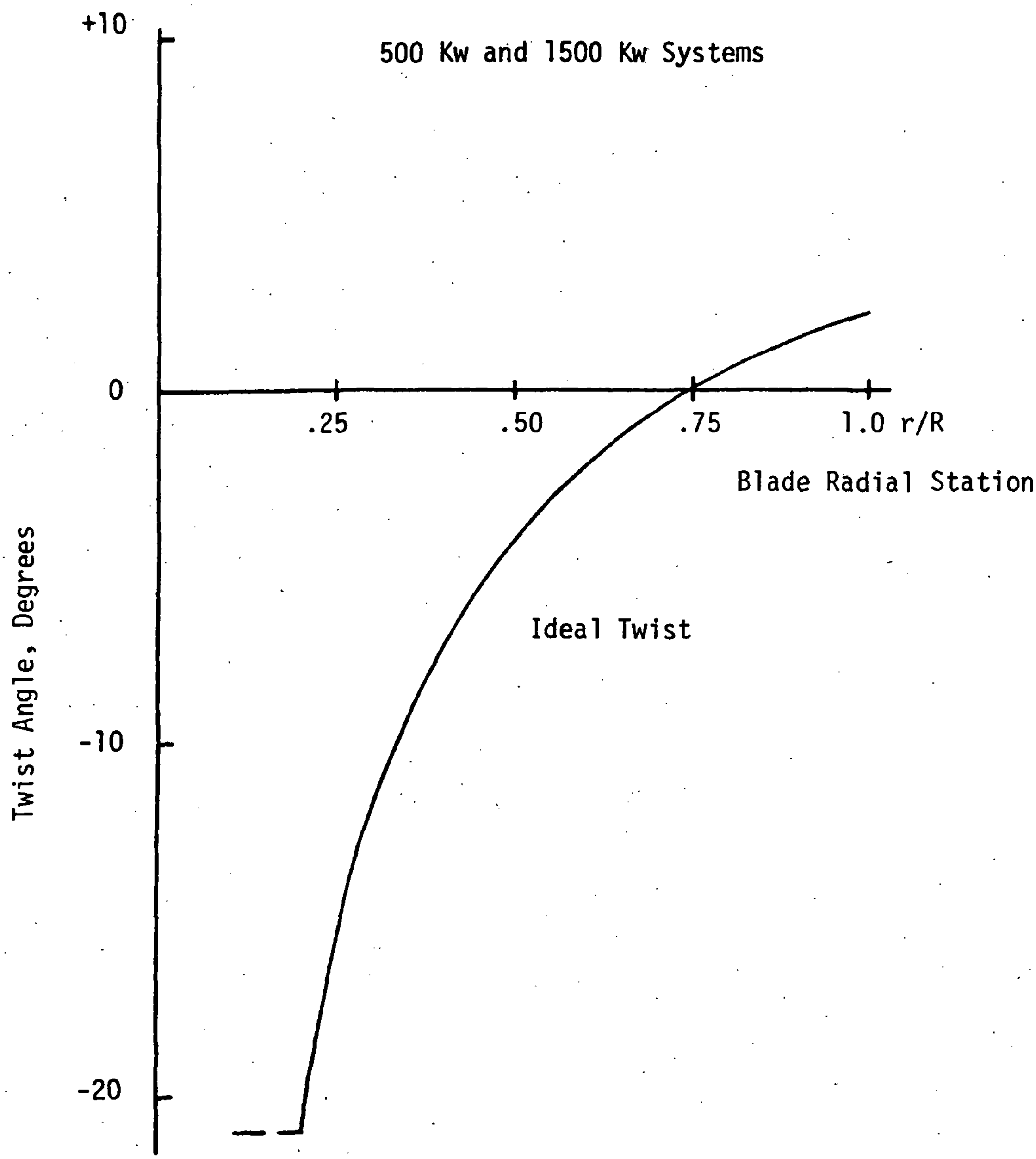


Figure 4-33. Blade Twist Distribution

4.5.2.1 Parametric Optimizations

Rotor geometry for the preliminary design WGS rotors was optimized on the basis of minimum overall energy cost as described in Section 3 and paragraph 4.4.2. Efficiency ratings and cost tradeoffs for several blade configurations are summarized in Table 3-12 of Section 3, in which the first configuration listed in each group is the one selected for preliminary design.

A summary of rotor design parameters for the 500 KW and 1500 KW WGS is presented in Table 4-1. The 45.7 m (150 ft) diameter rotor for the 500 KW system was designed for a median wind speed of 5.4 m/s (12 mph), and the 54.9 m (180 ft) diameter rotor for the 1500 KW system for 8 m/s (18 mph).

4.5.2.2 Performance Analyses

Variable pitch wind-driven rotors can be operated in either of two control modes; positive pitch or negative pitch. The former produces high thrust loads along the rotor axis which are non-productive, while the latter is primarily a torque-generating mode. The positive pitch mode at low wind speeds and low power levels is subject to rotor thrust instabilities familiar in the helicopter industry as the "vortex ring state" of rotor wake interference. (The vortex ring state is a condition in which the descent velocity of a lifting rotor approximates the rotor downwash velocity, causing development of a large recirculation vortex around the periphery of the rotor disc. For the wind generator system rotor, a similar condition exists when the thrust-induced wake velocity is approximately equal to the wind velocity.) Thrust fluctuations associated with the vortex ring state cause vibratory loads at the top of the tower up to $\pm$ 60% of the steady thrust load. Large blade tip deflections also occur, increasing the danger of blade tip intersection with the tower. For these reasons, the negative pitch control mode was selected for the WGS.

In the negative pitch mode, the pitch control system must be sufficiently responsive to changes in wind speed to avoid excessive power into the generator, which might cause it to drop off the line. Conversely, the pitch change mechanism need not be as precise as in the positive pitch mode, where small errors cause large thrust and power changes. The pitch control rate selected for the WGS is 5 degrees per second to meet these requirements. A description of the blade pitch control subsystem is contained in Section 5.

Power and thrust contours for the 500 KW and 1500 KW WGS are shown in Figures 4-34 through 4-37.

4.5.2.3 Rotor Overspeed Analysis

Normal long-period variations in wind speed are accommodated by controlling blade pitch to maximize wind energy recovery up to rated wind conditions and to maintain rated power and rotor speed at wind speeds above rated wind speed. Once on-line, rotor speed is stabilized by the synchronous electrical generator and utility network grid load. Within the capacity of the generator to absorb torque from the WGS rotor, on a short term basis equal to 200 percent of rated torque, rotor speed will be maintained at its rated value. At the cut-out wind speed, the WGS is taken off the line and shutdown by feathering the rotor blades.

TABLE 4-1. ROTOR DESIGN PARAMETERS

System Rated Power, KW	500	1500
Diameter, m (ft)	45.7 (150)	54.9 (180)
Solidity	.03	.03
Maximum Chord, m (ft)	1.55 (5.08)	1.86 (6.12)
Tip Chord, m (ft)	.78 (2.54)	.93 (3.06)
Rotor Power, KW	560	1648
Rotor Torque, kNm (ft-kips)	165 (122)	457 (337)
Tip Speed, m/s (fps)	77.4 (254)	98.8 (324)
Rotor Speed, rpm	32.3	34.4
Rated Wind, m/s (mph)	8.9 (20)	11.2 (25)
Cut-in Wind, m/s (mph)	4.0 (9)	5.4 (12)
Cut-out Wind, m/s (mph)	13.4 (30)	20.1 (45)
Total Rotor Weight, kg (lb)	8102 (17863)	17432 (38432)

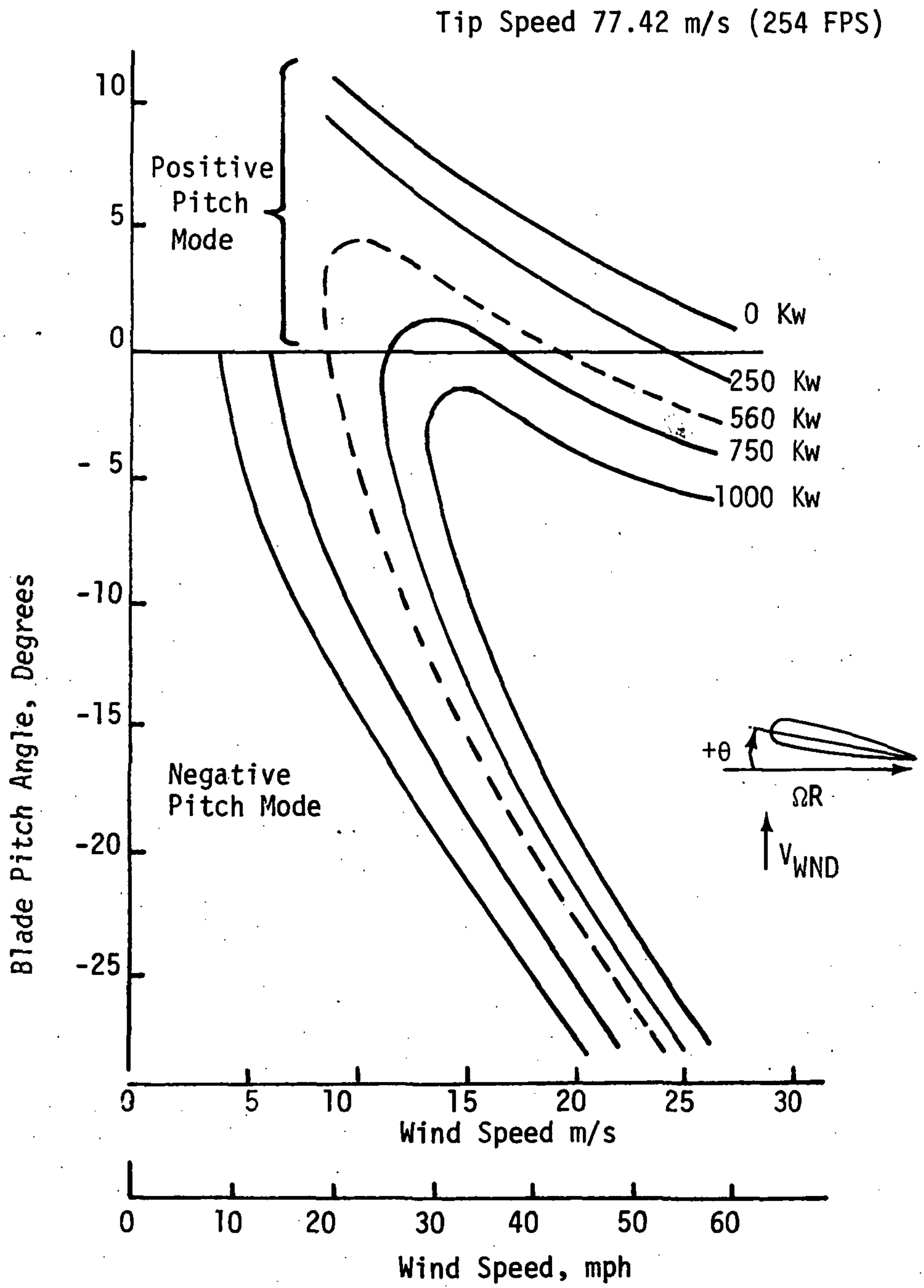


Figure 4-34. Rotor Power Contours, 500 Kw System

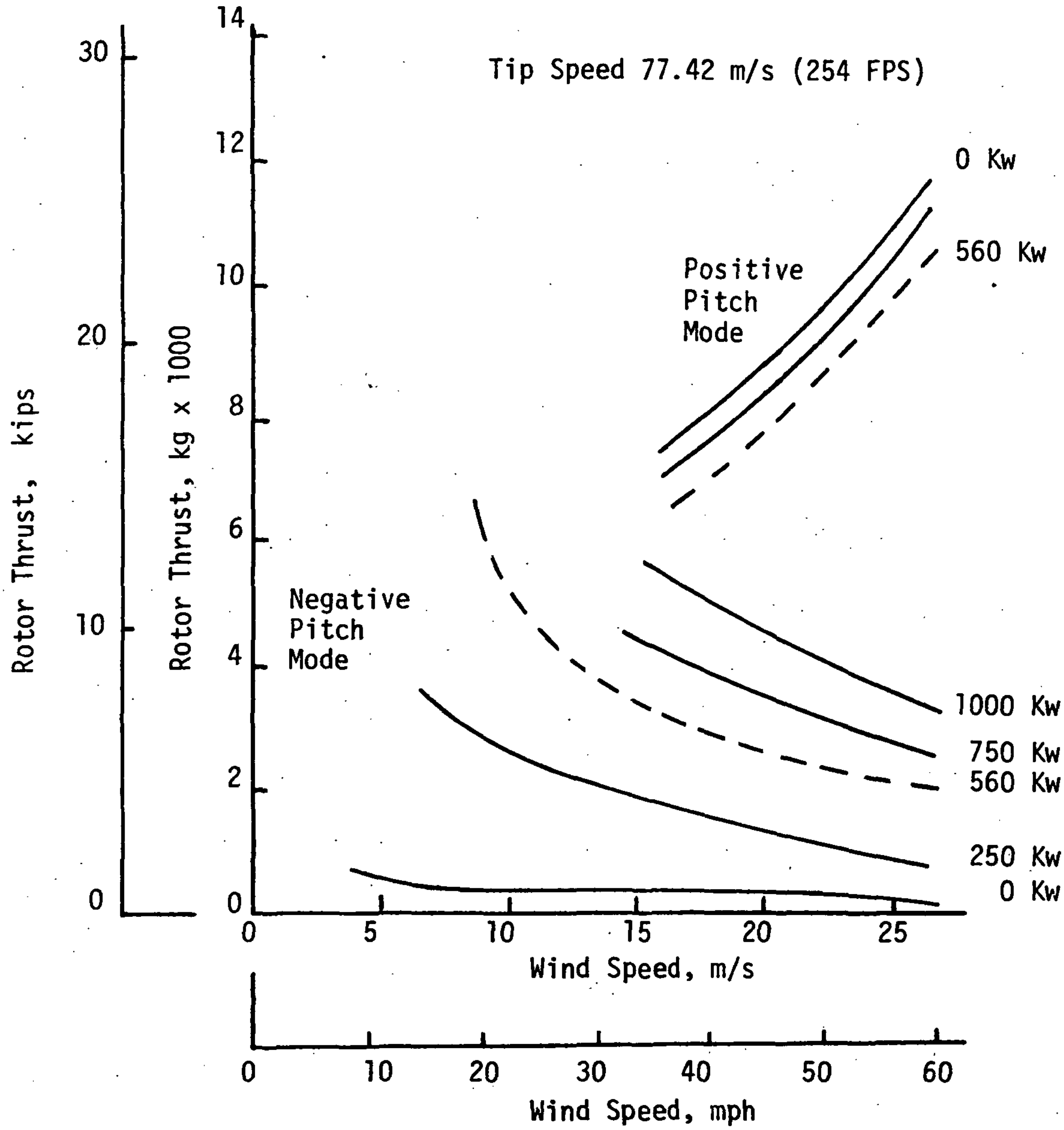


Figure 4-35. Rotor Thrust Contours, 500 Kw System

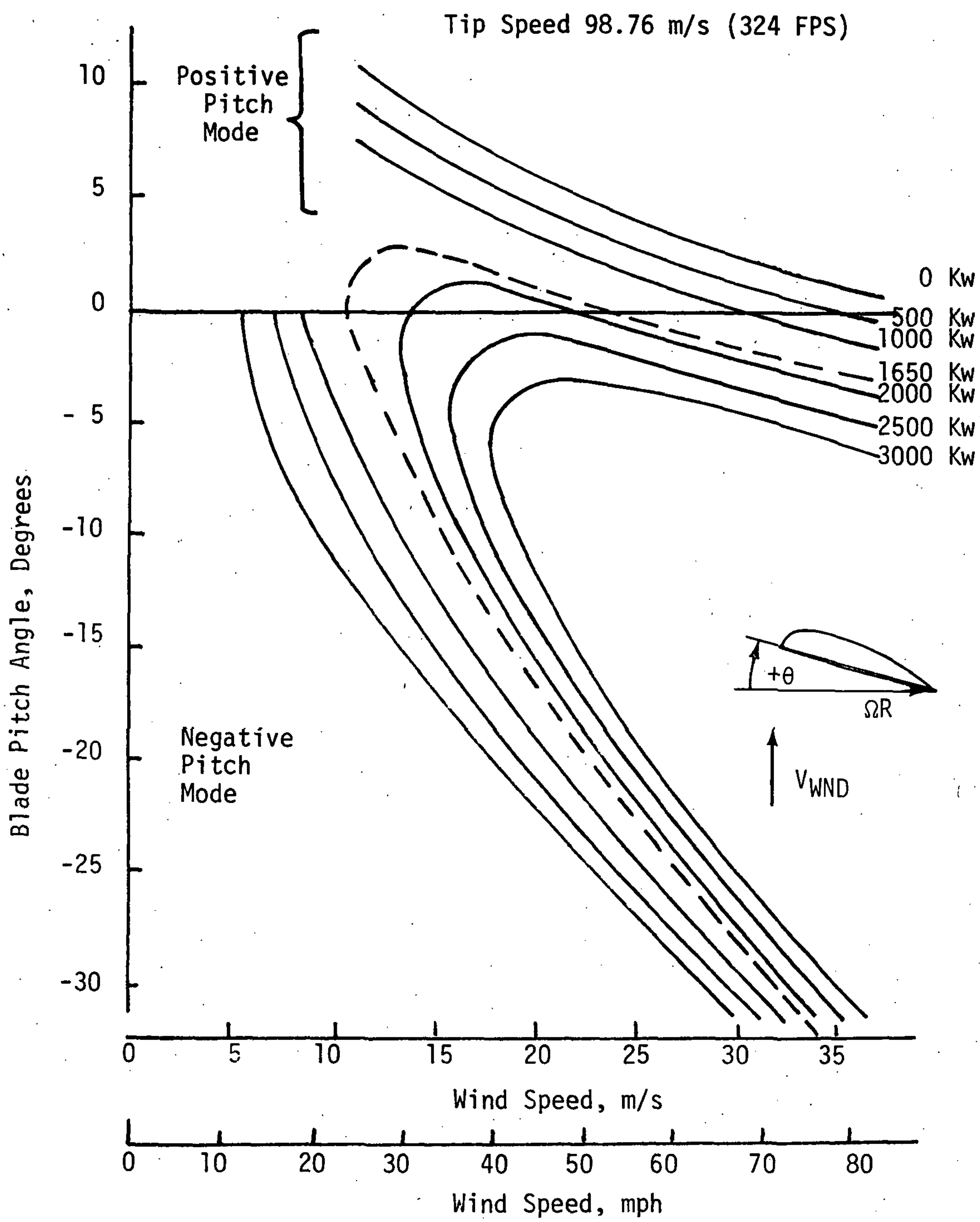


Figure 4-36. Rotor Power Contours, 1500 Kw

Tip Speed 98.76 m/s (324 FPS)

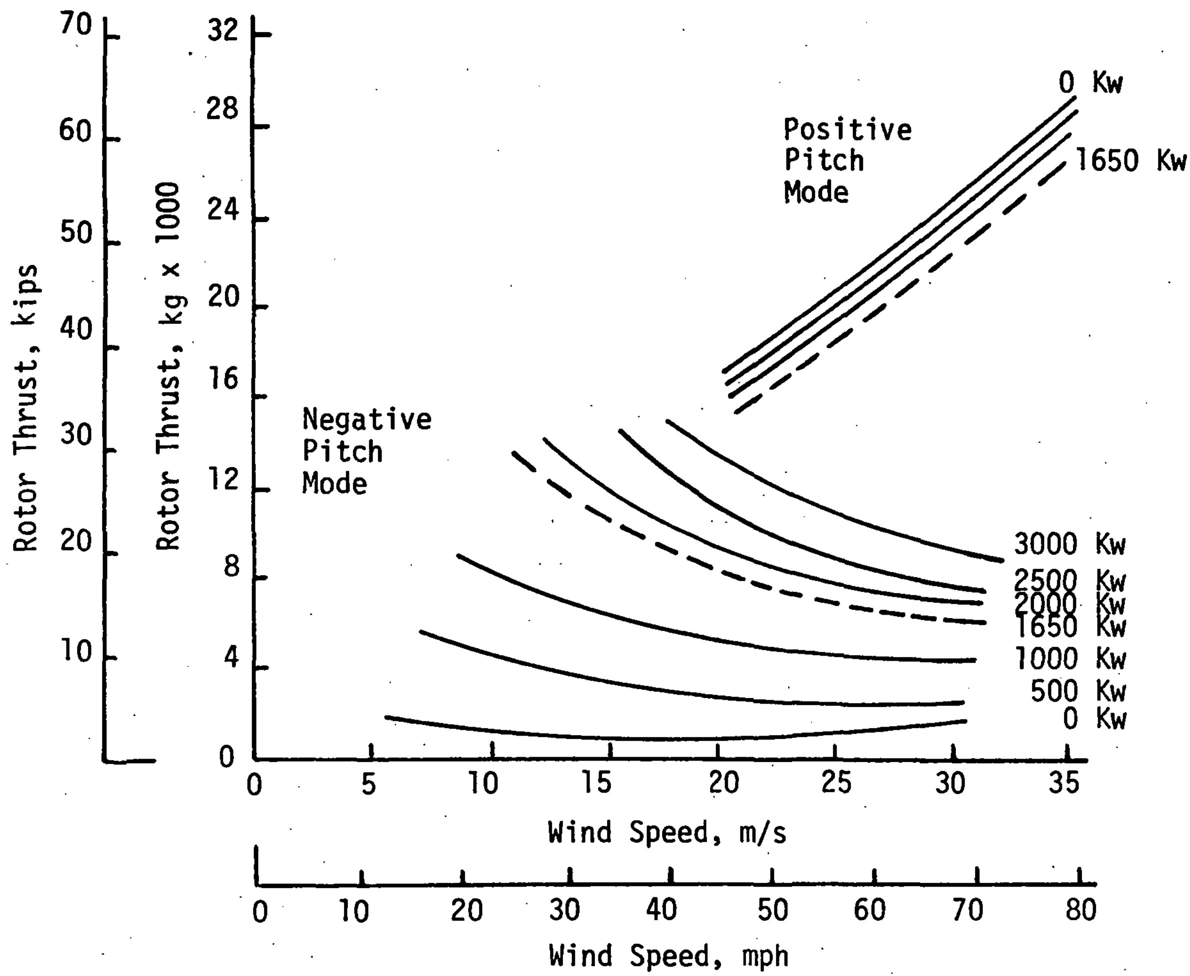


Figure 4-37. Rotor Thrust Contours, 1500 Kw

Rotor overspeed beyond the design rpm for normal operation can occur during large and rapid wind velocity changes associated with strong gust conditions, when the generator is abruptly disconnected from the network, or in the event of rotor control system malfunction. Loss of the balancing torque load imparts a net accelerating torque on the rotor, causing overspeed, as shown in Figure 4-38. The solid lines are constant wind speed curves showing equilibrium rotor tip speed vs blade pitch angle, and the dashed lines show the associated rotor power.

Rotor speed and torque control are accomplished by blade pitch changes responsive to wind velocity changes. A moderate rate of change of blade pitch angle, 5 degrees per second, was selected to provide the necessary overspeed and over torque control at rated wind speed and above. At wind speeds below rated, wind velocity changes produce torque changes at constant rotor speed. A blade pitch change rate of 5 degrees per second is sufficient to insure that the maximum rotor overspeed under the worst combination of conditions is not greater than 150 percent of rated rpm, the structural limit of the generator.

Peak gust amplitude for the rotor overspeed analysis was determined from a gust model supplied by NASA which shows amplitude variation as a function of gust period and steady wind velocity. The two steady operating conditions for gust application considered to be most critical for the rotor overspeed criteria were: (1) the rated wind speed condition at which maximum steady power is achieved, and (2) the cut-out wind speed condition which represents the maximum operational power potential of the rotor system. The 1500 KW WGS was analyzed for both these conditions to establish the relative severity of the gust response in terms of control system requirements and rotor overspeed. The rotor blade pitch change rate was evaluated for its ability to limit the maximum rotor speed reached during the transient response.

For the 1500 KW system, the rated wind speed is 11.2 m/s (25 mph) at the 9.1 m reference altitude. A gust amplitude of 4.5 m/s (10 mph) was added which corresponds to a gust period of about 2 seconds. At the cut-out wind speed of 20.1 m/s (45 mph), the same period gust has an 8.9 m/s (20 mph) amplitude. Assuming that blade pitch change remained fixed at steady state operating conditions until the full gust velocity was achieved, equivalent to a step-function gust input, the initial accelerating rotor torque was that corresponding to the fixed pitch setting at the higher velocity condition. For the 1650 KW (rotor torque) rated condition at zero blade pitch angle, the 4.5 m/s gust velocity increment increases rotor power to 2200 KW, giving the rotor an initial angular acceleration of 0.9 radians/sec^2, assuming simultaneous loss of electrical load due to circuit breaker opening.

At the cut-out wind condition, the steady state blade pitch angle setting is - 17° for rated power. The 8.9 m/s gust velocity increment at this pitch angle raises the rotor power output capability to 5000 KW, resulting in an initial angular acceleration of 2.1 radians/sec^2. Although the inital acceleration is more than twice as large for this higher velocity case, the accelerating torque vs pitch angle varies by a factor of three. Thus, the same pitch change rate, 5 degrees per second, is more effective in controlling rotor speed at the higher wind speed conditions.

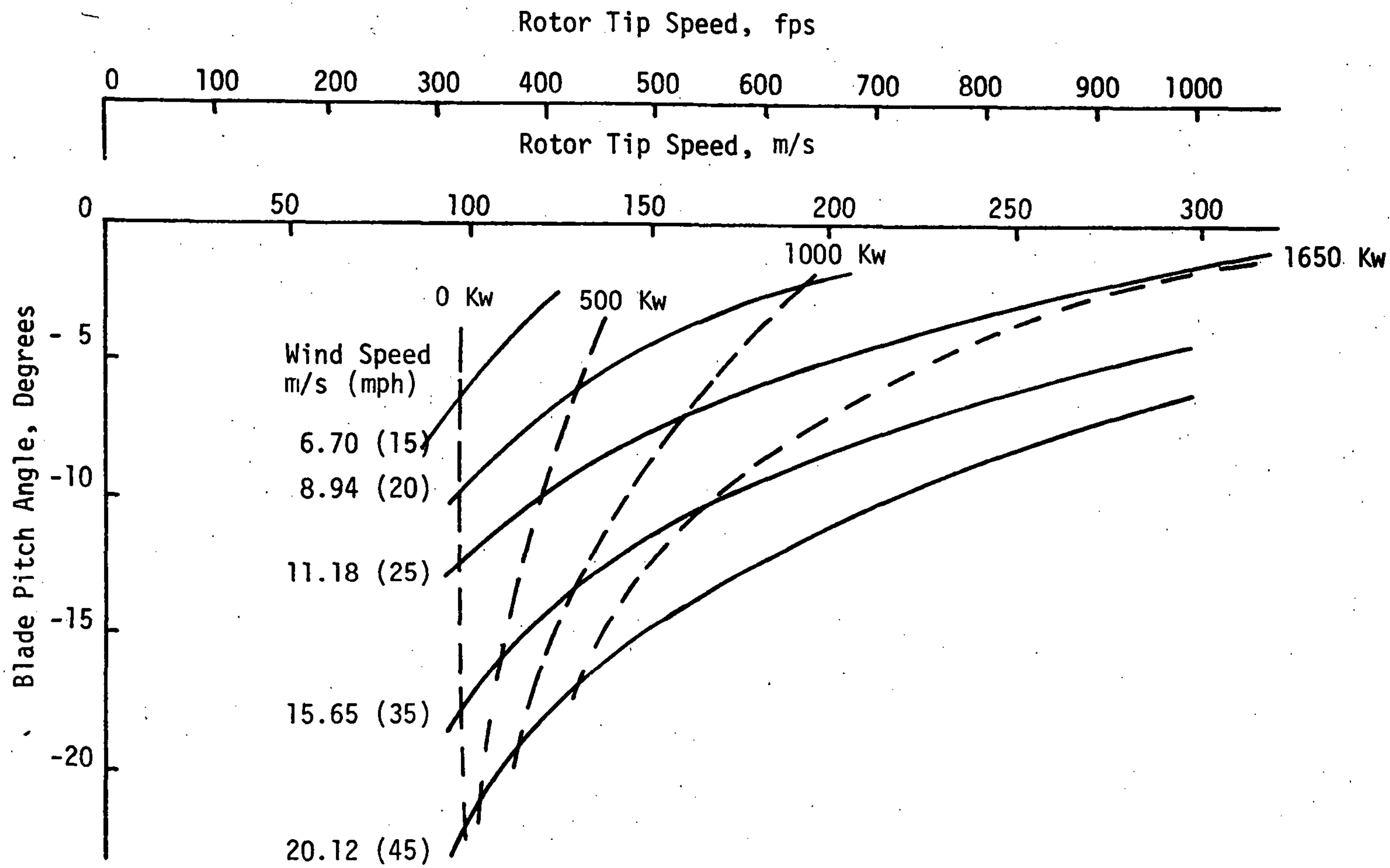


Figure 4-38. Equilibrium Rotor Tip Speed at No Load

Previously developed performance maps showing rotor power as a function of wind speed, pitch angle and rpm were used in the rotor overspeed analysis. Although the wind velocity was assumed to be constant during the overspeed period, the interdependence of rotor power, rpm and pitch angle necessitated an iterative solution of the single degree of freedom equation of motion of the rotor rotational mode. A pitch change versus rpm relationship was selected, and the rotor angular acceleration was developed from the performance map of the rotor system.

The rpm time history was determined from a numerical integration of the equation of motion, and the corresponding pitch rate variation was compared with the 5 degree per second rate. The procedure was repeated until convergence was obtained at the 5 degree per second rate.

Sample time histories of rotor overspeed characteristics for the 1500 KW WGS are presented in Figure 4-39 for a 5 degree per second blade pitch control input at rated wind and at cut-out wind conditions. Also shown for comparison is a 15 degree per second pitch change rate at rated wind. In all cases, the peak rpm was less than 150 percent of rated rpm, and the maximum rpm occurred within 1 to 2 seconds after load application from the assumed gust condition. Rotor overspeed as a function of pitch control rate is shown in Figure 4-40.

The time variation of velocity specified in the NASA gust model is a cosine function with a two second period, for the values of gust amplitude considered in this analysis. A step function is considered sufficiently conservative, by comparison, to account for system lag and blade feathering inertia effects which would produce a more nonlinear blade pitch change response to control application.

For the 500 KW WGS, the overspeed characteristics were compared with the 1500 KW system on the basis of the initial rotational acceleration resulting from the gust input, plus loss of the balancing rated power electrical generator load. For the smaller WGS, the rated velocity is 8.9 m/s and the cut-out velocity is 13.4 m/s (30 mph). For a 4.5 m/s gust at rated condition, the initial angular acceleration before pitch angle change is 0.64 rad/sec^2, or only 71 percent of that corresponding to the 1500 KW WGS at rated conditions with the same gust velocity amplitude. At the cut-out speed with a 6.7 m/s (15 mph) gust amplitude and release of electrical generator load, the initial angular acceleration is 1.3 rad/sec^2 or 63 percent of the value for the 1500 KW WGS at its cut-out velocity condition. Thus, with a pitch change rate of 5 degrees per second, this smaller WGS will also limit rotor overspeeds to less than 150 percent of rated rpm.

Operational rotor overspeeds for helicopters are usually limited to approximately 110 percent of normal operating rpm, as a result of the high centrifugal forces involved, although rotors must be structurally capable of withstanding overspeeds up to 125 percent of design rpm. The WGS rotors, on the other hand, operate at such low centrifugal force levels that overspeeds do not present significant structural limitations. The 150 percent overspeed limit discussed in this analysis is based upon an assumed structural limit of the electrical generator.

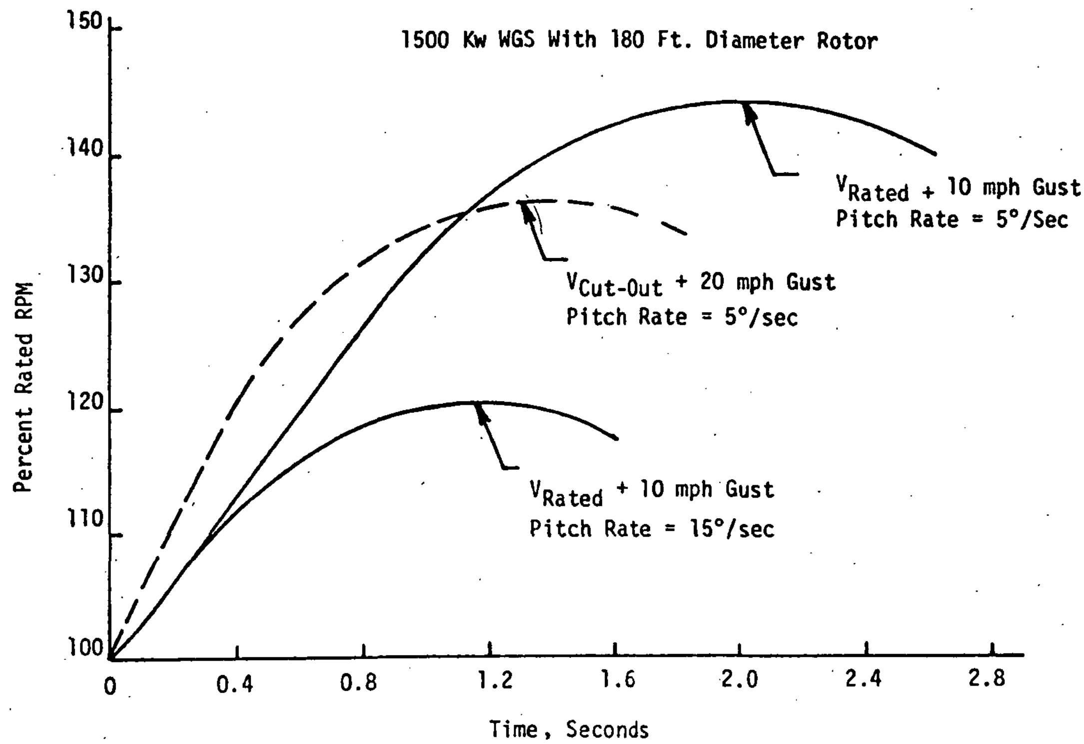


Figure 4-39. Rotor Overspeed vs. Wind Velocity and Pitch Change Rate

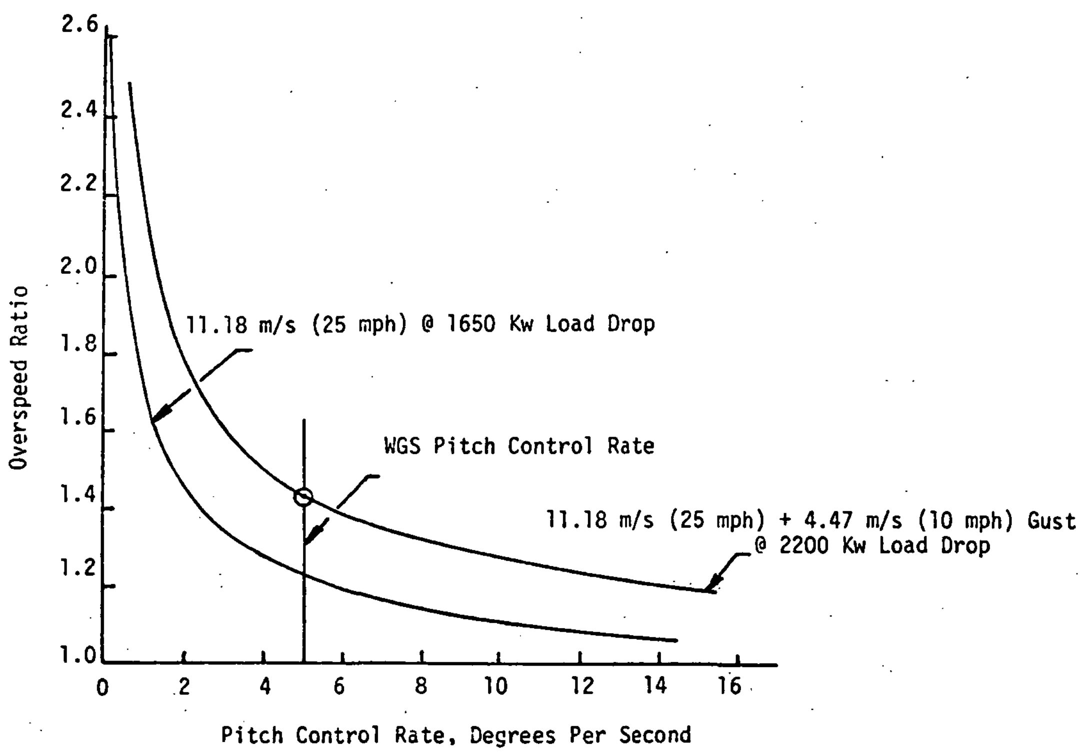


Figure 4-40. Rotor Overspeed Analysis, 1500 Kw System

4.5.2.4 Blade Tuning

The structural configuration and thickness distribution of the WGS rotor blades were dictated largely by blade tuning requirements. Of particular concern in the overall blade tuning problem is the need to avoid resonance crossings during rotor start-up in marginal wind conditions. Under such conditions, rotor acceleration to normal operating rpm will be quite slow, particularly in the 80 to 100 percent rpm range, where accelerating torques are low. Occurrence of a resonant mode just below the operating rotor speed could cause significant amplification of vibratory bending moments for a considerable length of time. Further, the resonant mode would be encountered on each rotor start-up and shutdown, although the latter would be of short duration.

It is, therefore, considered essential that the lowest blade bending frequency be above 2/rev rotor speed to avoid amplification of vibratory blade bending moments and tower loadings of two-bladed WGS rotors. Similarly, it is considered essential that blade first mode in-plane and out-of-plane natural bending frequencies be separated by two harmonic orders (2 Ω), to avoid intermodal coupling, dynamic amplification and possible stability problems.

Unlike helicopter rotors, WGS rotors have very little centrifugal stiffening as a result of their low rotor speed. This is evident in the Campbell diagrams (fan plots), shown in Figures 4-41 and 4-42, where very little increase in natural frequency occurs with increased operating frequency. These plots also show that blade resonances will occur during rotor overspeed, although such occurrences will be infrequent and of short duration. Associated bending mode shapes are shown in Figure 4-43.

These requirements resulted in blade configuration details such as 40 percent airfoil thickness at the blade root, unidirectional S-2 fiberglass stiffening, a trailing edge spline and aluminum honeycomb panels for the airfoil afterbody. These structural configuration details are discussed further in paragraph 4.5.2.7.

4.5.2.5 Blade Flutter and Divergence Boundaries

Blade flutter and divergence boundaries for the WGS hingeless rotor were defined using rigid blade coupled flap torsion equations developed by Miller and Ellis in Reference 4-3. The equations were modified to include coupling terms involving blade coning and collective pitch angles, as shown in Equations 1 and 2.

The first out-of-plane bending mode was simulated in the analysis by selecting an equivalent flapping hinge offset which approximates the mode shape, and an equivalent flapping spring which yields the correct natural frequency. The effective blade flapping inertia used in the flutter equations was thus established. Blade feathering inertia and product of inertia were calculated directly from the mass distribution.

No provisions for in-plane motions were included in this two degree-of-freedom analysis. It is recognized that flap-lag-torsion coupling may influence stability of the highly-twisted WGS blade; however, such coupling effects are

500 Kw System

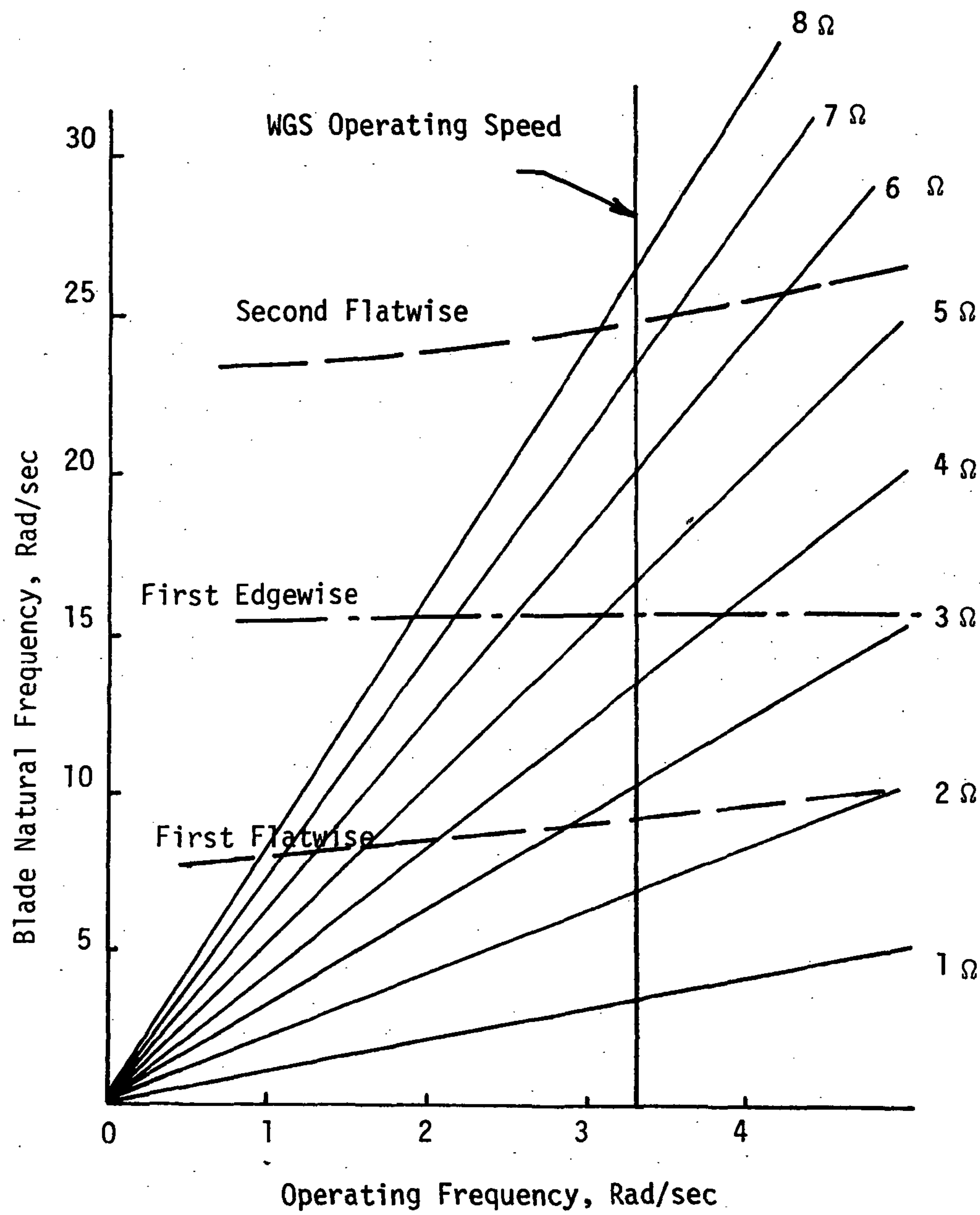


Figure 4-41. Blade Natural Bending Frequencies

1500 Kw System

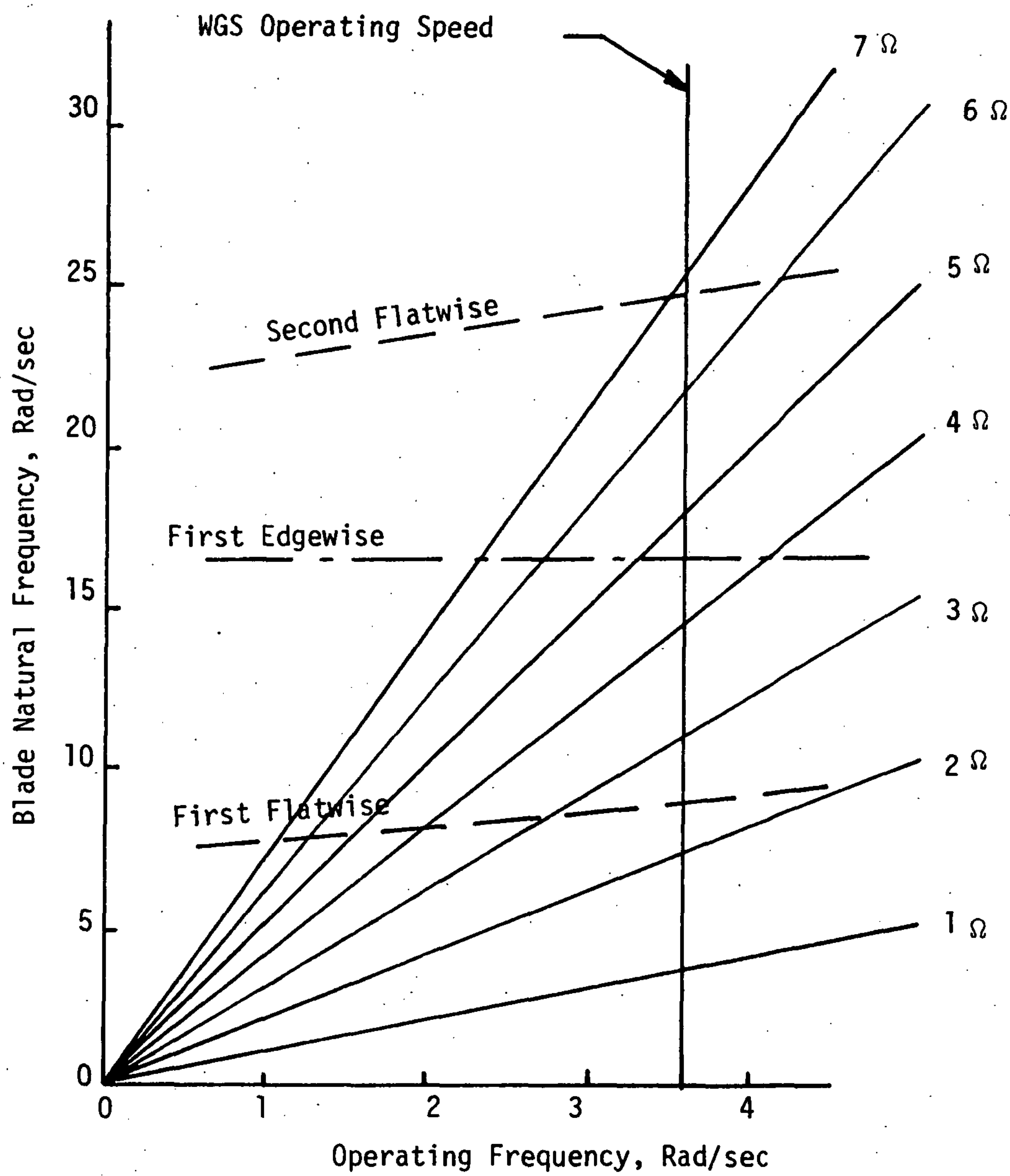


Figure 4-42. Blade Natural Bending Frequencies

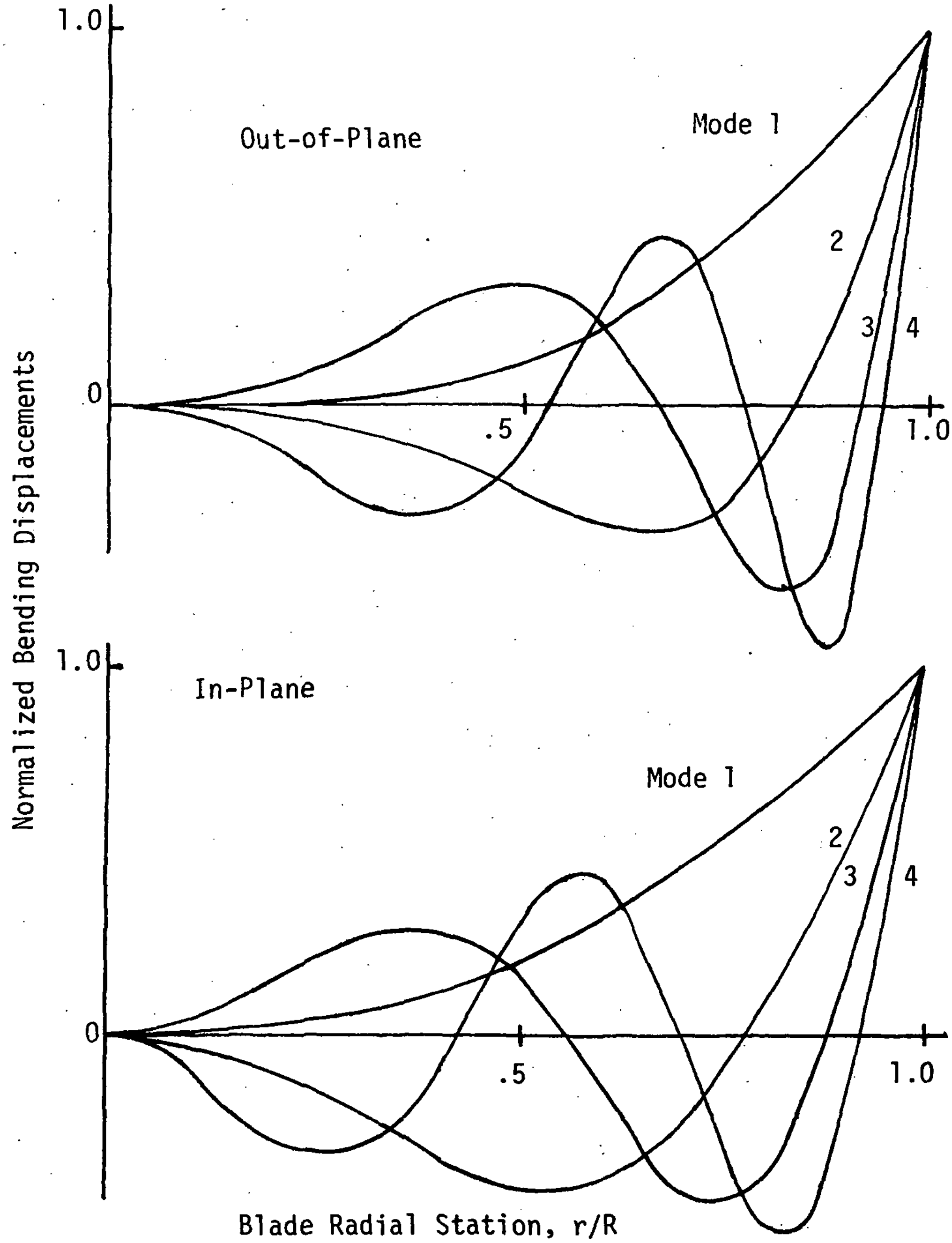


Figure 4-43. Blade Bending Mode Shapes (Normalized)

probably small because the in-plane and out-of-plane natural frequencies are well separated, as shown in paragraph 4.5.2.4. The lowest WGS torsional natural frequency ratio, $\frac{\omega_\theta}{\Omega}$, is approximately 27 per rev, which would effectively decouple torsion from the fundamental bending modes.

The blade feathering axis is on the quarter chord, therefore, $X_A = 0$ in this analysis. Also, $\mu = 0$ for WGS operation.

For this preliminary analysis, unsteady aerodynamics usually represented in classical flutter analysis by the Theodorsen lift deficiency function, $C(k) = F + iG$, were replaced by the quasi-steady aerodynamic approximation, $C(k) = 1$. This is a reasonable assumption because the reduced frequency, k, is of the order .02.

The resulting coupled flap-torsion equations of motion that were used are given by:

$$\frac{\ddot{\beta}}{\Omega^2} + m_{\frac{\dot{\beta}}{\Omega}} \frac{\dot{\beta}}{\Omega} + \beta - I_x \frac{\ddot{\theta}}{\Omega^2} + (2\beta_0^{\,2} + m_{\frac{\dot{\theta}}{\Omega}}) \frac{\dot{\theta}}{\Omega} + (m_\theta - I_x)\theta = 0 \qquad (1)$$

$$- I_x \frac{\ddot{\beta}}{\Omega^2} + (M_{\frac{\dot{\beta}}{\Omega}} - 2\beta_0^{\,2}) \frac{\dot{\beta}}{\Omega} + (M_\beta - I_x)\beta + (I + \beta_0^{\,2}) \frac{\ddot{\theta}}{\Omega^2}$$
$$+ M_{\frac{\dot{\theta}}{\Omega}} \frac{\dot{\theta}}{\Omega} + \{I[1 + (\frac{\omega_\theta}{\Omega})^2] - \beta_0^{\,2} + M_\theta\}\theta = 0 \qquad (2)$$

where:

$$I_x = \frac{I_{\beta\theta}}{I_{\beta\beta}} = \frac{\int_0^R mrx_I \, dr}{\int_0^R mr^2 dr} = \frac{\text{Product of Inertia}}{\text{Flapping Inertia}}$$

$$I = \frac{I_{\theta\theta}}{I_{\beta\beta}} = \frac{\int_0^R mx_I^{\,2} dr + I_0}{\int_0^R mr^2 dr} = \frac{\text{Feathering Inertia}}{\text{Flapping Inertia}}$$

$$m_\theta = m_{\frac{\dot{\beta}}{\Omega}} = \frac{\gamma}{8}$$

$$m_{\frac{\dot{\theta}}{\Omega}} = - \frac{\gamma}{8} \frac{c}{R}$$

$$M_\theta = M_{\frac{\dot\beta}{\Omega}} = -\frac{\gamma}{6}\left(2\lambda\beta_0 - \frac{3}{4}\beta_0\theta_0\right)$$

$$M_\beta = \frac{\gamma}{2}\left(\frac{C_{D0}}{4a} + \lambda\left(\frac{\theta_0}{3} - \frac{\lambda}{2}\right)\right)$$

$$M_{\frac{\dot\theta}{\Omega}} = \frac{\gamma}{16}\left(\frac{c^2}{2R^2}\right) + \frac{\gamma}{2}\left(\frac{1}{3}\theta_0\beta_0 - \lambda\beta_0\right)\frac{c}{2R}$$

$$\gamma = \frac{\rho acR^4}{I_{\beta\beta}} = \text{Lock number}$$

a = lift curve slope = 2π

c = blade chord

X_I = blade section cg location, positive when forward of the feathering axis

$$\lambda = \frac{Vw}{\Omega R} = \frac{\text{wind velocity}}{\text{tip speed}} = \text{inflow ratio, negative for WGS}$$

$$k = \frac{\omega c}{2V_{tip}} = \text{reduced frequency}$$

Using assumed solutions:

$$\beta = \bar\beta e^{\nu\Omega t}, \qquad \theta = \bar\theta e^{\nu\Omega t}$$

equations (1) and (2), with two unknowns ω_θ and I_x can be written in matrix form as:

$$[M]\left\{\frac{\beta}{\theta}\right\} = 0$$

The flutter determinant [M] must vanish. Expanding the determinant of [M], we obtain an equation of the form:

$$A\nu^4 + B\nu^3 + C\nu^2 + D\nu + E = 0$$

The static divergence boundary is determined by setting $\nu = 0$ and solving the resulting equation $E = 0$. The flutter boundary is defined by the Routh Discriminant:

$$BCD - AD^2 - B^2E = 0$$

which contains unknowns ω_θ and I_x. Solving the equation of coefficients results in an equation of the form:

$$a_4 \omega_\theta^4 + a_2 \omega_\theta^2 + a_0 = 0$$

in which the coefficients a_4, a_2 and a_0 are functions of I_x. Selecting values of I_x yields ω_θ at which the flutter mode is neutrally stable, i.e., the flutter boundary.

Results of this analysis are plotted in Figures 4-44 and 4-45 as $\frac{\omega_\theta}{\Omega}$ vs I_x. The latter was selected as the independent variable, rather than X_I, to include the span-weighted effect of the non-uniform mass distribution. For the 54.9 m (180 ft) diameter WGS rotor, I_x = - .0033, and $\frac{\omega_\theta}{\Omega}$ = 34.5 per rev. For the 45.7 m (150 ft) diameter rotor, I_x = - .019 and $\frac{\omega_\theta}{\Omega}$ = 26.8 per rev. This analysis shows both WGS rotors to be free from flutter and divergence for the rated operating conditions examined. However, note that the 500 KW rotor is comparatively less stable than the 1500 KW rotor as a result of a more rearward cg distribution, shown in Figures 4-30 and 4-32.

The WGS flutter boundary differs from conventional helicopter flutter boundaries, such as illustrated in Reference 4-3, as a result of negative inflow direction associated with WGS operation. This characteristic was verified by reversing the sign of the inflow parameter, λ, in proven flutter equations for a typical helicopter main rotor blade; the result is shown in Figure 4-46. Examination of the aerodynamic coefficients in the pitching moment equation, [Equation (2)] reveals the de-stabilizing influence of negative inflow, particularly in the coefficients of θ and $\dot{\beta}$, which represent feathering stiffness and flap damping, respectively.

Although the simplified flutter analysis described herein is adequate for preliminary design investigation of the WGS rotor system, a more complete analysis will be required during the detail design phase. Specifically, the following additional factors should be included:

1. The lead-lag degree of freedom should be added to the analysis to include flap-lag-torsion coupling of the highly-twisted WGS blade. The fundamental in-plane bending mode shape should be represented in the same manner as in the flapping equation.

2. Steady in-plane and out-of-plane bending displacements should be included, in addition to the built-in twist distribution.

3. Coupling effects of gravity, rotor shaft tilt and wind shear gradient should be evaluated for their potential influence on blade stability and dynamic response. Blade transient response to perturbations from the tower wake should also be investigated.

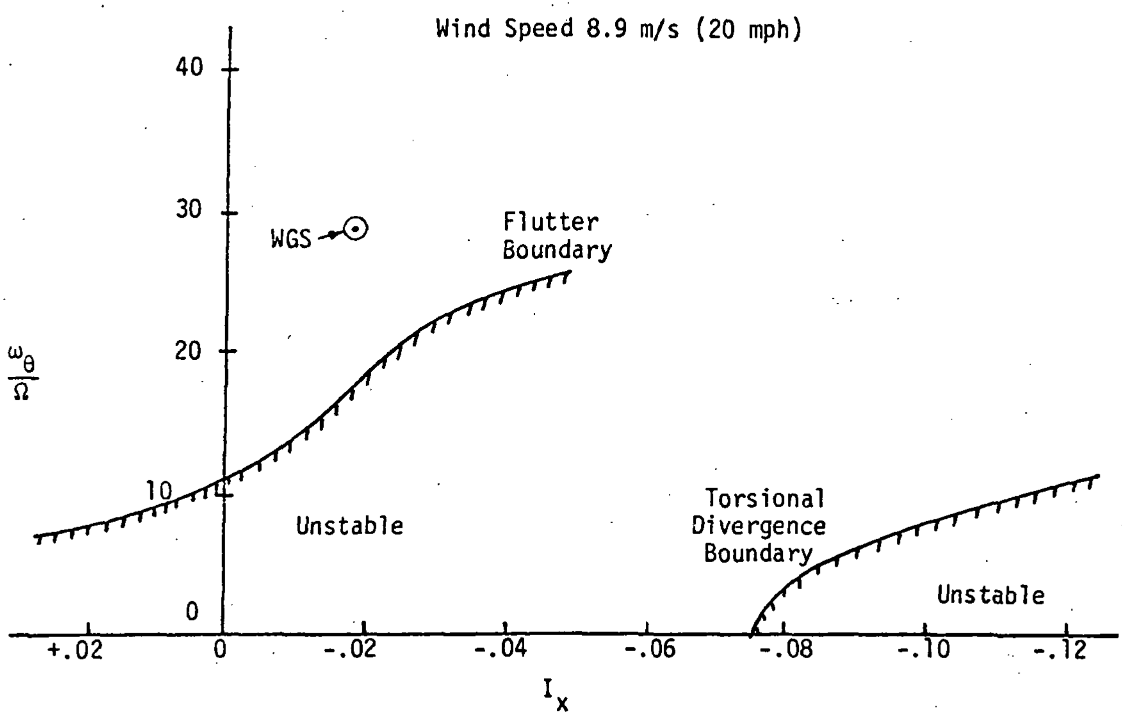


Figure 4-44. 500 kw WGS Flutter and Divergence Boundaries

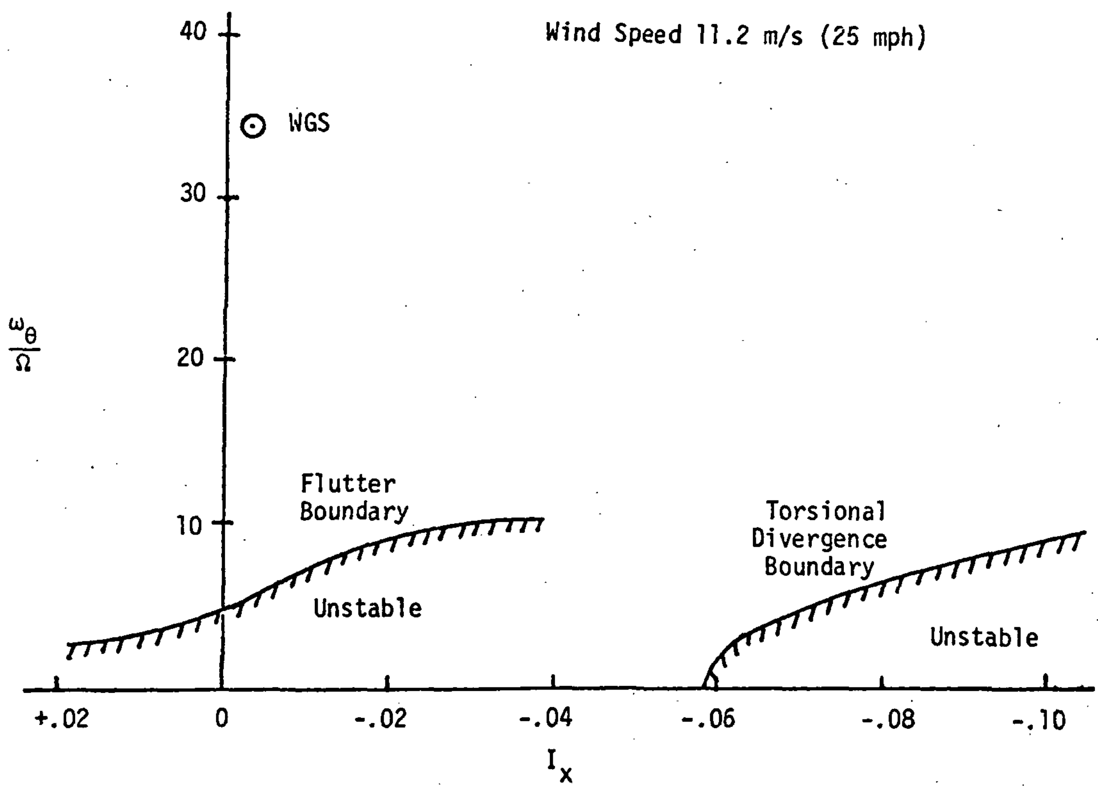


Figure 4-45. 1500 kw WGS Flutter and Divergence Boundaries

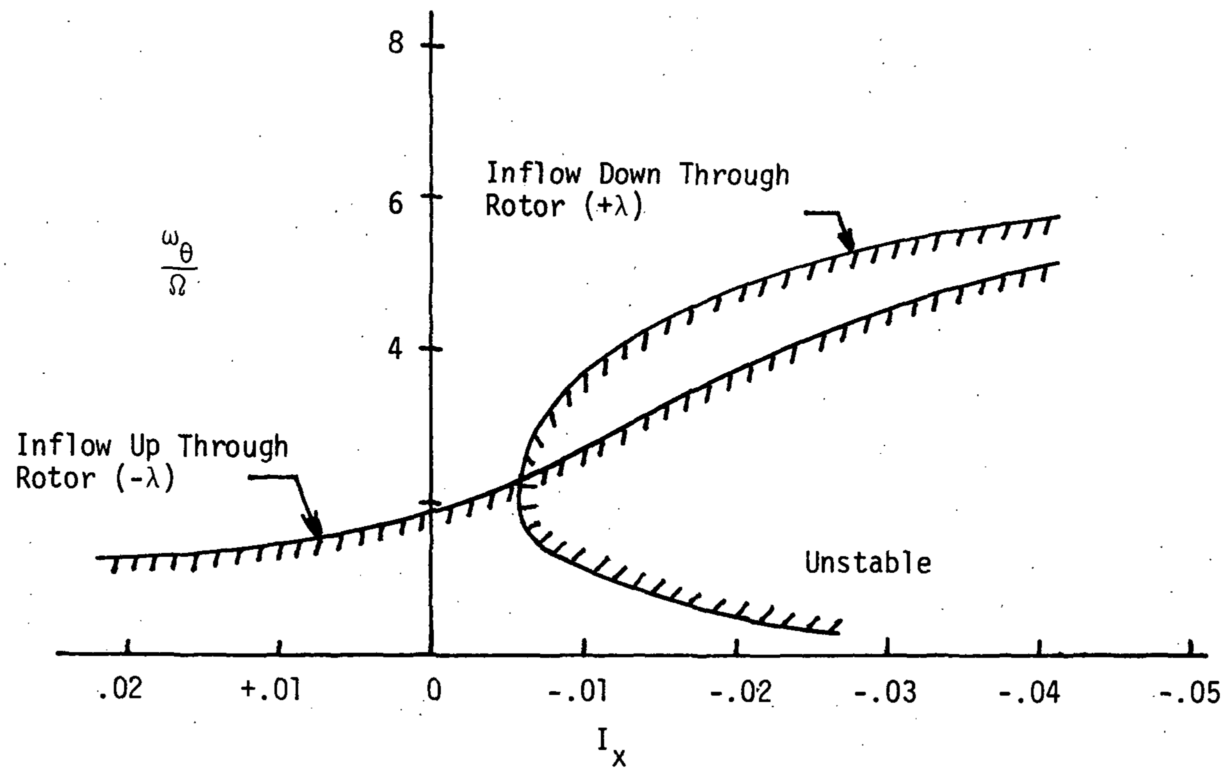


Figure 4-46 Effect of Inflow Direction on Main Rotor Blade Flutter for a Conventional Helicopter Rotor

4. Additional cases, representing different operating conditions, should be analyzed, particularly rotor overspeed at maximum wind speed, and rotor underspeed during start-up.

5. Possible coupling effects of higher harmonic blade bending and torsion modes should also be investigated during the detail design and analysis phase.

4.5.2.6 Whirl Resonance Analysis

The stability margins of the two-bladed WGS rotor/tower configurations were investigated for the following whirl modes, using the techniques outlined in Reference 4-2:

a. Critical shaft whirl speed range

b. Resonances excited by a steady force

c. Self-excited vibrations due to coupling between the blades and hub.

A two-bladed rotor differs from other multi-bladed rotors in that it has a range of critical whirl speeds, rather than a unique speed. The critical upper limit for an undamped system is the natural frequency of the total system, rotor and tower, in which the blades are treated as concentrated mass at the hub. The lower limit of the unstable band is dependent upon the ratio of the hinged blade mass to the system total mass. Since the WGS rotor features hingeless blades, it was necessary to calculate the fundamental in-plane natural frequency and mode shape so that a virtual hinge location could be established. The effective mass of the tower and hub was obtained by modeling the tower structure in a coupled-modes digital computer program with various weights at the apex of the tower. These weights simulated the WGS rotor, drive and generator systems. The computer program used was the "RASA" modes program developed for NASA Langley by Rochester Applied Sciences Associates.

Equation (3) was used to calculate the critical shaft whirl speed range of the classical "flywheel resonance" mode:

$$\left[\left(1 - \left(\frac{\Omega}{\omega}\right)^2\right)\left(\Lambda_2 + \Lambda_1\left(\frac{\Omega}{\omega}\right)^2\right) - 2\Lambda_3\left(\frac{\Omega}{\omega}\right)^4\right]\left[1 - \left(\frac{\Omega}{\omega}\right)^2\right] = 0 \qquad (3)$$

where:

$$\Lambda_1 = \frac{abm_b}{I_b}, \qquad \Lambda_2 = \left(\frac{\omega_0}{\omega}\right)^2, \qquad \Lambda_3 = \frac{nb^2 m_b^{\,2}}{2MI_b}$$

a = hinge radius (effective lag hinge)

b = distance from hinge to blade cg

m_b = mass of one blade

M = total effective mass of blades and pylon

I_b = moment of inertia of the blade about its virtual hinge

ω_0 = static frequency of the blade about its virtual hinge

ω = pylon frequency with mass M

Ω = rotor speed

n = number of blades

The real root of the first factor gives the lower boundary of the shaft critical whirl speed while the second term is the system natural frequency and the upper boundary of whirl speed. For typical WGS rotor designs, the critical speed range is a very narrow band because the blade hinged mass is a small percentage of the system total mass. This is attributed to the high in-plane stiffness of the rotors and the resulting large virtual hinge offset. The lowest "flywheel resonance" frequency for the 500 KW rotor is 4.28 per rev, and for the 1500 KW rotor, 2.53 per rev, well above rotor operating frequencies.

The second instability region identified was the resonance excited by the steady gravitational force, which produces a one per rev excitation in the rotating system. The dynamic response affects the magnitude of loads applied to the WGS rotor shaft and can be a factor in its fatigue life, if operated near resonance. The roots of Equation (4) represent the two system resonance frequencies associated with the gravity force. The lowest gravitational mode frequency for the 500 KW system is 2.14 per rev, and for the 1500 KW system is 1.26 per rev. These are sufficiently far removed from blade in-plane response frequencies for minimal amplification of shaft loads.

$$\left[\Lambda_1\left(\frac{\Omega}{\omega}\right)^2 + \Lambda_2\right]\left[4\left(\frac{\Omega}{\omega}\right)^2 - 1\right] - \left(\frac{\Omega}{\omega}\right)^2\left[\left(\frac{\Omega}{\omega}\right)^2(4 - 16\Lambda_3) - 1\right] = 0 \qquad (4)$$

The third instability region investigated is the classical helicopter mechanical instability mode, which results from a coupling between hub displacement motions and blade chordwise vibrations. In this instability range, the rotor system undergoes self-excited vibrations. Charts are presented in Reference 4-2 giving both boundary points of the instability region for a variety of parameters. The position of the instability region is very sensitive to the value of Λ_3. As Λ_3 increases, the region of instability occurs at higher rotational speeds. No region of instability exists for the 500 KW rotor; for the 1500 KW rotor, the boundary is located at 9.5 per rev, well above the maximum operating rotor speed.

This whirl resonance analysis utilized the preliminary design information available on the WGS tower and rotor head supporting structure. It was necessary to estimate the effective stiffness of the turntable bearing assembly, described in Section 7, to determine the natural frequency and effective mass of the tower and hub. It was assumed that displacements at the rotor hub were 50 percent greater than those attributable to elastic deflections of the tower alone. This assumption must be reviewed during the detail design phase to insure that tower frequencies and stiffnesses are properly represented.

4.5.2.7 Structural Analysis

Blade Design - Structural proportioning of the WGS blades started when the planform size and shape had been derived by the performance analysis. Airload shear and moment diagrams were prepared for the 53.6 m/s (120 mph) [at 9.1 m (30 ft) reference altitude], non-rotating case. These were used to make a tentative selection of EI distributions, which were converted to wall thickness and member sizes by selecting an allowable extreme fibre stress. These data made possible the calculation of a blade mass distribution. A check of frequencies showed this design to be tuned to excessively low first mode frequencies in both planes. A few trials with increasing rigidities showed that increases of approximately six times the beam sections as sized for the static wind strength would be required to approach the tuning requirement. To achieve this large increase in stiffness with reasonable economy, it was found necessary to exaggerate the orthogonality of the composite spar material by increasing the proportion of radially oriented unidirectional material to 2/3 of the total spar material, and also to radically increase the blade thickness ratio to 40% at the root. A further change was necessary to raise the in-plane first mode frequency to the desired value, since the spar alone, even with a width equal to 50% of the local chord, did not supply enough rigidity. The desired increment was achieved by adding a unidirectional glass bar at the trailing edge, connected as extreme fibre material via the skins for a running web. The glass bar, or spline, was terminated at the root by a simple diagonal truss, constituted by a drag link and a rib. These stiffness adjustments resulted in a design which could be adjusted to the desired tuning with only small further refinements of outboard stiffness and reduction of outboard masses.

Comparative loads analysis showed that the highest shears and moments were encountered for the 53.6 m/s (120 mph) case with each section generating maximum normal force. Although the direct stresses in the extreme fibre of the beam were not critical for this (or any other) case, the shears near the root were found to be critical. The three shear webs near the root have near zero margins based upon strength. Material properities for this analysis were generated by Kaman's CMEX composite materials physical properties program, which produces a reliable estimate of the strength and elastic properties of composite materials of arbitrary layup, working with the measured properties of single layers.

Elastic stability of the shell was considered to be especially important for a large rotor blade structure of this type for which long life is required. The hazard foreseen was that structural instabilities (even elastic buckling) might cause crazing of the resin in the composite matrix, due to secondary bending stresses, especially near the corners of the cells. If such crazing occurred, it could be expected that the subsequent fatigue performance would be degraded. Accordingly, the conservative criterion was established that panel buckling would not be permitted within the limit strength envelope. The design shown meets this criterion.

The composite rotor blade which resulted from the procedure described is a three cell beam, with vertical (perpendicular to chord) webs at 25% and 50% chord at the root, oriented radially so that the mandrels about which the individual cells are filament wound in fabrication can be withdrawn. This geometry places the last web considerably aft of the 50% chord line at the tip. The forward web is located at the nominal line of aerodynamic centers of the sections, and since the blade has zero sweep, that web maintains quarter chord position full span. The wall thicknesses of the three cells are graded in thickness, the forward cell being thickest and heaviest and the aft cell having lightweight sandwich walls. The skin is wound over all three cells and forms the outer surface of the aft blanket. The design is intended to make full use of the capability of the filament winding process to mechanically layup large quantities of fibrous composite.

The process starts with the independent pre-winding of material on the mandrels which form the two forward cells. Then, the two mandrels are mounted together for a helical overwind, which ties them together. At this point, unidirectional material is added manually, with mechanical assist, along with the pre-wound root end lug details, which are integral with a portion of the unidirectional cap material. The spar is then completed and compacted by over-winds of helically oriented material.

The rotor blade afterbody is then fabricated. A mandrel is mounted with the spar, which still contains its mandrels. Sheets of prepared inner skin with honeycomb core (constant thickness) are then placed upon the mandrel with inner skin splice and attach strips placed to join the panels. A prepared spline is added at the trailing edge near the root. Finally, the skin is wound over the entire blade. Upon completion of the filament winding, the blade will be covered with layers of peel ply and bleeder cloth, bagged and oven-cured, with mandrels still inside. After cure, the mandrels will be removed by pulling from the root.

This process requires geometry which permits mandrel extraction and accounts for the positioning of the webs and shell wall material. A highly efficient structure is realized. The fabrication of such massive, highly tapered, contoured beams requires an extremely flexible process, and makes full use of the fabrication flexibility of fibrous composite materials.

Because of the large amounts of material, and the stringent economic requirements, only glass fiber seems to be a practical choice. Other fibers, though possessing desirable properties, are an order of magnitude higher in cost, and thus not practical. E glass was eliminated as possessing inadequate Young's modulus. S2 glass was found to meet all requirements.

Fatigue Analysis - A potential for the accumulation of fatigue damage in WGS rotors could occur in the following regions of operation, listed in order of severity of cyclic loads:

a. Power generation in steady winds at or below rated wind velocity.

b. Same as a., above rated velocity, where torque and power are limited by the action of the control system.

c. In unsteady winds, including strong gusts. (Response will be a function of control characteristics, and will be different in regimes a. and b.)

d. During transients, such as starting, stopping (especially during stops near furl speed in rising winds, undertaken to prevent exposure to excessive loads).

e. Events which do not usually occur often, but must be expected periodically during life of a unit. Sudden wind changes which cause over-torque and loss of electrical load due to circuit breaker operation. Malfunction of the control system. Momentary operation at high yaw angles in sudden gusts, until yaw actuation realigns the WGS.

f. Exposure to sudden winds above design limits while operating before shutdown can be effected. Exposure to high winds, with gusts and turbulence during severe storms while furled.

Consider that the "knee" of the S-N curve of engineering materials is practically at about 3.0×10^7 cycles, and that in continuous operation, the WGS can accumulate 3.0×10^7 cycles in one year. Since some thirty years operation would be a reasonable design life, equivalent to about 9×10^8 cycles life, it is apparent that almost all operation must occur below the endurance limit in order to achieve desired performance. Accordingly, a design goal was selected which would place the endurance limit above operating conditions a. through d., leaving the possibility of occurrence of fatigue damage only for the severe condition classes e. and f. Blade bending moments for the WGS rated wind conditions are shown in Figures 4-47 through 4-50.

Figures 4-51 through 4-54 show the calculated fatigue strength (unlimited life) as a function of blade radius for the 500 KW and the 1500 KW rotor blades, respectively, out-of-plane and in-plane. The same figures show predicted blade loads for 22.8 m/s (51 mph), and 34.0 m/s (76 mph) wind speeds for the 500 KW and 1500 KW systems, respectively. These speeds represent operation at a 68% gust factor above furl speed. The large margin shown between allowable and expected loads indicates that the design fatigue life will be achieved.

Hub Design - The basic philosophy of the hub design is to supply the most straightforward load paths for the primary loads while achieving a producible design. The hub is comprised of a large diameter welded tube, very heavily proportioned, relative to working loads and, thus, operating at low stress levels. Blades are attached via single row, large diameter "X" style roller pitch bearings at each end of the cylinder.

The structural functions of the hub are to provide bending continuity between the two opposite blades, and to react one per rev and higher order hub moments and rotor torque to ground, through the rotor shaft load path. These goals are achieved using the shortest possible paths for each primary load. Hub moment is reacted directly to the non-rotating shaft via bearings mounted within the hub walls. The shaft accepts hub moment essentially as radial shears at the two forward and aft hub bearings, the pair generating a couple which, in the shaft reference system (stationary), changes one per rev moments, which are very large, to steady loads, thus minimizing the fatigue problem for the shaft.

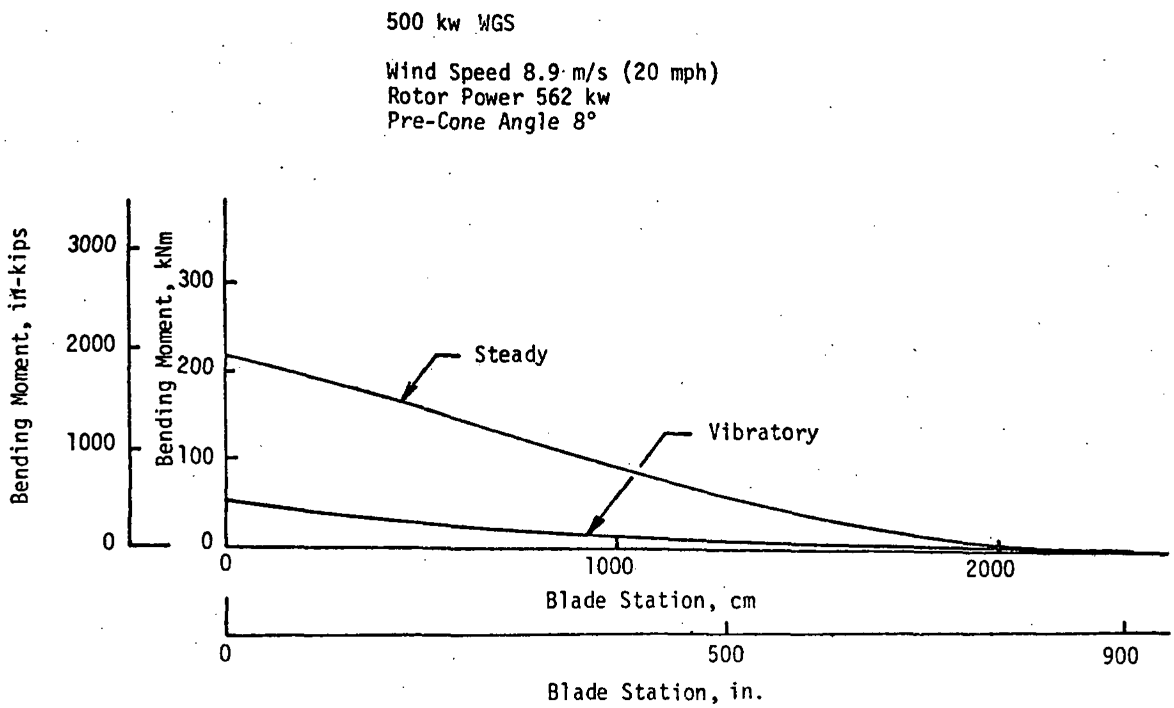


Figure 4-47. Out-of-Plane Blade Bending Moment Distribution

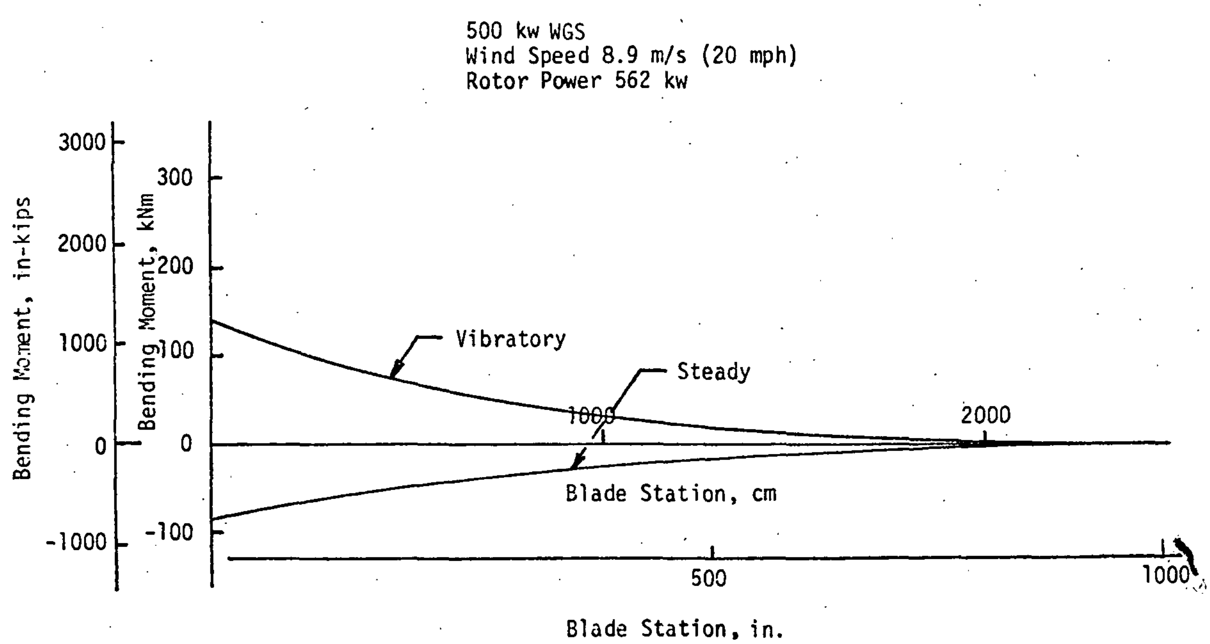


Figure 4-48. In-Plane Blade Bending Moment Distribution

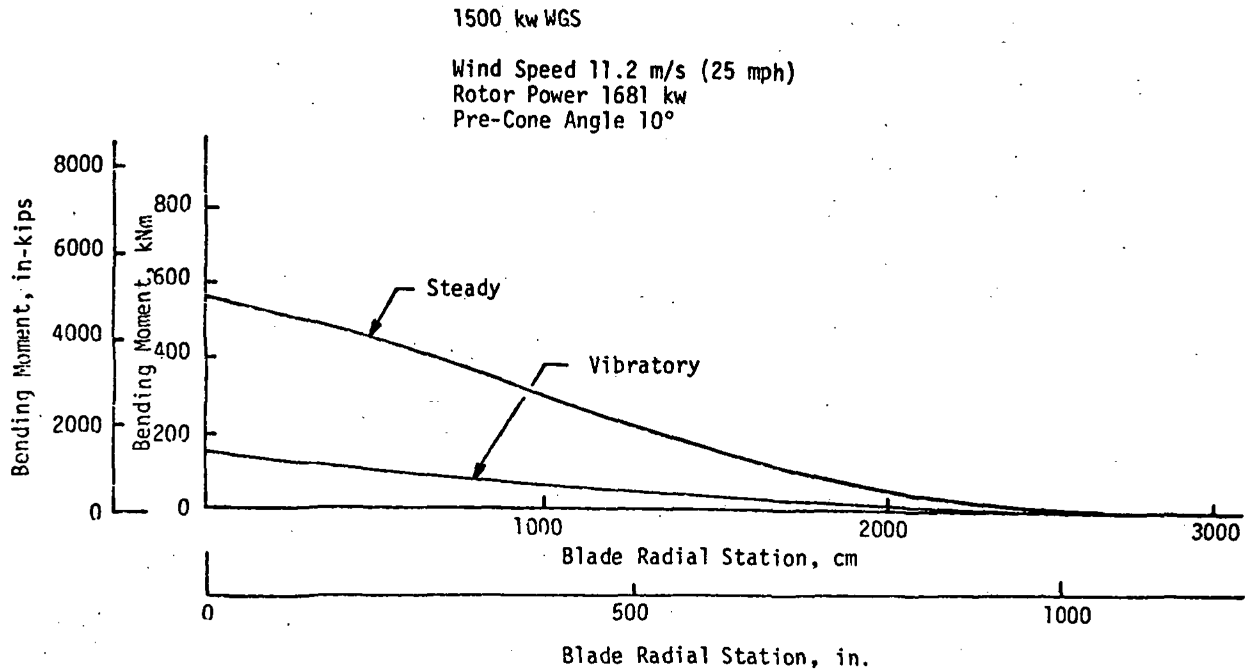


Figure 4-49. Out-of-Plane Blade Bending Moment Distribution

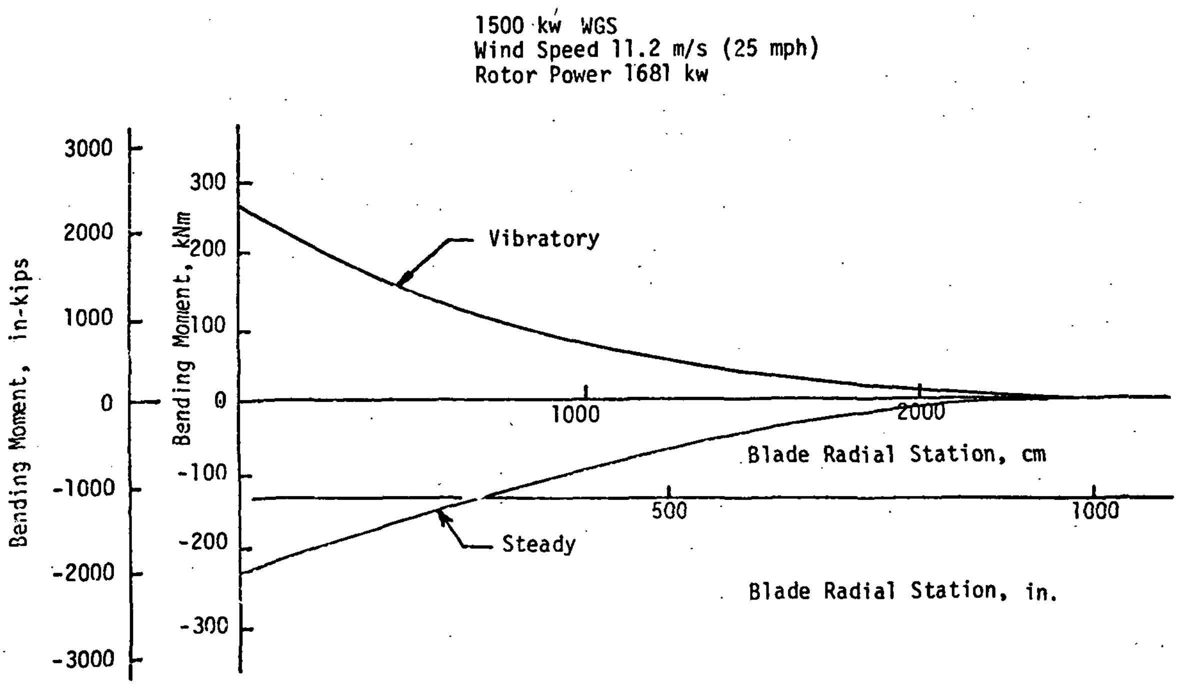


Figure 4-50. In-Plane Blade Bending Moment Distribution

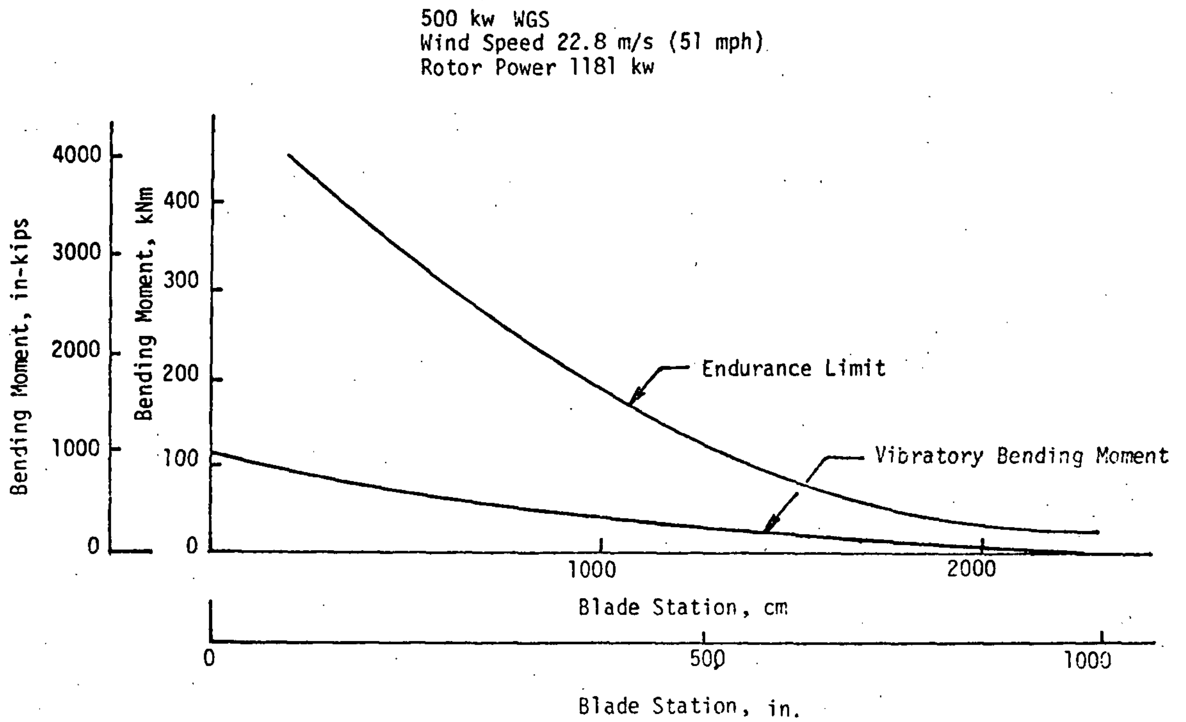


Figure 4-51. Out-of-Plane Bending Moment and Endurance Limit Distribution

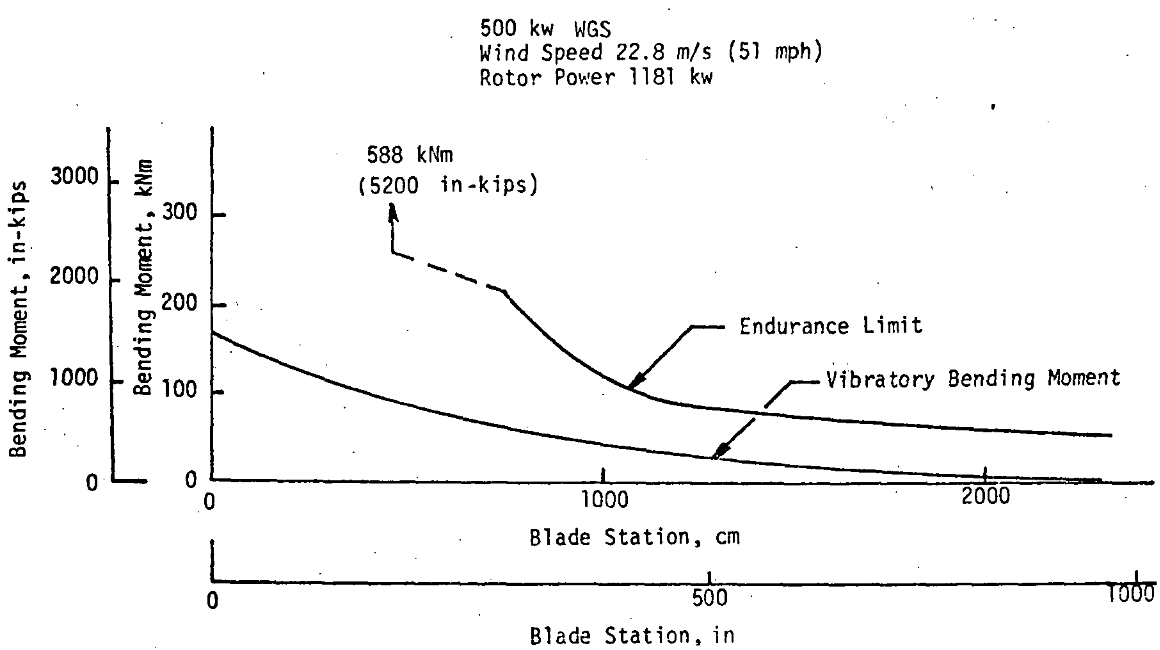


Figure 4-52. In-Plane Bending Moment and Endurance Limit Distribution

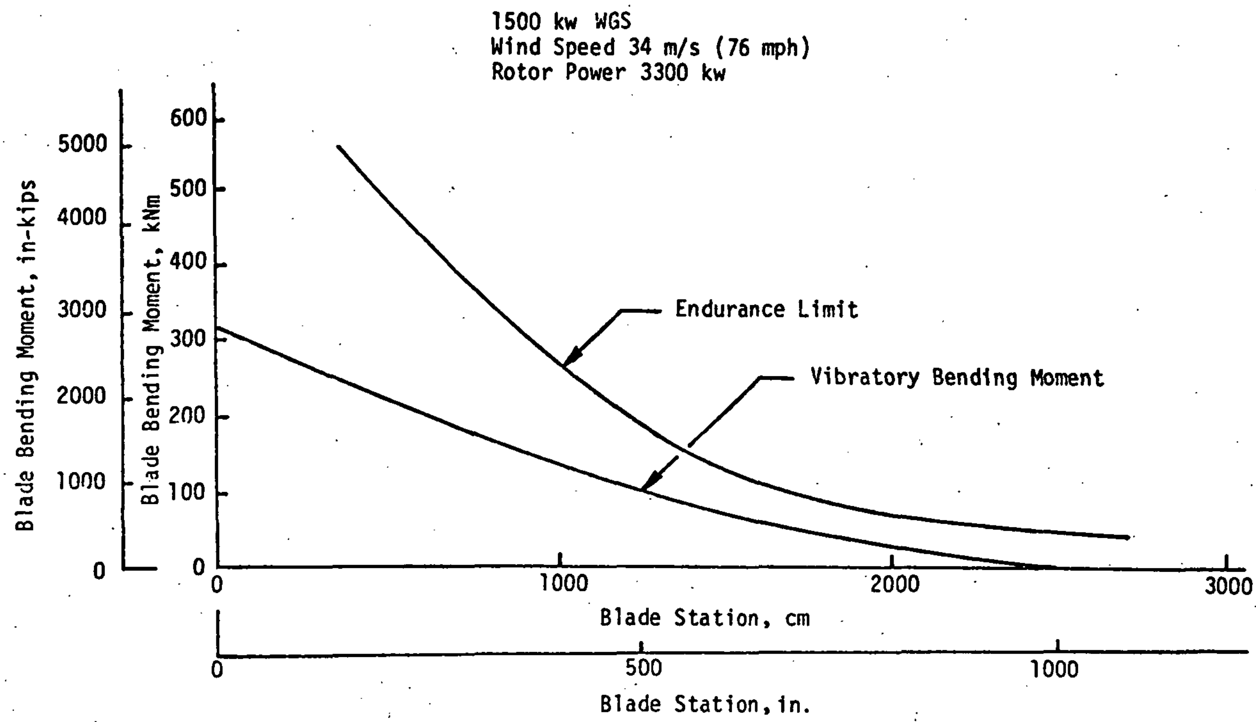


Figure 4-53. Out-of-Plane Bending Moment and Endurance Limit Distribution

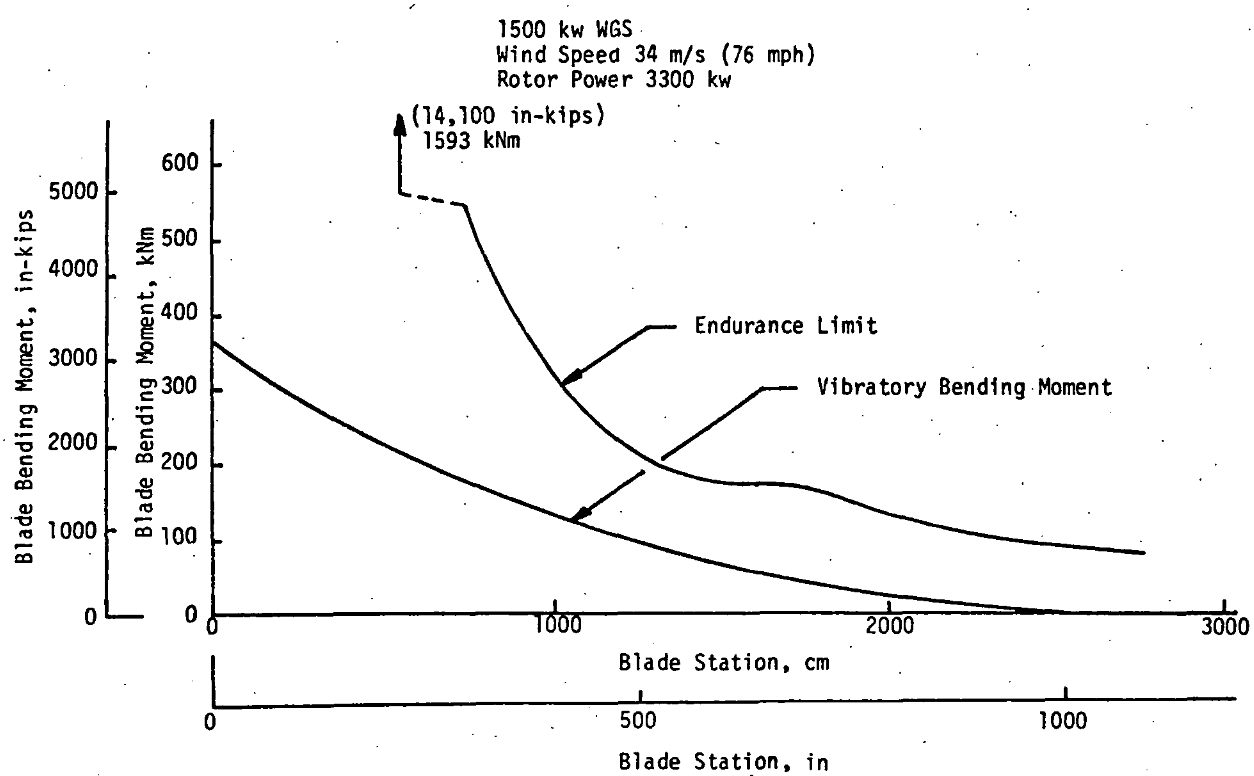


Figure 4-54. In-Plane Bending Moment and Endurance Limit Distribution

Steady and even order harmonics of shaft bending are reacted directly across the hub, which is a beam of large section compared with the blade sections. The use of heavy hub sections to supply large margins of strength in the hub is practical only because the diameter of the hub is very small (considering rotor diameter) and, therefore, the weight contribution of the hub is not excessive.

4.5.3 Weight and Cost Summary

4.5.3.1 Rotor Subsystem Weight

A weight summary of the 500 KW and 1500 KW rotor subsystems is shown in Table 4-2. Weight estimates were prepared for each fabricated element of the rotor blades and hub using standard helicopter weight categories. These estimates were calculated from the rotor subsystem and component layouts, supplemented by preliminary design weight estimating tools. The individual element weights were then combined into the major categories shown in Table 4-2.

4.5.3.2 Rotor Subsystem Cost

The weights in Table 4-2 were used for both the preliminary design rotor analyses previously described, and for calculating rotor subsystem production costs.

Blade Costs - A rotor blade cost analysis was conducted with the assistance of the Allegheny Ballistics Laboratory Division of Hercules, Inc. At the time these estimates were being prepared, Hercules was serving as a major subcontractor to Kaman in the design and development of an all-composite rotor blade for the Army's AH-1Q helicopter. The materials and fabrication technology for the rotor blade of the 500 KW and 1500 KW preliminary designs are similar to the AH-1Q blade, which employs filament winding techniques to fabricate the blade's primary structure. The experience gained from this program provided the basis for estimating blade costs.

The blade production cost estimates considered four categories of costs:

1. Amortized tooling
2. Tooling setup
3. Filament winding operations
4. Cost of recurring materials not included in the filament winding costs.

Estimates were made for production lots of 4, 200 and 2,000 blades. Development costs were not included.

Tooling costs were estimated on the basis of the number of tooling sets required for each of the three production quantities. Included were the costs of mandrels and assembly tools, and automation tooling for quantity production of 200 and 2,000 blades. Tooling costs were amortized on a straight line basis over the number of units in the production run.

TABLE 4-2. ROTOR SUBSYSTEM WEIGHT SUMMARY

	WEIGHT, kg (lb) 500 Kw	WEIGHT, kg (lb) 1500 Kw
BLADES		
SPAR	998 (2198)	1796 (3956)
ROOT FITTINGS AND PLATES	208 (458)	336 (739)
HONEYCOMB PANELS	109 (239)	143 (316)
TRAILING EDGE SPLINE	96 (211)	140 (308)
OTHER	134 (296)	171 (377)
TOTAL (EACH)	1545 (3402)	2586 (5696)
HUB		
HOUSING AND END PLATES	1047 (2307)	2551 (5620)
ROTOR SPINDLE BEARINGS, SUPPORT, SEALS	946 (2083)	1637 (3606)
PITCH BEARINGS, SUPPORT, SEALS	2441 (5377)	7274 (16021)
PITCH CONTROL MECHANISM	505 (1112)	814 (1793)
TOTAL	4939 (10879)	12276 (27040)
TOTAL (2 BLADES)	8029 (17682)	17448 (38432)

Tooling setup costs were estimated on a per blade basis and were adjusted to reflect learning curve improvements with increasing levels of production.

The cost of filament winding operations, the primary fabrication process, was estimated by Hercules on a cost per unit weight basis for each of the major components of the blade: spar cells, cheek plates, trailing edge spline and overwrap. Weights were obtained from the blade weight calculations. The unit costs for filament winding were adjusted for increasing production quantities to reflect learning curve benefits and the use of automated tooling.

The costs of recurring materials not included in the filament winding operations were estimated on the basis of weight or volume.

Table 4-3 gives the production cost estimates for the blades, including tooling, fabrication and materials. As shown, a considerable reduction in unit cost is forecast as production level is increased, due to both tooling amortization and labor and material learning curve effects. The rotor blades are one of the few system elements that significantly benefit from increasing production level due to these effects.

Hub Cost - Hub costs were also calculated on a cost per unit weight basis. Average costs for the hub bearings were obtained from typical catalog prices which were found to be relatively standard among several manufacturers for given types and sizes of bearings. Estimates for the structural elements of the hub, barrel and grips, were based on material and labor costs for the fabrication of similar components at Kaman. All costs were converted to a per unit weight basis for estimating purposes. Table 4-4 gives the cost breakdown for the first production article 500 KW and 1500 KW rotor hubs.

Total Subsystem - Table 4-5 summarizes the rotor subsystem costs for the 500 KW and 1500 KW units. Cost estimates for the pitch controls were obtained from an analysis of the costs of similar components manufactured at Kaman. Average cost per unit weight estimates were used. Subsystem integration costs are a fixed ratio of the major component costs and represent interfacing hardware items not included with the major components. Cost reductions for the two production levels represent learning curve improvements and the benefit of quantity purchase discounts. With the exception of the rotor blades, 10% and 5% average cost reductions are estimated for production lots of 100 and 1,000 units, respectively.

4.5.4 Operational Considerations

4.5.4.1 Starting and Stopping the WGS Rotor

Objectives of the Study - The objectives of the study were to examine and analyze the starting and stopping cycles of WGS operation in sufficient detail to fulfill the preliminary design requirements:

a. Assess the starting and stopping characteristics of rotors which are optimized for aerodynamic efficiency in the operating regime

TABLE 4-3 ROTOR BLADE UNIT COST ESTIMATES

	ESTIMATED COST					
	500 Kw WGS			1500 Kw WGS		
	4 UNITS	200 UNITS	2000 UNITS	4 UNITS	200 UNITS	2000 UNITS
TOOLING SETUP	$ 1,000	$ 700	$ 400	$ 1,000	$ 700	$ 400
BLADE FABRICATION (INCL. MAT'S)						
WIND SPAR CELLS	65,940	32,970	21,980	118,680	59,340	39,560
WIND CHEEK PLATES	11,370	5,685	3,790	17,640	8,820	5,880
ASSEMBLE CHEEK PLATES	850	680	510	1,000	800	600
WIND OVER TWO CELLS*						
LAY-IN HONEYCOMB	850	595	340	1,000	700	400
FAB. AND FIT SPLINE	8,480	4,240	3,180	12,320	6,160	4,620
WIND OVERWRAP*						
REMOVE MANDRELS	2,000	1,500	1,000	2,000	1,500	1,000
FINISH BLADE	1,700	1,275	850	2,000	1,500	1,000
SUBTOTAL	92,190	47,645	32,050	155,640	79,520	53,460
OTHER RECURRING MATERIALS	3,935	3,935	3,935	4,800	4,800	4,800
NON-RECURRING TOOLING	63,750	4,675	2,335	75,000	5,500	2,750
TOTAL UNIT COST	$159,875	$56,255	$38,320	$235,440	$89,820	$61,010

* Included in Spar Winding

TABLE 4-4. ROTOR HUB UNIT COST ESTIMATES

(FIRST PRODUCTION ARTICLE)

HUB COMPONENT	ESTIMATED COST 500 KW	ESTIMATED COST 1500 KW
Grips	$ 8240	28320
Barrel	8670	18880
Pitch Bearings	7295	14600
Forward Hub Bearing	1140	1845
Aft Hub Bearing	1285	1920
TOTAL	$26630	$65565

TABLE 4-5 UNIT PRODUCTION COST ESTIMATE - ROTOR SUBSYSTEM

	ESTIMATED COST (000 $)					
	500 Kw WGS			1500 Kw WGS		
	1 UNIT	100 UNITS	1000 UNITS	1 UNIT	100 UNITS	1000 UNITS
BLADES	$319.8	$112.5	$ 76.6	$470.8	$179.8	$122.0
HUB	26.6	24.0	22.8	65.6	59.0	55.8
PITCH CONTROLS	6.3	5.7	5.4	9.2	8.3	7.8
SUBSYSTEM INTEGRATION	17.6	7.1	5.2	34.8	12.3	9.3
	$370.3	$149.2	$110.0	$580.4	$259.4	$194.9

b. Identify the trade offs between start/stop optimized rotors and those optimized for the operating regime

c. Assess the significance of resonance crossings during start-up and shutdown

d. Establish criteria for the selection of V_{ci} (cut-in wind speed), consistent with wind energy capture and resonance dwell considerations

e. Investigate electrically augmented starting

f. Investigate the starting characteristics of servo flap controlled WGS rotors.

Summary of Results - The most significant results of the study are:

a. WGS rotors which are optimized for operating aerodynamic efficiency and for structural efficiency will also possess satisfactory starting and stopping characteristics

   1. Start-up cycle times will be acceptable.

   2. They will develop significant aerodynamic torque for acceleration at the critical rpm range just below N_{rated} when the most severe resonance crossings are encountered.

   3. Efficient structural design leads to minimum rotational inertia for a given blade weight, thus enhancing acceleration rates.

b. Rotor start-up should be initiated at wind speeds slightly higher than the wind speed for zero power output at rated rpm. Rotor accelerating torque reduces to zero and rotor speed approaches rated rpm asymptotically at the wind speed for zero power. Slightly higher wind speeds will avoid prolonged operation at rotor resonance frequencies encountered during runup.

c. Electrically augmented starting at torque levels as low as 2-1/2 to 5 percent of rated torque can significantly reduce rotor acceleration time and the fatigue cycle accumulation from resonance crossings at low wind speeds. The increase in cost and system complexity of electrically augmented starting does not trade off favorably against the additional energy captured. Pure aerodynamic starting associated with a V_{ci} set slightly higher than the wind speed for zero power appears optimum.

d. Servo flap control of blade pitch will require an auxiliary pitch positioning system for start-up and shutdown. Some degradation of total start cycle time is indicated, but acceleration rates at resonance crossings are predicted to be comparable to direct pitch control.

e. High deceleration rates for stopping the WGS are easily attainable. Resonance pass-through is not indicated to be a problem for this regime.

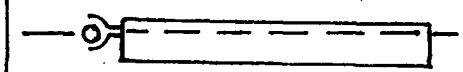
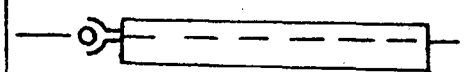
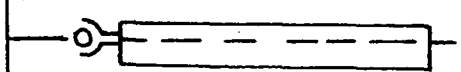
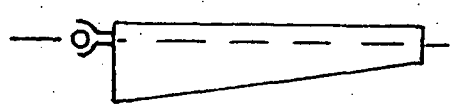
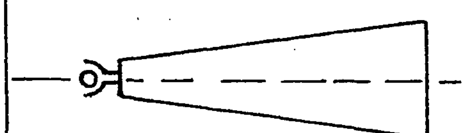
Discussion of Results

General Aspects of the Problem - For the purposes of this study, the WGS starting cycle is considered to encompass all events from the time of rotor parking brake release until the system is synchronized with the network and on line. A "normal" start is considered to be one in which the wind speed has increased to V_{ci} and has remained within the acceptable range for the minimum time, as monitored by the WGS control system. The preponderance of all system starts will be of this kind, and will involve start-up with minimum aerodynamic pressure available to overcome breakaway friction and rotational inertia. In this situation, the air velocity will typically be an order of magnitude slower than that at operating rpm (as experienced at the representative radial station at 3/4 radius). For example, the dynamic pressure at a velocity of 4.5 m/s (10 mph) is only 12 Nm^2 (1/4 psf), and at 8.9 m/s (20 mph), it is only 48 Nm^2 (1 psf). At an operating tip speed of 98.8 m/s (324 ft/sec), it is approximately 3350 Nm^2 (70 psf) at the blade reference station, .75R.

Upon initiation of the start cycle, assuming horizontal deployment of a two bladed WGS rotor, the available dynamic pressure will be essentially constant along the blade span. It can be readily seen that a rotor designed to maximize standstill torque and, hence, initial starting acceleration, would have aerodynamic design characteristics quite different from those resulting from optimization of operating efficiency. Some insight into the relative initial torque levels associated with these variables is shown in Table 4-6. The importance of chord distribution becomes immediately obvious as the torque generating capability of inverse taper planforms is shown to produce about 40% more standstill torque than conventional taper having the same numerical ratio and blade area. Similarly, twist distribution favoring the operating regime is shown to produce only one third of the standstill torque of a zero twist blade since, with the tip pitch angle set at the angle of attack for maximum lift, the root angles are negative, and produce drag rather than propulsive torque. It should also be noted, that while the chord distribution influence on aerodynamic torque is highlighted in the table, the rotational inertia also strongly influences acceleration rates. In practical blade design, significant differences are to be expected between constant chord and highly tapered blades, and dramatically higher inertia would be anticipated for the inverse taper case.

Figure 4-55 illustrates the variation of rotor shaft torque with wind speed for a typical 1500 KW WGS rotor, optimized for the operating condition. Standstill torque (N = 0 curve) can be seen to follow the expected square law variation with wind speed, while the rated rpm torque curve rises significantly faster. The branch of the N_R curve below the baseline represents the input torque

Table 4-6. Initial Torque at Standstill

Case	Planform	Twist	Taper Ratio	Relative Starting Torque
1		24° Linear	1:1	0.333
2		Zero	1:1	1.0
3		Zero	1:1	1.0
4		Zero	3:1	0.833
5		Zero	1:3	1.167

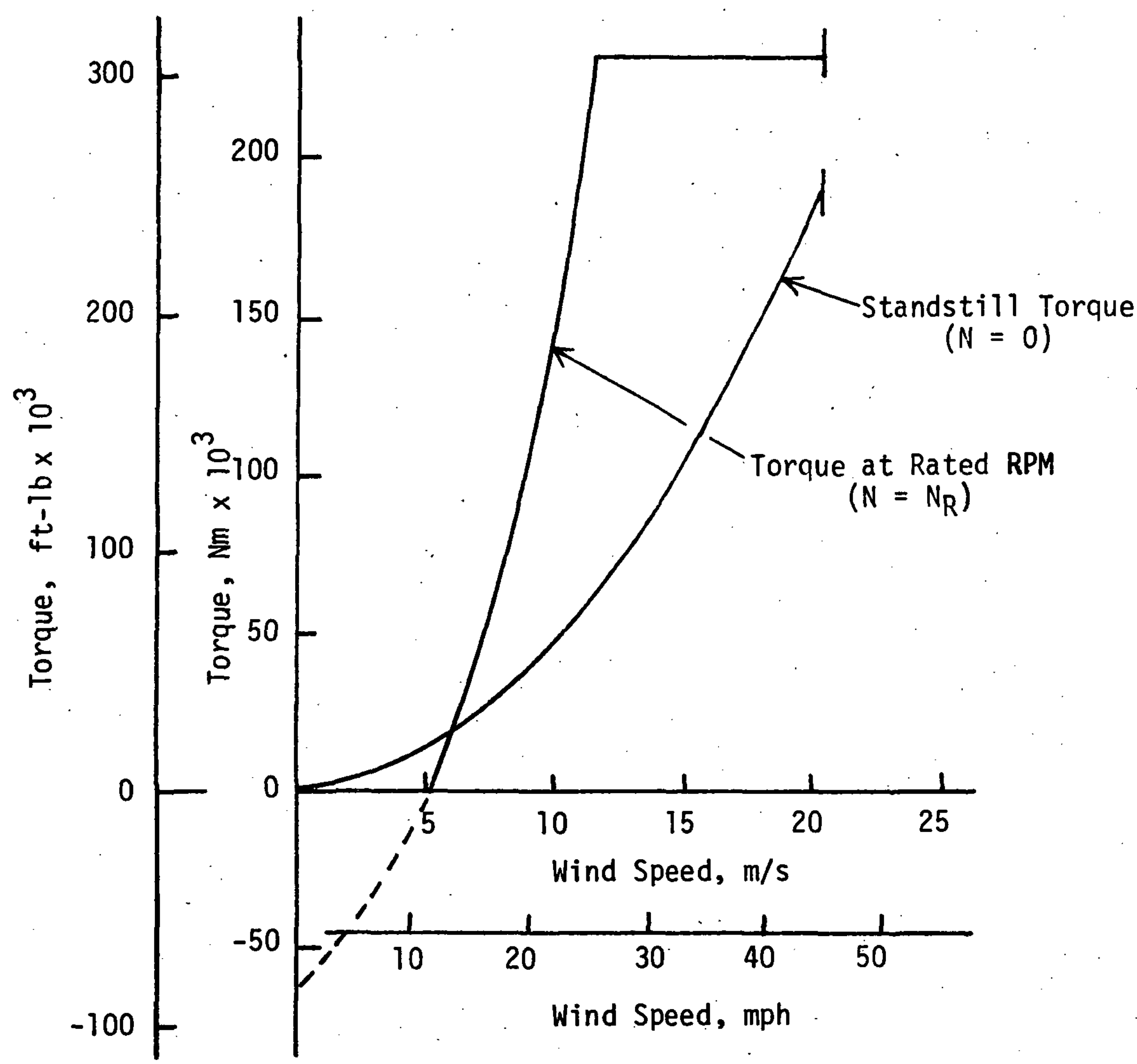


Figure 4-55. Rotor Shaft Torque Vs Wind Speed

theoretically required to sustain operation at rated rpm below $V_{minimum}$, and the intercept at V = 0 is seen to be the profile torque required by the rotor at zero wind.

Starting cycles can be considered to track vertical paths on the Figure 4-55 map at constant wind speed, starting at the N = 0 standstill torque level and terminating at the $N = N_{rated}$ curve by pitch regulation input. Typical cycles are qualitatively examined in Figure 4-56. Referring to this figure, it will be seen that three starting cases are graphically illustrated by vertical lines on the torque vs wind speed graph, and as time variations in the lower group of figures. For all cases, pitch control profiles which maximize propulsive torque across the rpm range are utilized.

Case A serves to illustrate certain problems associated with initiating a start (selecting V_{ci}) above, but very close to, the wind speed for zero power at rated rpm. Such an operating basis has been suggested in the literature for the purpose of maximizing energy capture by the WGS, and in recognition of the fact that with wind speed variations, even excursions which are symmetric with respect to the theoretical V_{min}, can produce time averaged positive power output. This is due to the fact that WGS output varies as the cube of the wind speed and, thence, the positive going excursions put more energy into the network than the (symmetric) negative going ones extract. The first time dependent curve for Case A shows the variation of torque and, hence, acceleration, with time; while the second graph illustrates the time build-up of rotor speed. The torque and, hence, the acceleration is seen to approach zero nearly asymptotically, with the result that total start cycle time is relatively long. Several other problems may be recognized from this case:

1. The time rate of rpm change, particularly during the upper 50% of the rpm range is relatively slow. There are clear implications that the pass-through of resonances may be critical

2. Effective performance of the synchronization cycle requires that rotor rpm regulation be tight, to hold the rate of $\dot{N}_R$ pass-through within acceptable limits for lock-in. It can be deduced that with maximum rotor torque barely adequate to insure reaching N_{rated}, small wind speed variations on the downside could abort control system efforts to establish the conditions for synchronization of the generator with the network.

3. The potential for an aborted start is suggested by the low torque margin at N_{rated}, which could decay to zero during the long run-up cycle.

Case B is chosen to illustrate a condition at which wind speed is adequate to provide a significant torque margin at N_{rated}. It was taken, for convenience, at a wind speed at which the standstill and operating torque levels are approximately the same. The convex character of the curve can be deduced from the

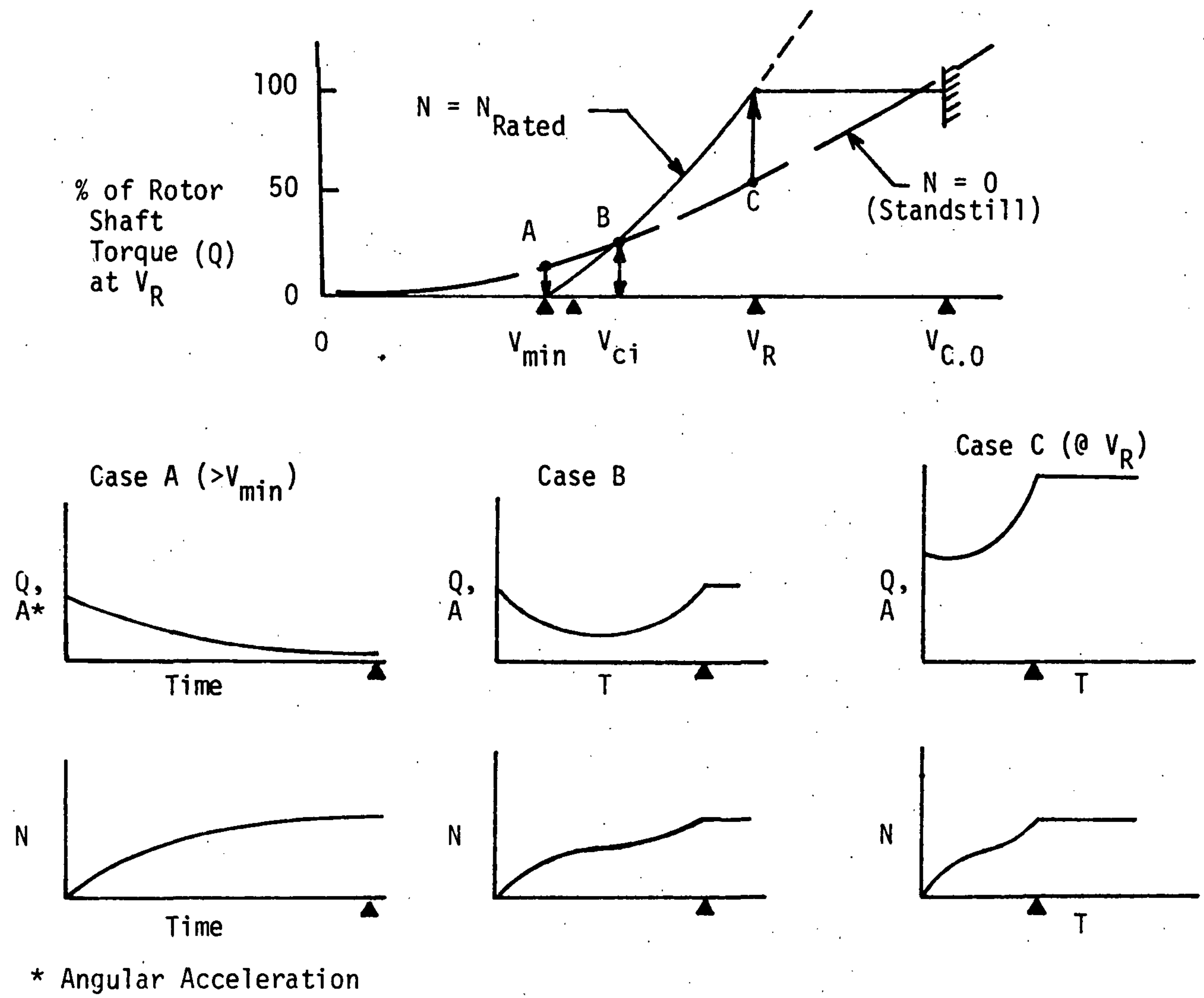


Figure 4-56 Typical Starting Cycles

changing flow states as the torque producing thrust distribution shifts spanwise along the blades and the rotor speed approaches the design optimized condition. It will be noted that, in addition to an appreciably quicker start, the rate of rpm build-up in the critical resonance pass-through region is much higher than in Case A, as is the torque margin for synchronization.

Case C is shown to illustrate the case of a start initiated at or near V_{rated}, with high standstill torque available at the start and rated torque levels approached in the critical resonances crossing region. Case C suggests that pitch control profiles for high wind starts might profitably employ a shift to off optimum pitch at rpms below N_{rated} to minimize overshoot.

Resonance Dwell - Reference has been made to the problem of resonance pass-through during the start-up cycles of the WGS. Figure 4-57 identifies, on the familiar Campbell diagram format, some of the more significant resonance crossings that a typical 1500 KW WGS rotor might encounter during start-up. The modes shown are rotor blade elastic bending modes. In a properly designed WGS rotor, they will be located between, and well removed from, the harmonic orders of excitation, at design rated rpm, where the constant rpm operating mode of the WGS will be a distinct advantage over comparable helicopter rotors and aircraft propellers. Theoretical pass-through of similar resonances occurs routinely in helicopter rotors during start-up, and fatigue damage accumulation from this source is not a problem. With the WGS rotor, however, some significant differences exist that demand consideration:

1. During run-up, the WGS rotor, unlike the helicopter rotor, is exposed to significant levels of excitation, since all its major sources of excitation are operative, and relatively independent of rotor speed:

   a. Wind Environment

      (1) Wind shear (vertical velocity gradient)

      (2) Tower shadow

      (3) Unsteady flow

   b. Gravitational Force Field

2. The WGS is dependent on wind energy for starting and, thence, has a very low ratio of torque to inertia. Starting time cycles are a large multiple of those associated with rotor craft, and much slower acceleration rates through the resonance regions are to be expected

3. The very long design life requirement for WGS rotors, together with anticipated frequency of starts (more than once per day) combines to produce a high exposure to potential fatigue damage, if resonance dwell permits large stress amplitudes to be attained.

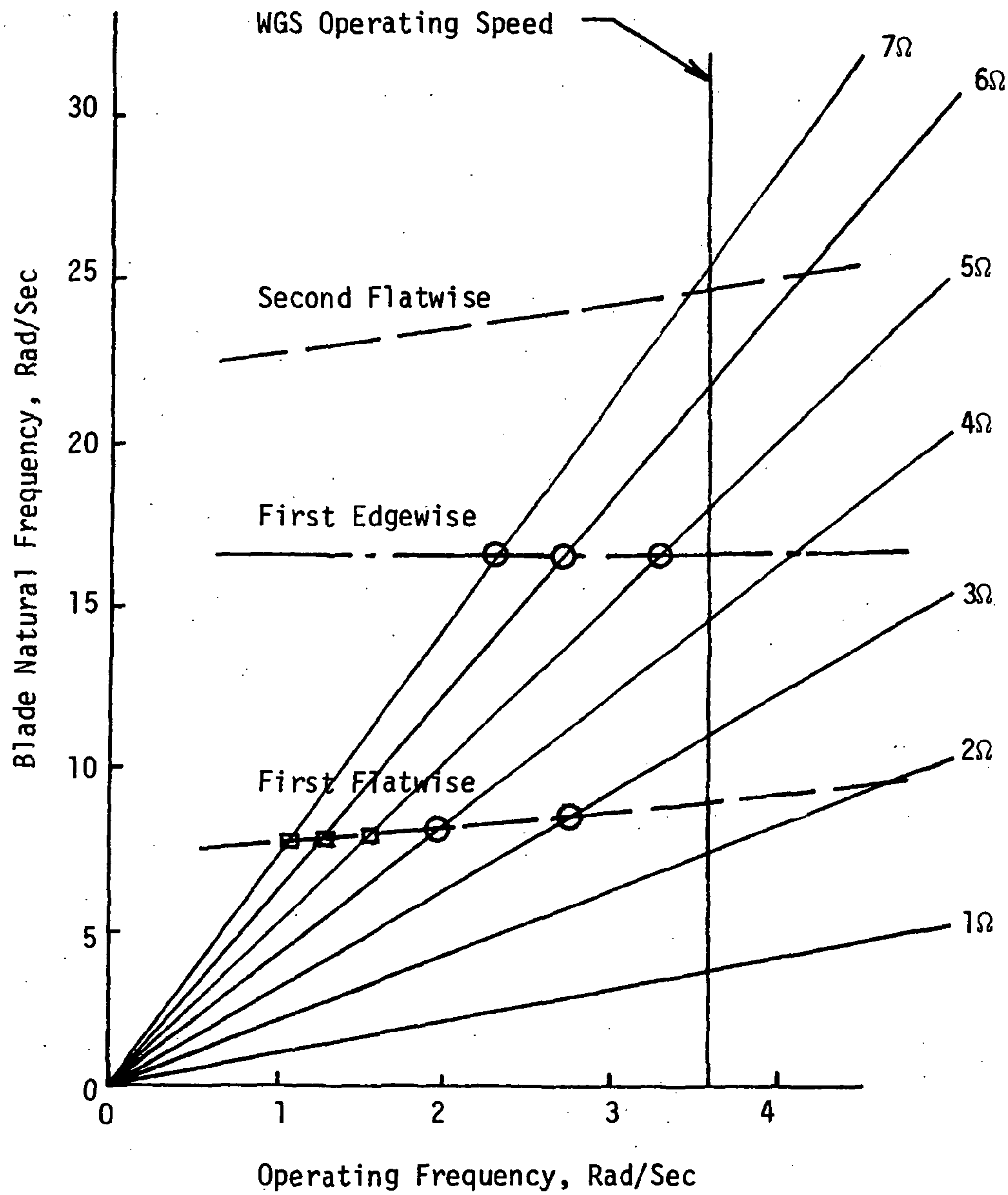


Figure 4-57. Blade Natural Bending Frequencies

Figure 4-58 and Table 4-7 illustrate the potential problem for a typical 1500 KW WGS rotor. Figure 4-58 shows graphically rpm time histories for a range of wind speeds from V_{min} to V_{rated}, and identifies an example critical speed crossing just below the operating range. The dramatic differences in rpm time rate at these crossings with increasing wind speed are evident from the graph. The curve for V_{min} was included to further emphasize the fallacy of setting V_{ci} at V_{min}. Table 4-7 presents the computed data illustrated in Figure 4-58. The number of potentially fatigue damaging load cycles for the example resonance pass-through is listed in Column 3 for an assumed high amplification bandwidth of $\pm$ 10 percent around the critical rpm. It will be noted that a small change in V_{ci} can result in significant reduction of fatigue damage exposure.

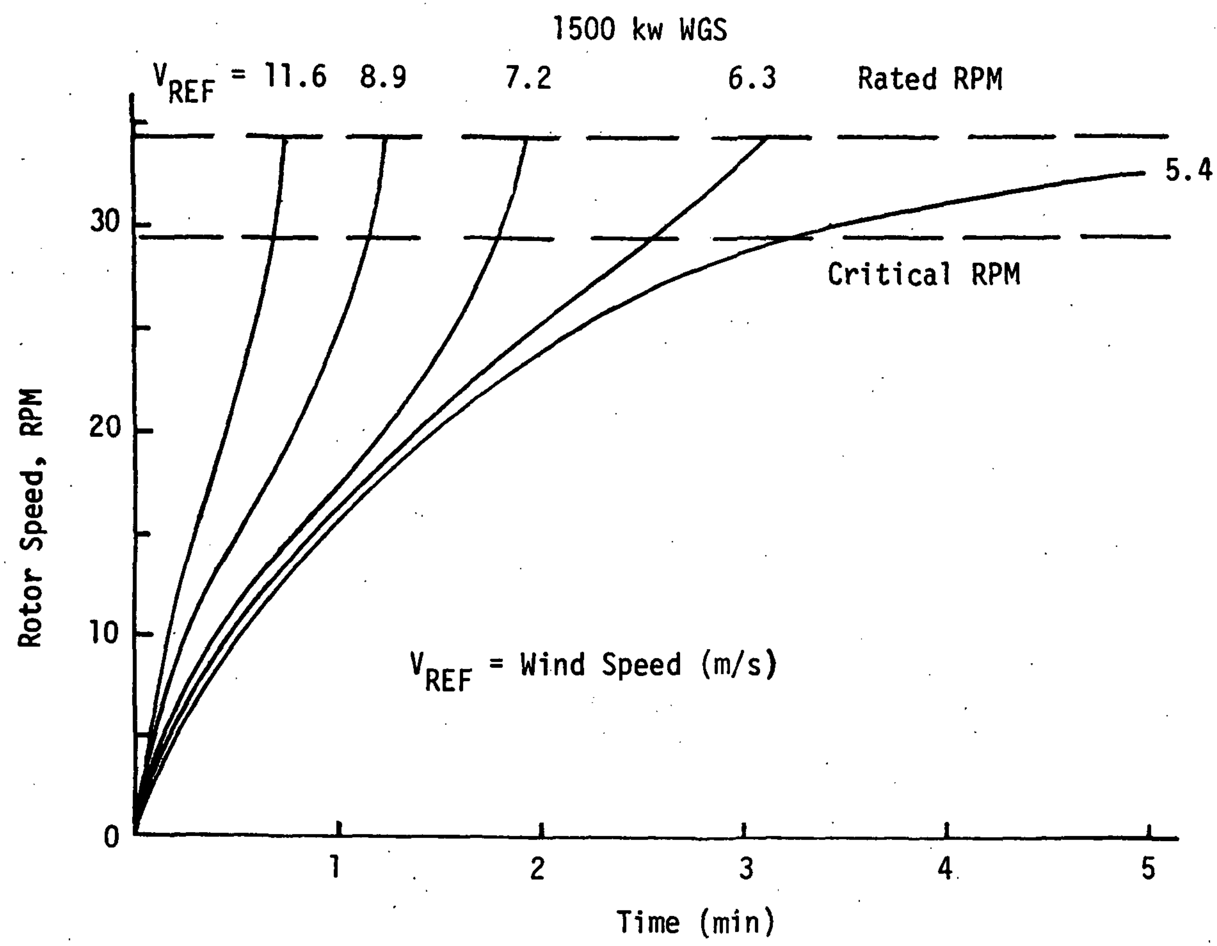
The augmented starting characteristics for the same rotor are presented in Table 4-8 to illustrate the capability of a relatively low boost torque to significantly reduce fatigue damage at low wind speeds. $V_{ci} = V_{min}$ is used in this example case, a situation in which rpm theoretically would approach, but never attain N_{rated} without boost. As noted in the results summary of this section, significant additional cost (of the order of \$20,000 - \$40,000) was estimated for implementation of electrically augmented starting, together with an increase in WGS system complexity and maintenance requirements. Pure aerodynamic starting, in conjunction with $V_{ci} > V_{min}$ was, therefore, recommended for the preliminary designs of this study.

The same analytical model used to perform the evaluation of starting cycles for the WGS, was used to examine the stopping cycle. It was immediately evident that high deceleration rates are easily attainable and that structural considerations provide the limitation on deceleration torques. Stopping times approximating one minute for a 1500 KW WGS appear achievable and safe.

4.5.4.2 Blade Tip Deflections and Tower Clearance

The basic tower/rotor geometry is shown in Figures 4-59 and 4-60 for the 500 KW and 1500 KW systems, respectively. The rotor was located downwind of the tower to provide adequate blade tip-to-tower clearance without excessive rotor head overhang. Rotor head overhang is excessive when the effective bending stiffness in the plane of the rotor is reduced to the extent that rotor dynamic responses described in paragraph 4.5.2.6 become critical. Since bending stiffness at the rotor head varies as EI/ℓ^3, it was judged that acceptable stiffness would be provided for the least structural weight and cost with the downwind location.

Rotor head offset, rotor shaft tilt, and blade preconing angles combine to provide clearance between the blade tips and the tower, as shown in Figure 4-61. Blade steady bending deflections also increase tower clearance when the rotor is downwind of the tower in normal operation. Rotor shaft tilt and rotor head offset were conservatively selected to allow for blade dynamic bending deflections during wind gusts, even wind direction reversals, which reduce blade tip-to-tower clearance. The degree of conservatism built into these selections can



Notes: 1. Pure Aerodynamic Start
2. Optimized Pitch Schedule
3. Rotor Blade Critical RPM Shown at 78% N_R Representative of First Bending Modes

Figure 4-58. Typical Starting Cycles

TABLE 4-7 COMPARATIVE FATIGUE CYCLE ACCUMULATION PER START, DUE TO RESONANCE CROSSINGS VS WIND SPEED

WIND SPEED V_{REF} m/s	(mph)	TIME TO N_{RATED} (min)	CYCLE FROM 78% ± 10% N_R PASS THRU	TOTAL REVS TO N_R
5.4	(12)	∞	32	∞
6.3	(14)	3.14	22.	45.6
7.2	(16)	1.95	6.6	33.6
8.9	(20)	1.25	4.3	21.5
11.6	(26)	0.74	2.5	12.8

TABLE 4-8 AUGMENTED STARTING CHARACTERISTICS FOR 1500 KW WGS

V_{REF} = (12 MPH) 8.18 (m/s)

BOOST TORQUE* %RATED	TIME TO N_{RATED} MIN	CYCLES FROM 78% $\pm$ 10% N_R PASS THRU	TOTAL REVS TO N_R
ZERO	∞	32	∞
2 1/2	2.0	8.9	41
5	1.29	6.2	25
10	0.76	3.4	14
20	0.43	1.8	7.7

* FULL TIME DUTY CYCLE DURING START-UP

RATED TORQUE = 228600 N-M
= (310000 FT-LBS)

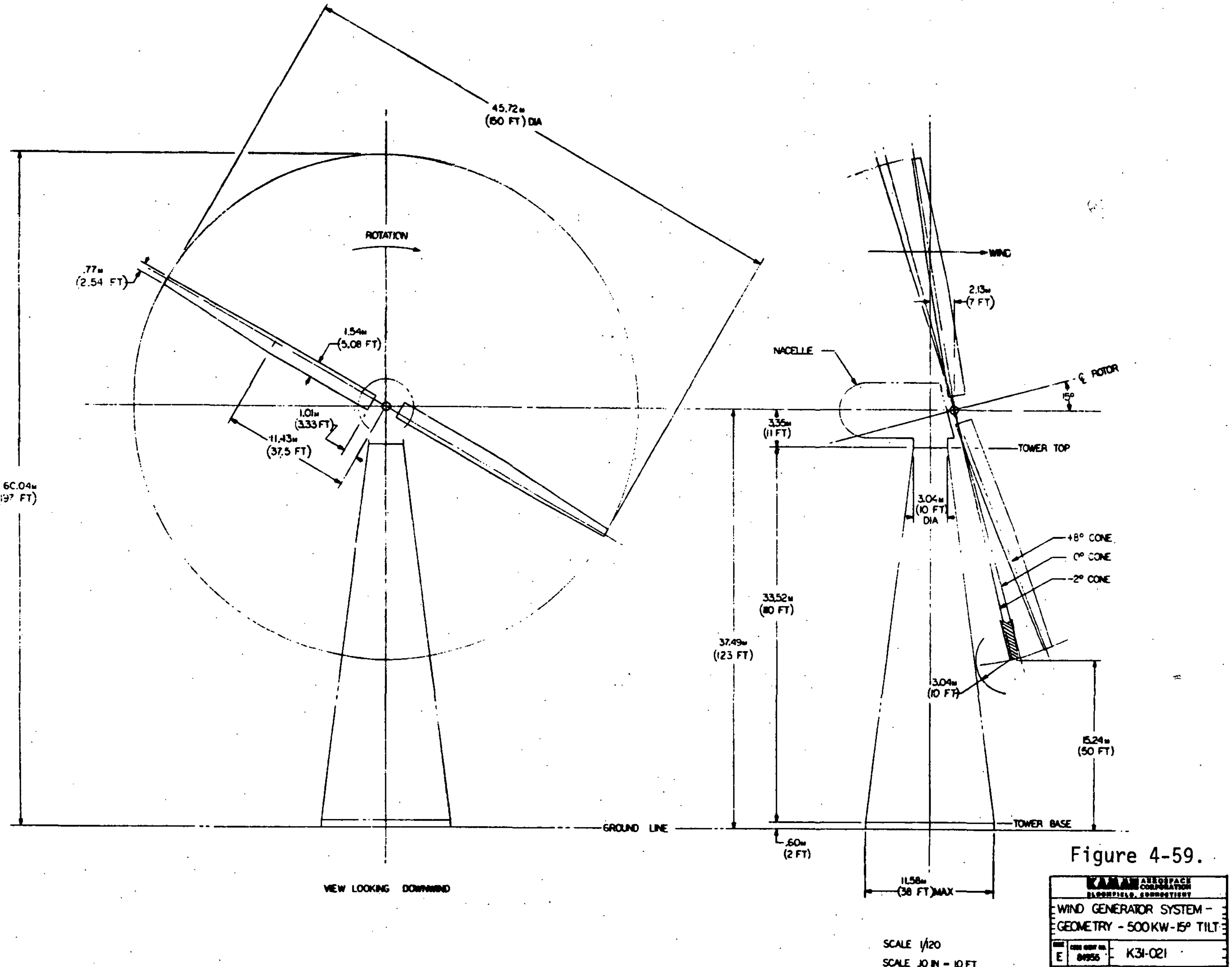

45.72m
(150 FT) DIA
ROTATION
.77m
(2.54 FT)
1.54m
(5.08 FT)
1.01m
(3.33 FT)
11.43m
(37.5 FT)
60.04m
(197 FT)
GROUND LINE
VIEW LOOKING DOWNWIND
WIND
2.13m
(7 FT)
NACELLE
℄ ROTOR
15°
3.35m
(11 FT)
TOWER TOP
3.04m
(10 FT)
DIA
+8° CONE
0° CONE
-2° CONE
33.52m
(110 FT)
37.49m
(123 FT)
3.04m
(10 FT)
15.24m
(50 FT)
TOWER BASE
.60m
(2 FT)
11.58m
(38 FT) MAX
SCALE 1/120
SCALE 1.0 IN = 10 FT
KAMAN AEROSPACE CORPORATION
BLOOMFIELD, CONNECTICUT
WIND GENERATOR SYSTEM -
GEOMETRY - 500KW-15° TILT
E
K31-021

Figure 4-59.

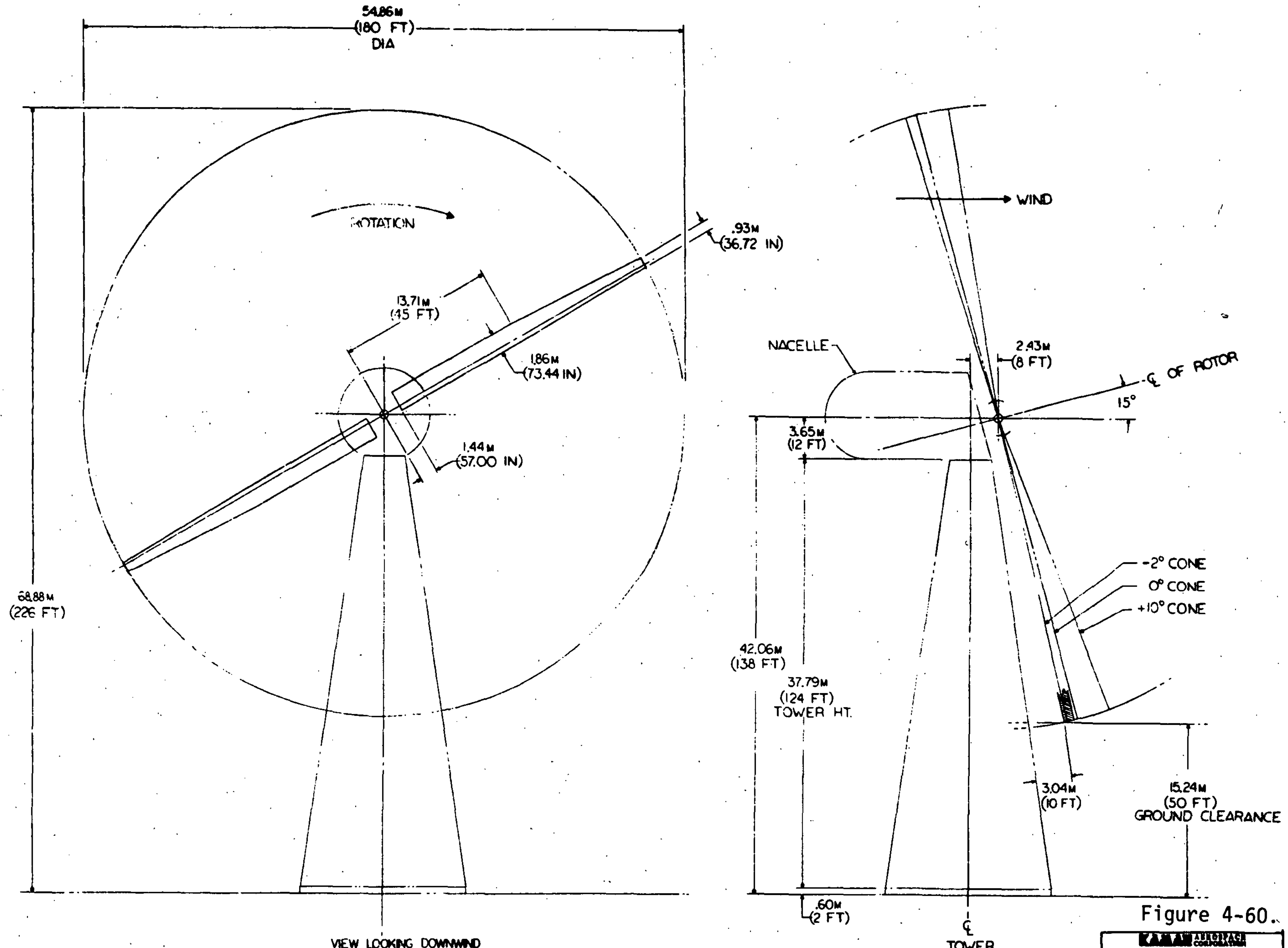


Figure 4-60.

KAMAN AEROSPACE CORPORATION
BLOOMFIELD, CONNECTICUT
WIND GENERATOR SYS.
GEOMETRY – 1500 KW
E 94855 K31-015

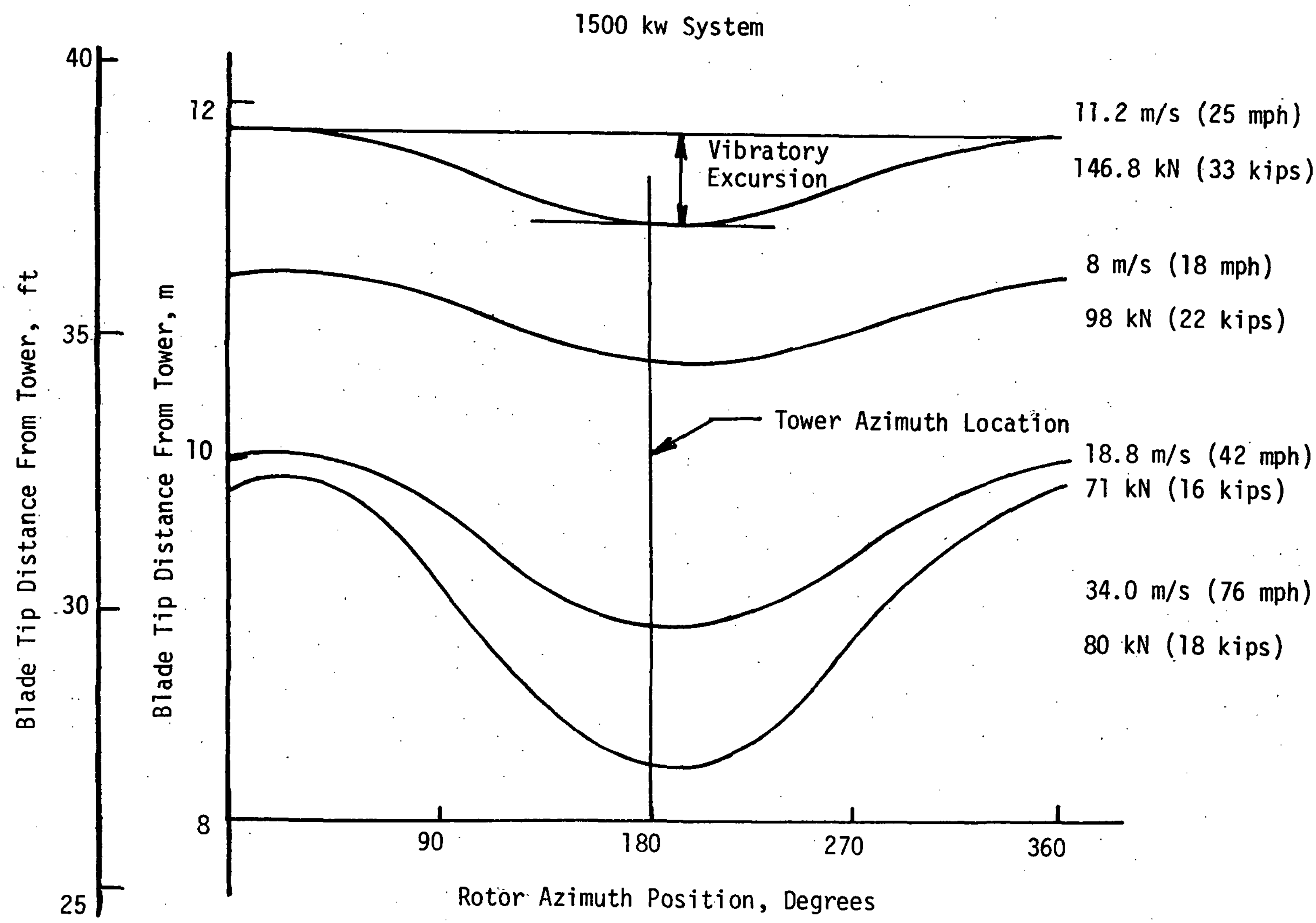


Figure 4-61. Blade Out-of-Plane Bending Deflection

be evaluated by a multiple degree-of-freedom dynamic analysis which includes deflections of the blade tips. Such an analysis is expected to show that tip deflections are not as large as has been allowed for in the present design, thereby allowing a reduction of rotor shaft tilt from 15° to 10°, or lower.

Figure 4-61 also shows the effect of increasing wind speed on vibratory tip deflections for the wind shear effect only; greater deflections occur at higher wind speeds, as would be expected. Note that steady deflections decrease at higher wind speeds because the blade pitch angle setting for rated power produces less thrust.

4.5.4.3 Rotor Gyroscopic Forces

Substantial gyroscopic forces can be developed during WGS operations while reorienting the rotor into the wind in response to wind direction changes. The magnitude and occurrence of such forces are discussed in Section 7.

The rotor will also have a normal aerodynamic tendency to weathercock into the wind, as a result of its location downwind of the tower pivot axis. In addition to steady yawing moments, this will produce vibratory gyroscopic moments which will probably occur more frequently than the moments initiated by the WGS yaw mechanism. The latter has a built-in time delay to minimize "hunting" response to short term wind direction changes and gusts.

Additional aerodynamic analysis should be conducted during the detail design phase to evaluate the magnitude and frequency of occurrence of rotor response to transient wind direction changes. It may become necessary to alter the yaw time delay to minimize the aerodynamically induced moments.

In addition to rotor and tower structural loadings imposed by the gyroscopic forces, yaw motions will be somewhat erratic as a result of two-per-rev gyroscopic yaw reactions. A slow yawing rate is required to minimize such erratic motions in the commanded mode; however, uncommanded aerodynamically induced gyroscopic forces will still be present. The magnitude of such loadings should be investigated during the detail design phase.

4.5.4.4 Foreign Object Damage

The WGS rotor blades will be exposed to foreign object damage from wind-driven debris, objects thrown by hand, small arms fire and, possibly, bird strikes. It is likely that small arms fire will be a significant source of damage, because the large rotating blades will present an attractive target.

The WGS blades described in this report are particularly tolerant of foreign object damage, especially small arms fire, by virtue of their composite construction, which provides many redundant load paths around ballistic damage areas. This feature is a major requirement of the design of a composite blade for the Army's AH-1Q helicopter, currently under contract for design and development at Kaman. The same basic construction is proposed for the WGS rotor blade.

With respect to bird strikes, the rotational speed of WGS rotors (32 - 34 rpm) is sufficiently slow that most birds will be able to avoid the blades without being hit by them. The blade leading edge has sufficient wall thickness and curvature, to resist damage, even if struck by large birds.

4.5.4.5 Rotor Blade Repair

The capability of performing extensive field repairs of rotor blades constructed of composite materials was demonstrated under a program conducted by Kaman for the Army's Air Mobility R & D Labs at Fort Eustis, Virginia. In this program, a new rotor blade was designed for an existing Army helicopter with the objective of reducing life-cycle costs through low acquisition cost, high reliability and the ability to perform major repairs of the installed blade in the field. The resulting design was comprised of an extruded aluminum spar and a blade afterbody constructed of polyamide paper (Nomex) core and glass skins.

The most extensive repairs were developed for the afterbody section, the area of the blade most susceptible to foreign object damage, ballistic strikes, etc. Two basic types of repairs were developed: (1) a skin patch for superficial damage not affecting the core material and (2) a plug patch for damage penetrating into or through the core. The skin patch, a buildup of impregnated glass fabric and scotchply material, is applied by cleaning and abrading the blade surface and applying the patch under heat and pressure using an epoxy adhesive. The plug patch, a skin patch to which a cylindrical section of core material is bonded, is installed by removing the damaged core material and bonding the plug in place under heat and pressure. Through-damage is repaired with two plugs installed from opposite sides of the blade.

All of the consumable materials required to make a repair, except hazardous substances and materials with limited shelf life, are packaged in kits. The epoxy adhesive is packaged in a separate kit in two-part, premeasured packages. A portable bonding fixture weighing less than thirty pounds is secured to the blade to supply local heat (160° F) and pressure (4 to 5 psi) for curing the repair. Army mechanics, after minimal training, successfully made numerous repairs of the installed rotor blades without difficulty. The repaired blades were subjected to extensive fatigue and whirl testing without failure.

Kaman believes that these same techniques can be adapted to field repair of the WGS rotor blades. Some modifications would be needed to accommodate differences in materials and methods of construction and also to adapt the techniques to the more difficult working environment presented by the WGS blade installation. It is expected also that techniques for repairing some types of spar damage in the field, a procedure not generally permitted with the highly loaded rotor blades on helicopters, can be developed. Consideration may have to be given to a method of adjusting balance after extensive repairs of the blade have been made. It is believed that techniques can be developed which will allow most of the normal types of damage to be repaired with the blades installed.

4.6 Adaptability to Other Systems

The WGS rotors are adaptable to a variety of worldwide site locations and environments, including extremes of temperature, precipitation and wind conditions. The two rotors investigated provide near optimum operating efficiency for a range of wind speeds from 4.5 m/s (10 mph) to 8.9 m/s (20 mph). A small reduction in efficiency will occur beyond this range.

Both rotors have adequate structural and performance capability for use in WGSs up to 3000 KW or higher, although some loss of aerodynamic efficiency will result from off-design operation. The two WGS designs are sufficiently similar that either rotor could be adapted to the other generator drive system with relatively minor modifications.

4.7 Conclusions and Recommendations

4.7.1 Conclusions

1. The WGS optimization study conducted under this program resulted in selection of large diameter, low solidity, two-bladed, hingeless rotors having variable pitch control and constant rotational speed. This configuration yields the lowest cost and least development risk.

2. The large diameters selected for the WGS require blades of composite glass fiber construction, which can be made 30 m (100 ft) long, or longer, while maintaining efficient structure and mass distribution. The automated filament winding technique offers a practical and economic method of fabricating long WGS blades in one piece.

3. Although two-bladed rotors have more dynamic response modes than rotors with three or more blades, troublesome resonances can be avoided by proper blade tuning. It is considered essential that the rotating natural frequency of the first out-of-plane bending mode be above 2/rev. Similarly, the first in-plane mode frequency should be above 4/rev to avoid coupling with out-of-plane modes. These requirements dictate a rigid hub and very stiff blades.

4. Composite construction lends itself to achieving the required structural stiffness and mass distributions for proper blade tuning. It also permits optimization of blade taper, thickness and built-in twist distributions.

5. Starting characteristics of the WGS dictated a direct blade pitch control system to achieve maximum torque at the lowest wind. The cut-in wind speed should be set slightly higher than the speed to achieve rated rpm at zero torque. The direct pitch control is also required for blade stability during rotor shutdown in high winds.

6. Rotor overspeed can occur during high power operation if the generator goes off the line. The pitch control rate proposed for the WGS is sufficiently responsive to prevent rotor overspeeds in excess of 150 percent of normal rpm, well within the structural capability of the rotor. Additional overspeed control can be provided by auxiliary drag devices, if required.

7. Preliminary flutter analysis indicates that the WGS blades are free from flutter and divergence for the cases examined. The preliminary study reported herein revealed the importance of quarter chord mass balance and feathering axis location. The 500 KW rotor, as defined in this preliminary design, is significantly less stable than the 1500 KW rotor as a result of the former's more rearward center of gravity location.

8. No critical whirl resonances were revealed by the preliminary analysis of the WGS rotor/tower system.

9. The structural design imposed by blade tuning requirements results in blade section properties which accumulate no fatigue damage in all but extreme wind conditions, which occur only rarely. Similarly, the rotor hub is a massive structure to provide the stiffness and low stress levels imposed by the design. Consequently, the 30 year life goal can be met for the entire rotor.

4.7.2 Recommendations

1. It is recommended that an investigation of the gust alleviation benefits of a servo flap control system be conducted for enhanced rotor stability of future wind generator systems.

2. WGS rotors should be operated in the negative pitch mode to obtain the benefits of lower thrust than in the flat pitch mode at the same power. The vortex ring state of thrust instability will be avoided in the negative pitch mode.

3. It is recommended that a more detailed flutter analysis, which includes the lag degree of freedom, be conducted. Effects of intermodal elastic coupling, built-in twist distribution, wind shear and other factors unique to WGS rotors should also be included in the analysis.

4. It will be necessary to review the analysis of the WGS rotor/tower system during the detail design phase to evaluate the validity of assumptions made during the preliminary design phase. In particular, rotor response to the half-per-rev gravity excitation must be evaluated in more depth.

NOTE:

Tower Shadow (see page 4-25, paragraph 4.4.3.4).

Subsequent information, obtained from NASA testing after completion of this study, quantified wind velocity reductions behind towers having high solidity and angular structural elements. The wind shadow behind such towers can produce substantially higher blade bending moments, with associated reductions in blade fatigue lives. Therefore, it is considered essential that tower design criteria include the requirement for minimum shadow effect by minimizing tower solidity and selecting structural elements having low drag coefficients.

5.0 CONTROLS SUBSYSTEM

This section describes the processes which were used to develop the Controls Subsystem concept and preliminary design for the WGS. A primary tool for this process was the Failure Modes and Effects Analysis (FMEA) presented in Appendix C of this report. The FMEA was used to guide the concept selection and design of the control system, based on the types of failures which might occur, the severity of their effect on the WGS and the ability to automatically detect and compensate for such failures. The control system which evolved from this process utilizes a microprocessor for data telemetry and for startup and shutdown sequencing of the WGS with hydraulic servos and analog equipment for the primary rotor controls. A pure mechanical control backup is provided for emergency shutdown of the rotor.

Descriptions of the Controls Subsystem hardware in this section are limited to the electrical and electronic portions of the Controls Subsystem. Mechanical and hydraulic portions of the Controls Subsystem hardware are described in Sections 4 and 7 (Rotor and Drive Subsystems).

5.1 Requirements

The WGS Controls Subsystem must be designed for operation at a remote, unattended site. The system must be fail-safe and self-monitoring, that is, it must be capable of detecting any failure within the WGS which is capable of causing secondary damage to WGS equipments and taking the appropriate protective action. The control equipments must be capable of maintaining proper operation of the system under extreme environmental conditions, such as wind gusting, temperature, etc. Safety and protective functions must be capable of being executed independently of the availability of external power. The Controls Subsystem must be conservatively designed to maintain high reliability and be properly protected against induced transients from the power line or from lightning strikes.

The functional requirements of the control system are:

1. Startup of the WGS from rest to design rotor speed

2. Shutdown and parking or securing of the WGS

3. Control of blade pitch to regulate the rpm of the WGS rotor when disconnected from the utility network

4. Control of blade pitch to regulate power output when connected to the utility network

5. Control of the yaw orientation of the WGS tower head

6. Telemetering and supervisory functions

7. Fault monitoring and protection of WGS functions (except generator protection)

8. Recording of significant data at the WGS site.

Both these functional requirements and their corresponding quantitative values were derived from the initial WGS concept control requirements as the system design was expanded, modified and refined. This process is described in the following paragraphs.

5.2 Control System Approach

The basic approach followed in the initial concept selection and design process for the control system was to make maximum use of available previous experience from earlier systems. This included a review and evaluation of previous development and research efforts on wind generators. It also included maximum use of Kaman's previous experience in rotor and control system design applicable to wind generators.

Conferences were held with Northeast Utilities to determine any special control requirements or limitations imposed by utility industry practices. Possible failure modes of the control functions were evaluated for their effects on both the WGS and the utility interface, from the standpoint of safety and possible methods of failure detection.

Selection criteria were established, as outlined in 5.4.1, to guide in the concept selection process. The preliminary selections and findings were reviewed and the process was repeated in greater detail to come up with the final control system design. The primary tool in this effort was a detailed failure modes and effects analysis, which is presented as Appendix C to this report. This analysis was actually used as a design tool and, in many cases, dictated features of the design as it finally evolved.

5.3 Component Availability and Technology Utilized

Based on the selection criteria of 5.4.1, primary emphasis was given to readily available, proven, off-the-shelf components for the Controls Subsystem. This includes standard analog building block devices, such as operational amplifiers and function modules, as well as standard digital logic building blocks. Digital microprocessors were initially excluded from consideration by Kaman because they represented a relatively new technology with a limited experience base in utility applications. Reservations were expressed by Northeast Utilities concerning the use of such devices for remote site application and possible reliability problems under severe environmental conditions. This concern was further reinforced by Mr. John Robb of the Lightning and Transients Research Institute (LTRI) regarding the lightning and transient susceptibility of such devices, as opposed to analog circuits.

Further investigation during the course of the study modified this position. A product search led to Data Signal Corporation which is presently manufacturing and supplying microprocessor-based equipment to utility companies for remote substation telemetering and supervisory functions. The equipment includes special provisions to adapt it to the severe environment at the remote site, such as optical isolators and filters for input isolation, conventional electro-mechanical relays for output isolation, and special coding provisions to provide high noise immunity for the telemetry link. After further consultation with Mr. Robb at LTRI, Data Signal Corporation and the using utilities, Kaman decided to utilize microprocessor-based equipment where applicable. LTRI indicated that this equipment could be satisfactory if designed specifically for this type of application, and provided with proper transient isolation devices.

Field experience on the Data Signal equipment to date does indicate satisfactory performance and reliability in remote site utility applications. Although the experience base is limited, it was decided that microprocessor equipment would be considered for possible use in the WGS where it offered special advantages. It was excluded from use on the generator protective relaying function and from the primary rotor pitch and yaw controls because of the limited utility experience base with microprocessor equipment and the critical nature of these particular functions.

5.4 Control Concepts

5.4.1 Selection Criteria

The following selection criteria were established for evaluating various control system concepts:

1. Concepts must be acceptable to the potential user (the utility company)

2. Preference is given to equipments with which utilities have previous experience in remote site applications

3. Concepts must be compatible with NASA specified environment, e.g., temperature, vibration, wind gusts, lightning and power line transients, etc.

4. The selected concepts must be the simplest practical to do the required task to assure low cost and high reliability

5. Control functions performed under emergency conditions must be able to be executed with minimum power demands on the emergency battery supply

6. Concepts used for critical functions must have readily predictable failure modes and must be easily amenable to automatic failure detection

7. Preference was given to concepts which have inherently long life with minimum maintenance.

5.4.2 Alternative Functions and Equipments

In considering alternative functions, it is helpful to categorize the control functions which must be performed into groups, since different classes of equipment may be better suited to satisfy each group of functions. These functional groupings are:

1. Primary control functions required to control rotor blade pitch and yaw orientation of the nacelle. These generally require continuous real time computation and analog type outputs. Inputs for these functions are also primarily analog signals.

2. Data recording functions at the WGS site.

3. Telemetering and supervisory functions between the WGS and the using utility.

4. Those functions required to control the timing and sequencing of the WGS during startup of the WGS, synchronization with the utility, and shutdown of the WGS. All outputs relating to these functions can be discrete (binary) and inputs are both discrete and analog.

Analog equipments are well suited for real time continuous control. They are somewhat cumbersome for timing and sequencing functions because extra equipment must be added to perform each function. The microprocessor, on the other hand, can add another function simply by adding steps in the software sequence. Analog devices do have the advantage of easily predictable and simple failure modes which make these devices easier to monitor for failures and provide automatic corrective action. Analog devices have the further advantage of a large experience base in utility applications.

Microprocessors are best suited for sequencing and timing functions which do not require continuous computation and which are amenable to time sharing. A limitation of the microprocessor device is the complexity of the various possible failure modes. This is a significant disadvantage if the microprocessor is used in the primary control loop. Failure modes and effects analyses of the primary control loop are more difficult and increase the probability of failure modes going undetected and causing possible damage to the system.

The initial cost associated with the development of the software for the microprocessor is a disadvantage. However, if several systems are built, the cost of the software can be spread over the several systems. The complexity of the software can also be considerably reduced if the microprocessor is not used as a part of the primary control loop, and is limited to sequencing and supervisory type functions. Typical microprocessor equipment for the wind generator (Data Signal Corporation) already has software routines written for the supervisory functions. Preliminary flow diagrams were developed by Kaman for the sequencing functions to enable the vendor to more accurately estimate hardware and software development costs. The microprocessor approach also has the advantage that it is easier to change sequences, time out intervals, etc., since hardware changes are not required.

Conventional digital random logic elements would have more predictable failure modes than a microprocessor, but lack the flexibility to make changes provided by the software, and would have similar development costs for the design of the logic interconnections. Random logic also has the same disadvantage as the analog devices, in that every additional logic function requires additional equipment.

Functional requirements relating to selection of actuators for control of blade pitch and control of yaw of the nacelle are discussed in detail under the drive system of Section 7.

Primary factors in the selection of sensors for the WGS were life and reliability. For this reason, devices which use brushes and/or commutators are not as desirable as devices where no mechanical or sliding electrical contact is needed as a part of the sensing operation. This tends to favor optical or brushless AC sensors, where outputs can be generated without sliding contacts. Sensors which must perform under emergency conditions or under conditions where power may not be readily available, should be arranged to be operated off an emergency battery or should be pure mechanical devices which require no external source of power.

5.4.3 Concept Evaluations

Since the cost and weight of the electrical portion of the control system are only a small percentage of the total system cost and weight, these were not major factors in the evaluation of control system concepts. The weight of the control system is less than one percent of the total system weight, and the cost is less than 7% of total system cost. Therefore, reductions of control system cost by as much as 50% will only affect the overall system cost by a few percent.

The major consideration in the evaluation of the candidate control system concepts was reliability, and the analysis of various failure modes that could occur in the control system and their effects on system operation and safety. Failure modes and effects analyses were performed during the concept selection process to aid in selection of feasible approaches and to eliminate those approaches that would have failure modes unacceptable for the WGS application. The failure modes and effects (FMEA) analysis is presented in Appendix C to this report and will not be repeated here. Results of the FMEA were used, in conjunction with the selection criteria of 5.4.1 and the functional and equipment considerations of 5.4.2, to select the controls concepts presented below.

5.4.4 Selected Concepts

A summary of the selected concepts incorporated into the preliminary design of the Controls Subsystem is given below:

1. The primary pitch and yaw actuators are hydraulic devices supplied by a variable displacement hydraulic pump directly driven off the main gearbox, as described in the Drive Subsystem, Section 7. This makes the actuators independent of external electric power. An auxiliary, electrically driven pump is provided for maintenance or when the rotor is shut down.

2. The electronics to control the primary pitch and yaw actuators are based on analog techniques using standard analog building blocks.

3. Pitch controls are critical to WGS safety. Pitch linkages are simple, reliable and conservatively designed.

4. Control actuators must be rate limited. This allows more time to detect improper control motion in the event of a hardover failure of the control system. It also limits the distance travelled by the controls before the failure is detected and corrected, thus reducing the chance of damage to other components of the system. The rate-limiting feature is inherent in the actuator itself and does not require external rate sensing or electronics.

5. Sensing devices avoid the use of sliding contacts and/or slip rings, insofar as possible.

6. All critical sensing, signal processing and actuation functions are continuously monitored for failures, and appropriate action taken to prevent secondary damage in the event of a failure. This includes, as a minimum, the following:

    a. Blade Pitch Servo Control

    b. Yaw Servo Control

    c. Wind Speed and Direction Sensing

    d. Rotor RPM Sensors

    e. Critical Portions of Startup and Shutdown Sequences (SSC control)

7. A pure mechanical system is provided to initiate shutdown of the rotor independent of any sources of external power. (See Drive Subsystem, Section 7.)

8. A microprocessor-based computer system is used to control startup and shutdown sequencing and telemetry and supervisory functions. Critical functions performed by this device are continuously monitored for failure by a set of external logic or by a redundant microprocessor which monitors critical steps in the sequences.

5.5 Control System Preliminary Design and Analysis

Based on the evaluation and selection of the control system equipment candidate concepts described above, the preliminary design of this subsystem was developed. A description of the Controls Subsystem and the supporting analyses are detailed in the following subparagraphs. This description covers the electronic and electrical portions of the controls; the mechanical and hydraulic portions of the system are described in Section 7, Drive Subsystem.

5.5.1 Design Description

The design description of the control system is divided into four sections. The first section describes the primary control equipments; the second section describes the supervisory and sequencing control equipments; section three describes the recording equipment; and section four covers requirements for safety interlocks.

5.5.1.1 Primary Control Equipments

These equipments perform the following functions:

a. Control of the yaw orientation when the WGS is operating

b. Control of the speed, torque and load on the rotor system

c. Primary sensing of wind speed, wind direction, rotor rpm and blade pitch position

d. Fault monitoring of the primary control functions.

5.5.1.1.1 Yaw Servo Control

The primary function of the yaw servo is to keep the shaft axis of the rotor aligned with the average wind when the rotor is turning. The yaw rate is limited to approximately 1/3 RPM to prevent sudden motions of the tower head which could result in large forces on the system due to gyroscopic and/or moment of inertia effects. The yaw servo is only intended to trim the system to the average wind direction.

The presently selected 1/3 rpm (2 deg/sec) rate allows the rotor to follow small wind shifts of the order of 10 to 20 degrees in a matter of a few seconds after the actuator is energized, and causes gyroscopic loads which are equal to about 10% of the maximum expected loads on the orientation mechanism. Rates faster than those selected for the preliminary design could be used, but significantly faster rates would result in increased cost of the hydraulics, the rotor hub and orientation mechanism. Rates slower than the selected 1/3 rpm could be used, but significantly lower rates would result in the loss of some of the available wind energy due to the slower response of the yaw system to changes in wind direction. This could be the subject of a trade study during the detail design, trading off the value of the wind energy recovered versus the effect of the yaw rate on capital and energy cost. This effort was beyond the scope of the preliminary design.

The primary controls provide solenoid valve signals for the yaw servo motor to orient the WGS rotor into the wind when the WGS is operating. A schematic block diagram of the yaw servo is shown in Figure 5-1. Design of the equipment is such that when no signal input is received, the motor remains stationary. Two outputs are provided; one for clockwise rotation of the tower head, and one for counterclockwise rotation. Manual override provisions are included for maintenance and/or troubleshooting. It is not normally a requirement for the yaw servo to operate when the system is shut down, except for short periods to untwist the control cable, or for maintenance purposes.

Referring to Figure 5-1, the primary wind direction error signal from the synchro is introduced at the left of the diagram. The error signal then passes through two sets of contacts which are arranged to allow injection of + or - phase signals from the manual controls or from the SSC equipment to untwist the

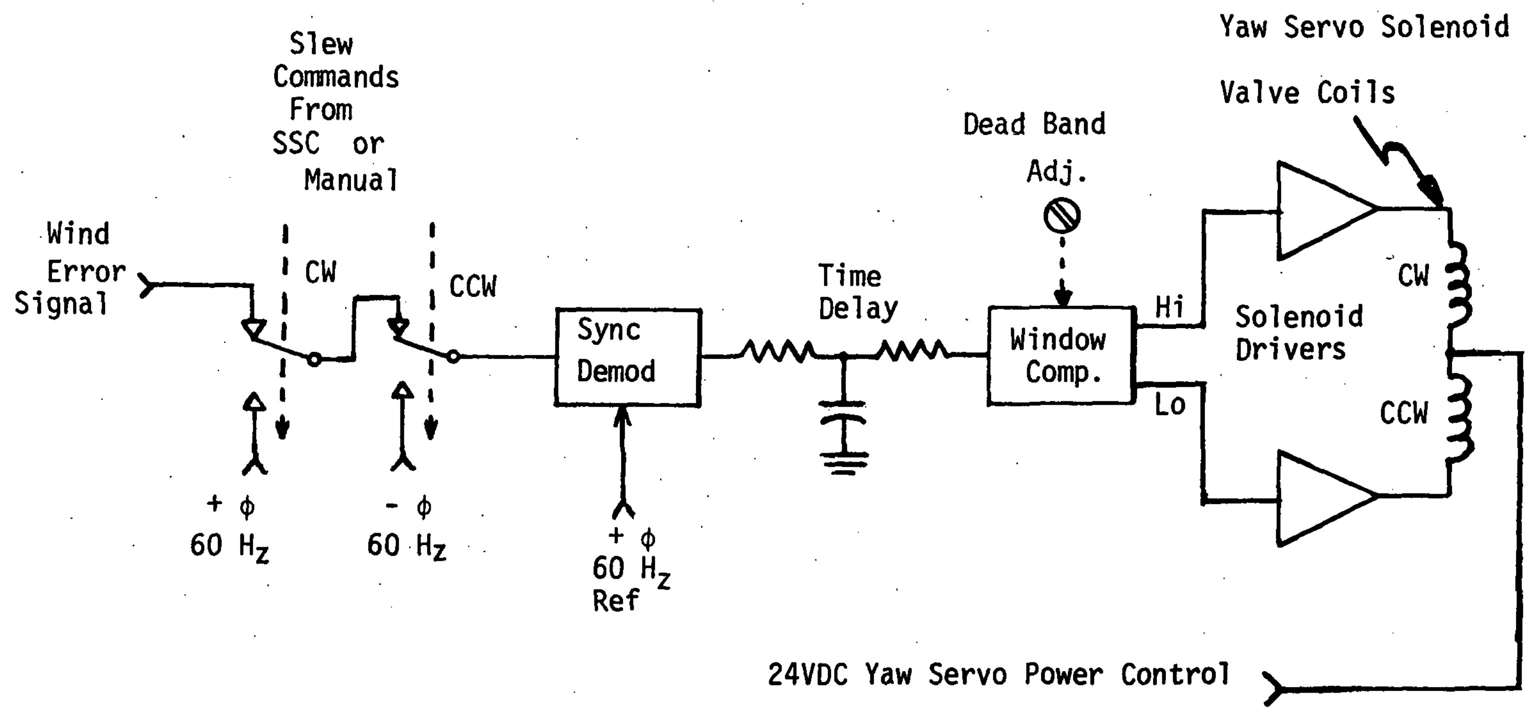


Figure 5-1. Yaw Servo Electrical Diagram

cables from the tower head after shutdown. A synchronous demodulator then converts this + or - phase AC signal to a + or - voltage DC signal. The signal is passed through a time delay network to prevent servo operation during momentary wind deviations and then to the input of a comparator (Burr Brown 4022/25 or equivalent). The comparator includes provisions for dead band adjustment and its output, in turn, operates the solenoid drivers (Sprague ULN 2064A or equivalent) which control the yaw servo motor. Dead band of the yaw servo electronics should be adjustable over a range of wind direction errors from $\pm$ 5° to $\pm$ 20° off the rotor shaft axis, since the final setting will depend, to a large extent, upon field experience with the WGS and also upon the wind characteristics at the site.

Continuous fault monitoring of the yaw servo during servo operation should be provided to prevent possible reverse wind and subsequent reverse thrust on the rotor blades (to preclude any possibility of a blade/tower intersection). It is suggested that this be accomplished by a monitoring device which checks for proper motion of the tower head, via a yaw tachometer, after establishing that a wind error of sufficient magnitude and direction has existed to trip the servo solenoids. This device includes non-linear time delays to assure that monitoring occurs after the yaw servo energizes and monitoring ceases before the yaw servo shuts off. The monitor is interlocked with the following signals as a minimum to prevent false alarms:

1. Wind speed must be above 10 MPH

2. Yaw servo must be energized

3. Hydraulic pressure must be "ON"

Configuration of the fault monitoring device is essentially the same as the yaw servo shown in Figure 5-1, except that the output consists of a comparator circuit, such as Burr Brown 4082/32, which compares the yaw tachometer with the monitor output. The fault monitoring device is supplied from a power supply which is independent of that used for the yaw servo. The same sensors may be used (with redundant wiring) providing all critical sensors have independent built-in fault sensing provisions.

The following additional monitoring functions are provided for the yaw servo by other equipments in the WGS system:

1. Servo is monitored by the SSC during the cable untwist sequence.

2. Excessive yaw rate at any time (beyond design limit) indicates mechanical failure in yaw servo drive train.

3. Vibration sensors provided for the rotor will also indicate yaw servo failures if resulting vibration is excessive.

4. Monitors provided for hydraulic/electric power provide backup fault detection for servos.

5. Wind direction error input to servo is monitored separately to provide backup fault protection.

5.5.1.1.2 Blade Pitch Servo Control

The pitch servo electronics provide a proportional DC output signal to operate the blade pitch servo valve. The electrical portion of the pitch servo is an operational amplifier with a booster capable of driving the valve coil. The amplifier summing bus is reconfigured for each mode of operation using relay contacts or solid state multiplexers. Blade pitch position is fed back to the summing bus via an LVDT or DC potentiometer. The LVDT is preferred for long life and low maintenance. The pitch servo provides proper pitch control for the following operating modes of the WGS system:

1. Start-up
2. Standby/Synchronize
3. Operate
4. Normal Shutdown

Each mode of operation is described in the following paragraphs:

1. Startup mode. Blade pitch should be positioned as close as practical to proper angle for maximum acceleration when starting the WGS to minimize time spent at resonances. To accomplish this, the servo summing bus is supplied with a pitch command signal from a function generator which accepts rotor RPM and wind velocity as inputs, and which generates a family of curves representing optimum blade pitch angle as a function of the above variables. A Burr-Brown 4302 multi-function converter is typical of a device which could be used to generate such a curve family.

2. Standby/synch mode. The method by which synchronization is accomplished is described below. First, the synchronizer matches the generator voltage to the network voltage by means of a control from the synchronizer to the generator field regulator. The synchronizer also exerts a fine control on rotor RPM to trim the average value of generator frequency to match that of the network. There will be small perturbations of RPM around this set due to wind gusting. The rotor pitch control system will limit the size of the perturbations, and the synchronizer will match the average frequency of the generator with the network. The synchronizer will monitor the relative phase of the generator and the network while these perturbations are taking place, and will close the breaker when the phase difference and rate of change of phase are within limits. Small changes in frequency and phase, due to small perturbations in the wind, actually work to our advantage since they create a continuously changing pattern of relative phase between the generator and network which aids the synchronization process. Larger gusts, which cause larger changes in RPM will cause rates of change of phase which exceed the allowable limits for synchronization. Even at the higher winds, the distribution of gusts

should be such that synchronization can still be achieved (i.e., all gusts will not be large). The disturbances produced by the large gusts will produce RPM variations; however, these should still be within safe limits since the control rates have been sized to safely handle the worst case gust conditions (reference 5.5.2.1). The extent and characteristics of RPM variations due to gusts which occur during the standby/synch modes should be further investigated during the final design.

Gross positioning of blade pitch is accomplished by providing the summing bus with a signal proportional to the component of wind speed along the axis of the rotor shaft. A tachometer signal fed to the summing bus then provides finer adjustment to hold RPM close to the rated set point. When the synchronizer is engaged, it provides a slow rate integrating control to closely match the average RPM with the network frequency. Provisions are made for a frequency compensating network on the tachometer input to the summing bus to assure control loop stability in this mode of operation. This configuration is illustrated in Figure 5-2.

3. Operate mode (generator connected to line). Gross positioning of blade pitch is accomplished by providing the summing bus with a signal which is a function of the component of wind speed along the axis of the rotor shaft. When the rotor is operating and supplying power to the network, this function is approximately equal to the magnitude of the axial wind speed above rated, raised to the .64 power. A DC torque signal derived from the generator power output monitor is used to provide a fine adjustment to hold the power output at the rated set point for wind speeds above rated. Provisions are made for a frequency compensation network on the torque signal input to the summing bus to assure stabilization of the control loop. This configuration is illustrated in Figure 5-3.

4. Direct feathering (normal shutdown). Provision is made for direct feathering, i.e., when a normal shutdown is desired, the servo valve coils are connected directly to a non-interruptable DC source through a set of relay contacts. These same contacts disconnect the valve coils from the normal servo amplifier to make the shutdown independent of servo amplifier and/or summing bus and sensor failures. This provision is illustrated at the output of the amplifier in Figures 5-2 and 5-3.

5. Fault monitoring. Continuous fault monitoring of the pitch servo during startup, standby/synch and operate modes is required to prevent possible overspeed/underspeed and/or reverse thrust on the rotor due to pitch servo failures. The monitor device must be capable of differentiating between pitch control failures and sudden or unusual motion of the controls due to wind gusts. It is suggested that this be accomplished by a monitoring device which checks for proper blade pitch position via an independent feed-back sensor. Average position of blade pitch should bear

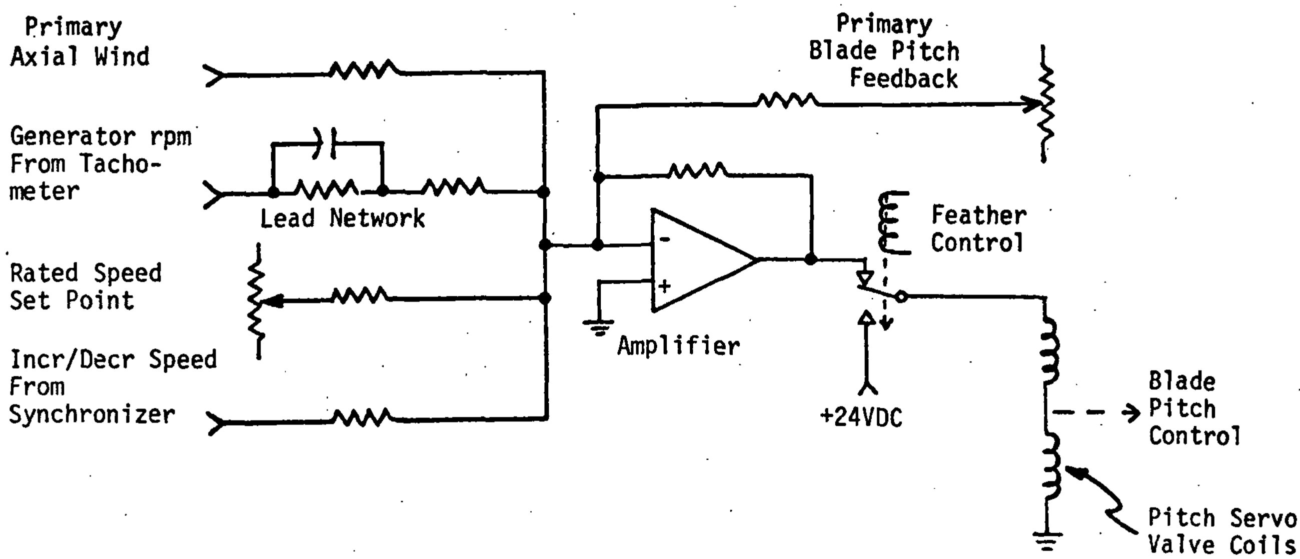


Figure 5-2. Pitch Servo Standby/Synch Mode

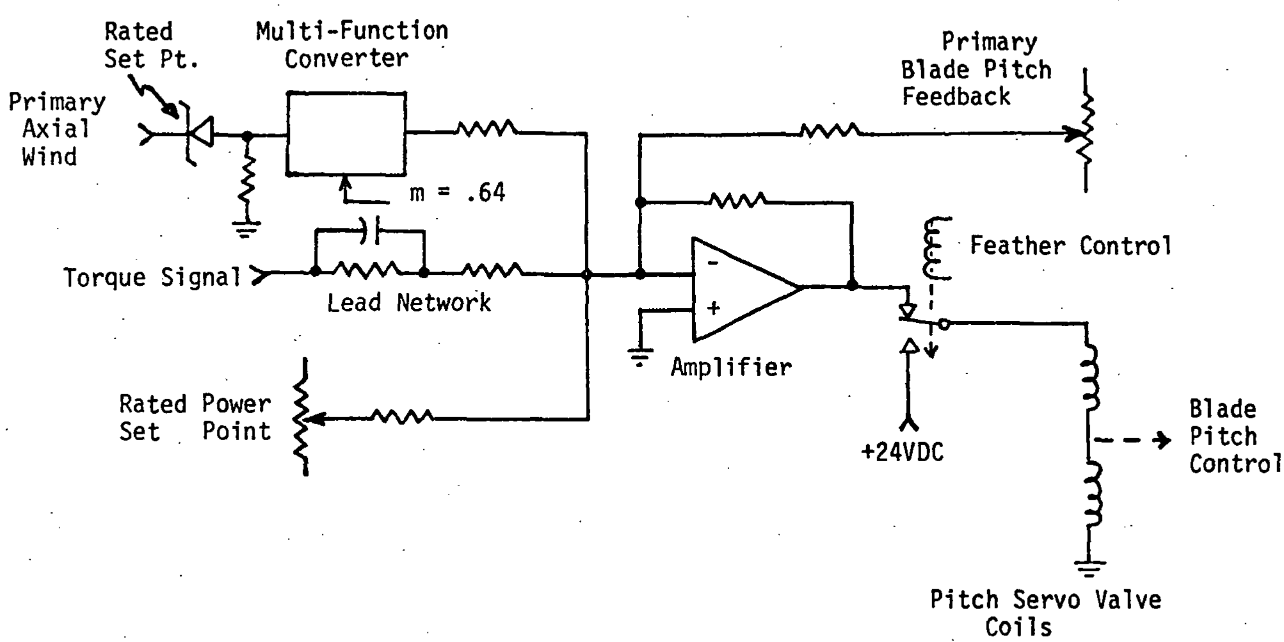


Figure 5-3. Pitch Servo Operate Mode

a direct relationship to wind speed in the standby/synch and operate modes, and should be a predictable function of rotor rpm and wind speed in the startup mode. Comparing this with the actual blade pitch position from an independent feed-back sensor allows monitoring for proper operation of the servo. Error bands must be set wide enough to prevent false alarms. Monitoring only occurs when the servo is actually engaged and is not in the shutdown mode. The fault monitoring device and feed-back sensor are supplied from a power supply which is independent of that used for the pitch servo. The same input sensors may be used (with redundant wiring) providing all critical sensors have independent built-in fault sensing provisions, as described herein.

The following additional monitoring functions are provided for the pitch servo by other equipments in the WGS system:

1. Servo is monitored by the SSC during the startup/shutdown sequences

2. Redundant rotor overspeed sensors can indicate failure of pitch control

3. Shaft torque (power output) is continuously monitored for sustained overtorque which could indicate failure of pitch control

4. Generator overtemperature monitor provides additional back-up for 3., above

5. Wind and rotor rpm sensors are monitored separately to provide back-up fault protection

6. Monitors provided for hydraulic/electric power provide back-up fault detection for servos

7. Pitch control forces monitored to provide advance warning of control jams.

5.5.1.1.3 <u>Wind Speed and Direction Sensing</u>

Wind speed and direction are sensed using a primary instrument which is equivalent to the Bendix Model 120 Aerovane. A second reference instrument is also used for fault monitoring purposes, since gross errors in wind speed or direction sensing are potentially hazardous to the WGS. The synchro signal representing wind direction, with respect to the rotor shaft axis, is fed through a transolver to obtain both the normal error signal (proportional to the sine of the direction angle) and a complementary output proportional to the cosine of the direction angle, as illustrated in Figure 5-4. The cosine output is used as follows:

1. To assure that a reverse wind direction does not exist at the rotor head by confirming that the sign of the cosine function is positive under all operating conditions and before start-up is allowed. (A negative sign indicates a 90° to 270° wind error.)

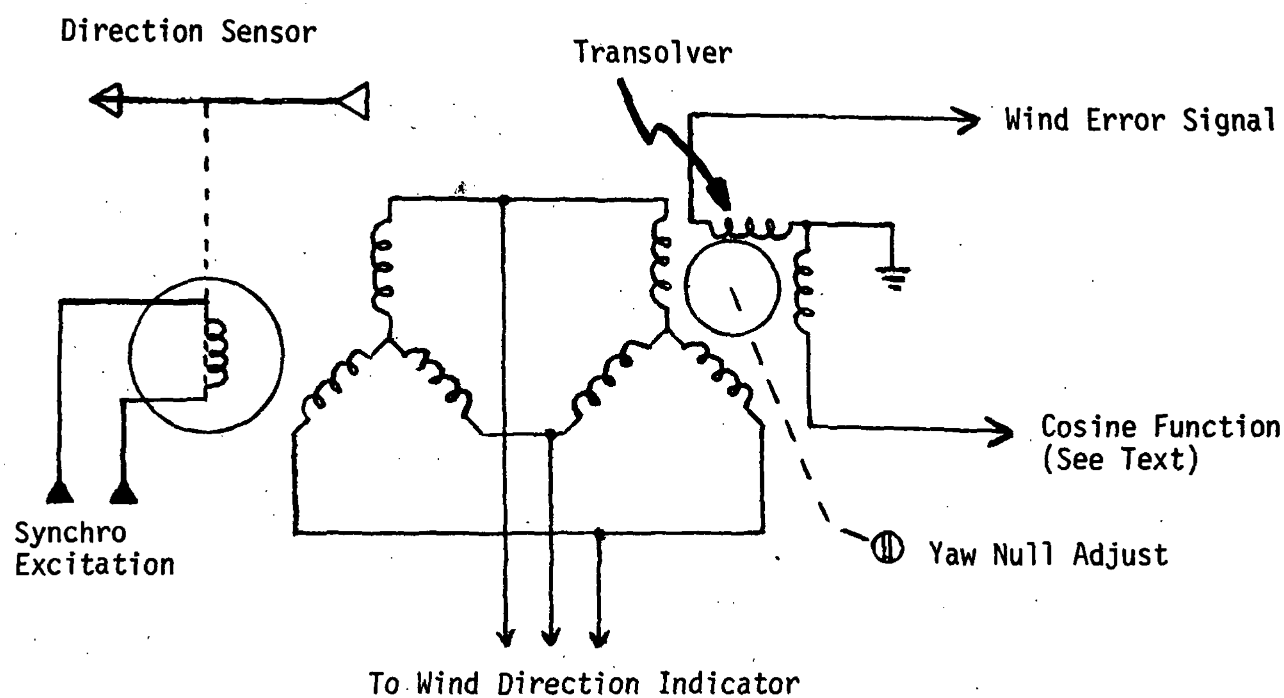


Figure 5-4. Wind Direction Sensing

2. The product of the cosine function and the wind speed magnitude is used to compute the component of wind speed along the axis of the rotor shaft, which is required as an input to the pitch servo and pitch servo monitor, the on-site recorder, and the generator field forcing circuit.

3. This same component of velocity is also computed using the reference instrument. The components from the primary and reference instruments are then compared through a window comparator to detect any gross wind sensing errors which could be hazardous to the WGS.

Both the primary and reference instruments can trip overspeed devices, either of which are capable of initiating shutdown of the system if the average wind exceeds a preset value (approximately twice rated).

An indicator, similar to Bendix Model 135, is provided to afford visual readout of wind speed magnitude and direction with respect to the rotor shaft axis. A transfer switch is also provided so the indicator may be used to read outputs from either the primary or the reference sensors.

5.5.1.1.4 Power Monitoring

A bi-directional watt hour transducer is provided to measure generator power output and energy output (equivalent to Scientific Columbus Model DL324K5A2-6096). Energy output is supplied to the SSC (5.5.1.2) for telemetry to the central station. Power output is proportional to shaft torque, and is processed by the primary controls as follows:

a. If the average power or torque exceeds 115% of rated for more than one minute, a shutdown and lockout is initiated.

b. If the average power or torque becomes negative (generator absorbing power from the line) and remains so for more than two minutes, a shutdown is initiated.

c. A proportional torque signal is provided to the pitch servo as a means of controlling blade pitch when connected to the network.

5.5.1.1.5 RPM Monitoring

RPM monitoring is accomplished by means of redundant magnetic pickup sensors (one mounted on the rotor shaft and one on the generator shaft). Signals from these pickups are processed by means of two Airpax 300 series Control Tachometers, or equivalent, to provide the following functions:

1. Outputs of the two tachometers are smoothed and compared through a window comparator to provide a self-check of the speed sensing operation and simultaneously verify integrity of the drive shafts, couplings, and gearbox.

2. Proportional DC signal from the rotor shaft tachometer is used to provide inputs to the pitch servo monitor, the SSC and a visual indicator (Airpax Style D1, or equivalent).

3. Proportional DC signal from the generator shaft tachometer is used to provide inputs to the pitch servo and a visual indicator (Airpax Style D1, or equivalent).

4. Both tachometers include provisions for adjustable set points. These are used to provide contact closures at approximate rpm settings to the SSC for sequencing purposes, as well as upper and lower rpm limits for shutdown initiation, independent of the SSC.

5. If an induction generator is used, rpm set points on the tachometer are used as a back-up for the speed matching interlock (Section 8, herein) to match the induction generator shaft speed to the network before closing the breaker.

5.5.1.1.6 Failure Signal Processing

A summary of the fault sensors located throughout the WGS system, and the action to be taken by the control system in the event of various failure categories is specified in Tables 5-1 through 5-4. The three center columns in each table show the category or classification of the failure for telemetry back to the master station, the type of shutdown required and the requirement, if necessary, to lock out or prevent further operation after shutdown. Failures are classified in the first of these columns into four categories: mechanical (M), electrical (E), fire (F), and icing (I). These categories were suggested by Northeast Utilities as sufficient to determine the type of personnel needed at the site to perform repairs and restore operation after the failure. A dash (-) in this column indicates that no repair is required, as in the case where shutdown is occasioned by low wind. In the second column, failures are classified as being one of two types: normal or emergency. Normal shutdown uses the standard hydraulic pitch control system. Emergency shutdown, employed when normal control has been rendered inoperative by the failure, uses the emergency feathering device. A dash (-) in this column indicates no shutdown is required. The third column indicates whether the WGS must be "locked out" and prevented from restarting until the failure has been repaired (YES); or whether lockout is unnecessary and it is safe to automatically restart the WGS when conditions are proper for a restart (NO).

5.5.1.1.7 Emergency Shutdown

Where an emergency shutdown is required, as defined in Tables 5-1 through 5-4, this should be accomplished using the mechanical energy stored in the rotating rotor to change blade pitch. External electrical or hydraulic power should not be required to initiate an emergency shutdown. The equipment should be designed such that a shutdown is initiated, automatically, when hydraulic power is lost. The details of how this is accomplished are covered in the Drive System, Section 7.

TABLE 5-1. FAILURE REPORTING AND ACTION CONTROLS

FAILURE CLASS	TELEM. CLASS	SHUTDOWN CLASS	LOCK-OUT	COMMENTS
Average torque exceeds 115% of rated for 1 minute	M/E	NORMAL	YES	Controls failure
Average torque negative for 2 minutes	-	NORMAL	NO	Low wind shutdown
Wind sensor failure	M/E	NORMAL	YES	Advisory only for Dev. Systms. (no shutdown)
Average wind exceeds operating limit	-	NORMAL	NO	High wind shutdown
Pitch servo failure	M/E	EMERGENCY	YES	Advisory only for Dev. Systms. (no shutdown)
Yaw servo failure	M/E	NORMAL	YES	Advisory only for Dev. Systms. (no shutdown)
Rotor brake will not release	M/E	NORMAL	YES	Do not start WGS
Pitch angle will not initialize	M/E	NORMAL	YES	Do not start WGS
Rotor will not accelerate	-	NORMAL	NO	Wind may not be adequate
RPM will not hold	-	NORMAL	NO	Wind may not be adequate
No generator voltage (synch only)	E	NORMAL	YES	
Synch aborted	M/E	NORMAL	YES	System will not synchronize within 20 minutes
Shutdown power fault	M/E	EMERGENCY	YES	Power will not reduce on shutdown command
Main breaker fails to trip	E	NORMAL	YES	Provides signal to trip utility feeder breaker
Deceleration fault	M/E	EMERGENCY	YES	Rotor fails to decelerate properly
Rotor brake does not apply	M/E	EMERGENCY	YES	Possible failure of emergency battery supply or brake
Rotor will not stop completely	M/E	EMERGENCY	YES	Possible failure of emergency battery supply or brake or controls
Hydraulics/emergency feathering fault	M/E	EMERGENCY	YES	
Reverse wind	M/E	NORMAL	NO	Do not start - shut down if operating
RPM < 80% during synch/standby	-	NORMAL	NO	Low wind
No pitch rate on shutdown	M/E	EMERGENCY	YES	Controls failure

TABLE 5-2. FAILURE REPORTING AND ACTION, ELECTRICAL AND GENERATOR PROTECTIVE CONTROLS

FAILURE CLASS	TELEM. CLASS	SHUTDOWN CLASS	LOCK-OUT	COMMENTS
Overvoltage (induction only)	-	-	-	Trip breaker and cap switch and go to standby mode
Under and over-frequency (synch only)	-	-	-	Trip breaker and go to standby mode
Overcurrent and/or phase unbalance	-	-	-	Trip breaker and go to standby mode
Reverse power > 20% rated	M/E	NORMAL	YES	Possible controls failure plus reverse thrust
Differential Current trip	E	NORMAL	YES	Electrical fault inside WGS
Excitation loss (synch only)	E	NORMAL	YES	Electrical fault inside WGS
Overvoltage (synch only)	E	NORMAL	YES	Electrical fault inside WGS
No station service power	E	NORMAL	NO	Shutdown after time delay of 1 minute
No charge alarm (sta. service power ok)	E	EMERGENCY	YES	
Emergency battery breaker alarm	E	EMERGENCY	YES	
Generator bearing overtemperature	E	NORMAL	YES	
Generator winding overtemp. (redundant)	E	NORMAL	YES	
Power transformer overtemperature	E	NORMAL	YES	
Excess cable windup	-	NORMAL	NO	
Fire detector (ionization)	F	EMERGENCY	YES	

TABLE 5-3. FAILURE REPORTING AND ACTION, DRIVE SYSTEM/HYDRAULICS

FAILURE CLASS	TELEM. CLASS	SHUTDOWN CLASS	LOCK-OUT	COMMENTS
Gearbox Chips	M	NORMAL	YES	
Gearbox Overtemperature	M	NORMAL	YES	
Gearbox Low Oil Level	M	NORMAL	YES	
Gearbox Low Oil Pressure	M	NORMAL	YES	
Output Overspeed 5%	-	-	-	Actuate feathering relay
Output overspeed 5% and no pitch rate	-	EMERGENCY	YES	Dump hydraulic pressure
Output underspeed < 80%	-	NORMAL	NO	Speed checked 5 minutes after startup
Low hydraulic fluid level	M	NORMAL	YES	
Low hydraulic pressure (while operating)	M/E	NORMAL	YES	
Excess yaw rate	M/E	EMERGENCY	YES	
Speed Error (Rotor/Generator)	M/E	EMERGENCY	YES	Sheared shaft or coupling or failed speed sensors

TABLE 5-4. FAILURE REPORTING AND ACTION, ROTOR

FAILURE CLASS	TELEM. CLASS	SHUTDOWN CLASS	LOCK-OUT	COMMENTS
Excessive vibration	M	NORMAL	YES	No lockout if combined with icing
Excessive pitch control force	M	EMERGENCY	YES	Potential jam condition
Rotor overspeed - 5%	-	-	-	Actuate feathering relay
Rotor overspeed 5% and no pitch rate	-	EMERGENCY	YES	Dump hydraulic pressure
Rotor underspeed < 80%	-	NORMAL	NO	Speed checked 5 minutes after startup
Icing	I	-	-	(Prevents lockout for high vibration [see above])

5.5.1.2 Sequencing and Supervisory Functions

Sequencing and supervisory functions handled by the control system are described below.

The equipment selected to perform these functions is an adaptation of the microprocessor-based SCS 4002 equipment, which is a radio-operated or wireline Supervisory Control System developed by Data Signal Corporation. Data Signal equipment is presently being used by several companies in the New England area. The WGS version will include a remote Supervisory and Sequencing Control (SSC), which is located at the remote WGS site, and a master station which may control and monitor several remote WGS sites. This report describes both units, hereafter called the "remote" and "master" stations, respectively.

Tables 5-5 and 5-6 show the inputs and outputs required by the remote station SSC to provide the sequencing functions needed for startup/shutdown and synchronization of the WGS at the remote site. The monitoring and supervisory functions are shown in Table 5-7. The items of Table 5-7 represent the minimum data which must be telemetered to the master station, as recommended by Northeast Utilities. Since the microprocessor type of system can handle additional data items with very little added cost, Kaman is recommending that the additional items listed below also be available for callup by the operator at the master station. Field experience with the WGS system may modify this list:

1. Hydraulic Pressure
2. Windspeed (Axial or Total)
3. Wind Direction
4. Power Output (VARS or KW)
5. Rotor RPM
6. Blade Pitch Angle
7. Station Service Status
8. Bus Voltage
9. Generator Voltage
10. Generator Field Current
11. Yaw Rate
12. Operating Mode

The sequencing functions which must be performed by the remote station microprocessor, were defined by preliminary flow diagrams. These flow diagrams were prepared primarily to define the relative complexity of the function to be performed by the microprocessor and to allow the cost and lead time of the equipment

TABLE 5-5. INTERFACE DEFINITION - WIND GENERATOR SYSTEM

WIRED INPUTS TO MICROPROCESSOR REMOTE STATION FOR SEQUENCING ROUTINES

LEGEND

D = Digital M = Manual Control
A = Analog
I = Interrupt T = Discrete

SIGNAL IDENTIFICATION	TYPE OF SIGNAL	EXPLANATION, NOTES
Station Service Power	DT	Indicates presence/absence of power
External Lockout	DT	Inhibits restart if unsafe - Reset manually
Restart Discrete	DT	Inhibits restart if unsafe - Automatic reset
Average Total Wind Speed	A	Averaging external to SSC
Wind Monitor Override	DTM	Overrides wind monitor and holding function
Wind Direction Error	A	Wind deviation from axis of rotation
Blade Pitch Angle	A	0° = Flat pitch, 90° = Feathered
Rotor Brake Applied	DT	Absence of signal indicates brake in transition
Rotor Brake Released	DT	Absence of signal indicates brake in transition
Rotor RPM	A	
Generator Output Voltage	A	
Main Breaker Status	DI	Main breaker tripped/closed
Cable Twist	A	Measures cable twist ($\pm$ 10 turns)
Real Power Delivered	A	Power being delivered to utility (+ or -)
Null Check, Reverse Wind	DT	Confirms proper yaw orientation
Rotor Shaft Angle	A	Used to park blades, horizontally
Shaft Angle Override	DTM	Overrides horizontal blade parking function

TABLE 5-5. INTERFACE DEFINITION - WIND GENERATOR SYSTEM (continued)

WIRED INPUTS TO MICROPROCESSOR REMOTE STATION FOR SEQUENCING ROUTINES

LEGEND

D = Digital M = Manual Control
A = Analog
I = Interrupt T = Discrete

SIGNAL IDENTIFICATION	TYPE OF SIGNAL	EXPLANATION, NOTES
Shutdown Delay Override	DTM	Overrides delay function after shutdown
Standby Interrupt	DI	Externally commands Standby mode
Safety Interrupt	DI	Externally commands shutdown - with lockout
Lockout Reset	DTM	Manually resets internal lockout flag
Shutdown Interrupt	DI	Externally commands shutdown - no lockout
Manual Synchronous Enable	DIM	Manually enables synchronizer control

TABLE 5-6. INTERFACE DEFINITION - WIND GENERATOR SYSTEM

WIRED OUTPUTS FROM MICROPROCESSOR REMOTE STATION FOR SEQUENCING ROUTINES
(5 AMP CONTACTS UNLESS NOTED)

LEGEND

D = Digital
T = Discrete
I = Interrupt
V = Visual Indication (LED)

SIGNAL IDENTIFICATION	TYPE of SIGNAL	EXPLANATION, NOTES
Blade Pitch to Feather	DT+V	Feather command through hydraulics
Auxiliary Pump Motor Control	DT+V	Provides hydraulic power when rotor is stopped
Startup Flag	DT+V	Enables Startup mode
Apply Rotor Brake	DT+V (25 amp)	Absence of signal indicates brake to hold previous position
Release Rotor Brake	DT+V (25 amp)	Absence of signal indicates brake to hold previous position
Starting Augmentation Control	DT+V (25 amp)	Energizes generator as starting motor
Pitch Timeout Fault	DT+V	Failure of pitch angle to initialize
Starting Timeout Fault	DT+V	Failure to start in preset time
RPM Timeout Fault	DT+V	Failure to hold RPM after start
Generator Field Control	DT+V	Energizes generator field
RPM Limit Monitor Control	DT+V	Enables/disables RPM limit monitor
Synchronizer Flag	DT+V	Indicates Synchronization mode
Voltage Timeout Fault	DT+V	Failure of generator voltage to initialize
Synchronize Abort	DT+V	Synchronization not completed in preset time
Operate Flag	DT+V	Enables normal operating mode
Shutdown Flag	DT+V	Enables Shutdown mode

TABLE 5-6. INTERFACE DEFINITION - WIND GENERATOR SYSTEM (continued)

WIRED OUTPUTS FROM MICROPROCESSOR REMOTE STATION FOR SEQUENCING ROUTINES
(5 AMP CONTACTS UNLESS NOTED)

LEGEND

D = Digital V = Visual Indication (LED)
T = Discrete
I = Interrupt

SIGNAL IDENTIFICATION	TYPE of SIGNAL	EXPLANATION, NOTES
Synchronizer Control	DT+V	Enables automatic synchronizer and verifier
Inching Drive Control	DT+V (25 amp)	Engages inching drive to park blades
Yaw Servo Control	DT+V	Enables electric power to servo electronics
Pitch Servo Control	DT+V	Enables electric power to servo electronics
Shutdown Power Fault	DT+V	Dump hydraulic pressure and lockout WGS
Deceleration Fault	DT+V	Dump hydraulic pressure and lockout WGS
Braking Fault	DT+V	Release brake and lockout WGS
Shutdown Delay Flag	DT+V	Indicates Delay mode after shutdown
Monitor Flag	DT+V	Enables Wind Monitoring mode
Trip Main Breaker	DI+V (25 amp)	Trips main breaker
Standby Flag	DT+V	Indicates Standby mode
Yaw Servo Clockwise	DT+V	Commands tower head clockwise
Yaw Servo Counterclockwise	DT+V	Commands tower head counterclockwise
Main Breaker Failure	DT+V	Trip back-up protective devices (optional)
Close Main Breaker	DI+V (25 amp)	Closes main breaker

TABLE 5-7. INTERFACE DEFINITION - WIND GENERATOR SYSTEM

ADDITIONAL WIRED INPUTS AND OUTPUTS AT REMOTE STATION

TO PROVIDE TELEMETRY AND SUPERVISORY FUNCTIONS

LEGEND

A = Analog D = Digital
I = Interrupt T = Discrete

SIGNAL IDENTIFICATION	TYPE OF SIGNAL	EXPLANATION, NOTES
Inputs to Remote Station (to be displayed at Master Station)		
Real Power Delivered	A*	Peak value once per hour (6 - 10 minute option)
Megawatt Hours	A (5 sig.)	Once per hour
Main Breaker Status	DI*	As occurs
Station Service Status	DT*	As occurs
Fault Monitoring Inputs (to be displayed at Master Station)		
Hydraulic/Mechanical	DT	As occurs
Electrical	DT	As occurs
Fire	DT	As occurs
Icing	DT	As occurs
Hazardous Condition	DT	As occurs (immediate attention required to preclude further damage)
Voice-Input/Output	A	Standard telephone interface (radio link to Master Station)
Outputs (to be controlled by Master Station)		
Startup Inhibit	DI*	Prevents startup of system if commanded by Master Station or initiates shutdown if commanded

* This signal is already being provided to the microprocessor Remote Station for sequencing functions.

and software development to be estimated. Development of the final flow diagrams is beyond the scope of the preliminary design and should be completed as a part of the detail design effort.

5.5.1.2.1 Monitoring of Sequencing Functions

Critical sequencing functions must be redundantly monitored to prevent damage to WGS equipment in the event of a failure of the primary SSC equipment. This may be accomplished either with an additional redundant microprocessor set up only to monitor these critical sequences, or by an independent set of conventional logic elements and time delays to accomplish the same result. The critical sequencing function which must be monitored according to the failure modes and effects analysis performed during the preliminary design are listed below. A detailed failure modes and effects analysis should be developed as a part of the detailed design to identify more specifically the monitoring requirements for the sequencing functions.

1. Failure to Initiate Shutdown Sequence (Hangup in operate, standby, or sync modes)

2. Hangup in startup mode

3. Inadvertent startup

4. Oscillation between modes

5. Does not alarm for critical failure

5.5.1.2.2 Monitoring of Supervisory Functions

Supervisory and telemetering functions will be protected against noise and monitored for faults by several redundant features similar to those which presently exist in the SCS 4002 system. These are listed below:

1. Redundant tone encoding of each bit (touch tone system)

2. Tones must be within 3 db in amplitude to be accepted

3. Interdigit timing, next tone burst must be within time interval

4. Every word has escape key, which must be present to be accepted

5. Every message must be of proper coded length to be accepted

6. Automatic exchange of test data word format:

   Checks both master and remote stations
   Checks both telemetry links (Data and Supervisory)
   Checks both data and address functions
   Automatic alarm if check fails

5.5.1.3 Data Recording

Data to be recorded at the WGS site consists of the following:

1. Meteorological data (for efficiency evaluation)
2. Power output - taken at same time as meteorological data
3. Total number of hours on line
4. Failure monitoring data (flag indicators)
5. Rotor speed
6. Pitch angle
7. Continuous windspeed
8. Mode and fault signals

(Items 5–8: Time history only during overspeed and/or emergency shutdown conditions)

9. Rotor torque
10. Pitch Angle
11. Continuous windspeed
12. Mode and fault signals

(Items 9–12: Time history only during overtorque conditions)

5.5.1.3.1 Recording Equipments

Discrete data, such as failure signals, are recorded on magnetically actuated flag indicators which do not require electric power to hold information, (i.e., non-volatile memory). Flag indicators are reset manually when the fault is repaired. All other analog and/or digital data can be recorded on a magnetic recorder similar to Datel Systems Model LPS-16-10B, or equivalent.

5.5.1.4 Safety Interlocks

Certain combinations of conditions should never be allowed to occur, because of the potential of damage to equipment. It is recommended that interlocks be provided to safeguard against the conditions listed below. These interlocks are simple electromechanical devices, limit switches, etc., and should be independent of the SSC:

1. The rotor brake should not be able to be applied unless the rotor rpm is below 15% of rated.
2. The inching motor should not be able to be operated until the rotor brake is fully applied and rotor has stopped.
3. WGS must always be in the synchronize mode just prior to closing the main breaker.

4. Check for low rpm 5 minutes after startup, and initiate shutdown if rpm is below specified limit.

5. Microprocessor should not be able to control yaw servo (to untwist cable) unless WGS is shut down.

5.5.2 Design Analysis

Primary emphasis was placed on the pitch control from an analysis standpoint, since this control must be sufficiently responsive to follow wind gust behavior. Results of this analysis were also needed to establish the size and duration of the torque impulses to be used in the Northeast Utilities stability analysis of the generator connected to the network. Previous experience with rotor controls indicates that stabilization of the servo control loops for yaw control, rpm control and torque control should not present any difficulty. Detailed stability analyses of the control loops which do not include a dynamic model of the rotor could be misleading, therefore, a dynamic control model of the rotor should be developed during the detail design of the WGS to support this analysis. Since rate feedback terms in the primary control loops should stabilize any of the above control loops, provisions have been included for rate sensing and rate inputs in the preliminary design of the primary control loops.

5.5.2.1 Effect of Wind Gusts on Pitch Control

The wind gust model used for this analysis is shown in Figure 5-7. It was based on Reference 5-1, and was supplied to Kaman by NASA. The model specifies a cosine shaped gust superimposed on the average wind. The magnitude of the gust varies primarily as a function of the average wind and gust period. Gusts can be introduced into the system when the generator is connected to the network, or when it is disconnected from the network. The distribution of gust periods is not defined by the model, therefore, a range of periods was investigated to find those with the most severe effect on the WGS.

When the generator is connected to the network, an increasing gust will cause increasing torque and power output, whereas a decreasing gust will cause decreasing torque and power output, and may also cause a thrust reversal on the rotor if the pitch control system is not sufficiently responsive. Based on typical generator short term overload capability, short term overload due to increasing gusts should never be allowed to cause input torque to exceed 200% of rated. The control system must also be sufficiently responsive to prevent decreasing gusts from causing a thrust reversal of the rotor, to preclude any possibility of a blade/tower intersection.

Since a dynamic model of the WGS rotor is not available from the preliminary design effort, a full dynamic analysis of the WGS is not possible at this time. However, using the static control characteristic of the rotor (Figure 5-5), in conjunction with the wind gust model described above, it is possible to estimate the effect of gusts on the system and to estimate the control response needed to satisfy the above torque and thrust limits.

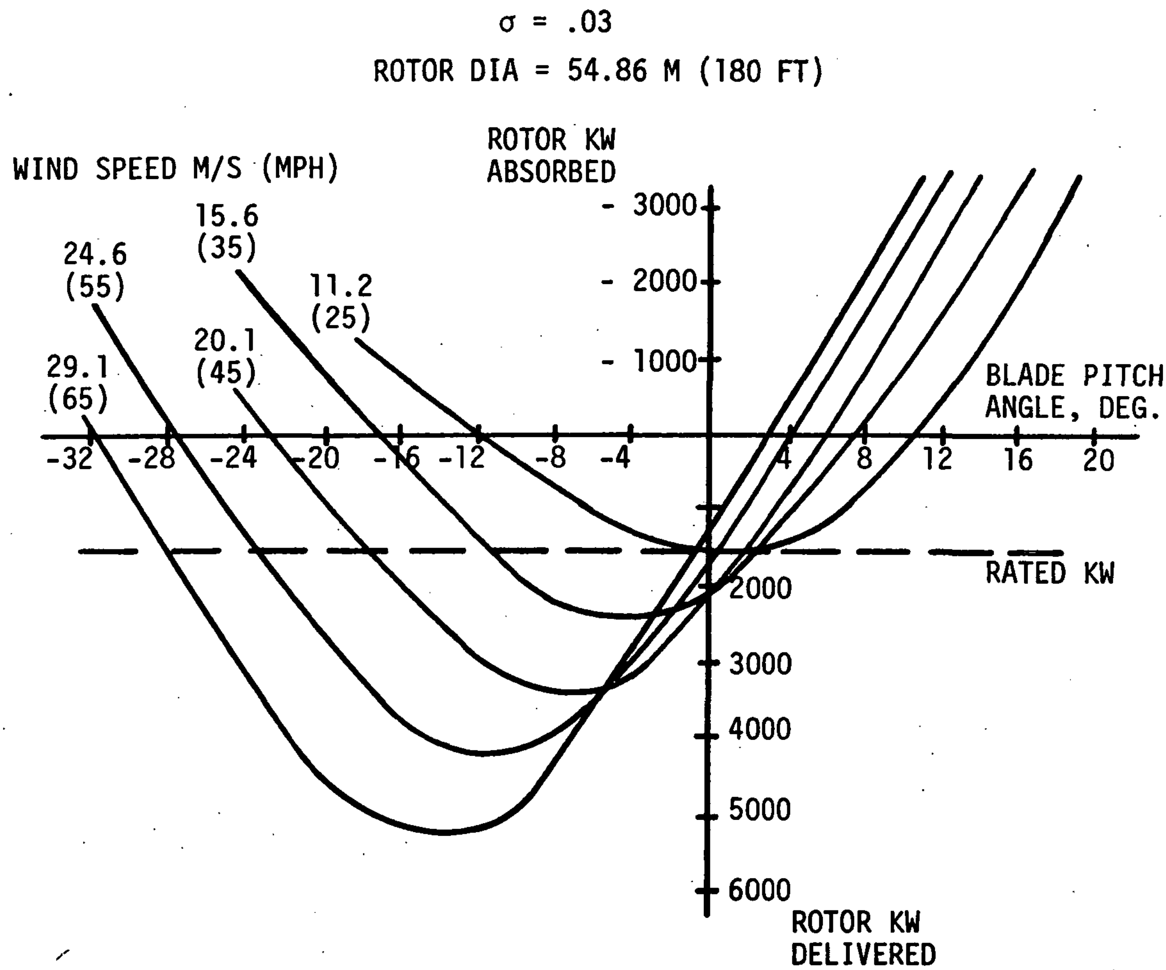


Figure 5-5. Rotor Characteristic Curves

This was accomplished as follows. At the highest steady state operating wind condition, which produced the largest gusts at each period, an operating point (pitch angle) was located on the curves to provide rated power output. A selection of gust amplitudes at various periods was then graphically superimposed on this steady state operating point with no pitch change. The change in power level (torque input) was noted. If the resulting power exceeded 200% of rated, a sufficient change in blade pitch angle was allowed to limit the torque to 200% of rated. It was assumed that the pitch control input was applied at a constant rate and had to reach its final value in 1/2 of the gust period, or by the time the gust reached a peak value. This determined approximate control rates and travel requirements at each gust period to stay within the established torque limits. A similar technique was used for decreasing wind gusts to establish the control rates necessary to prevent reverse thrust (negative power flow into the rotor). It must be realized that the accuracy of this technique is limited since it is based on a quasi-static analysis of the rotor and controls. The above procedures allow an approximate definition of the power required to operate the pitch controls, the required control rates and an approximation of the resulting torque impulses due to gust disturbances which are then used in the generator/network stability analysis of Section 8.

These results are plotted in Figure 5-6 for the 1500 KW system using the highest average operating windspeed for this system of 20.1 meters/second. This indicates the torque overload limit to be the controlling requirement on pitch rate and shows the worst case occurs for gusts of about 1.5 seconds period. Gusts faster than 1.5 seconds are smaller in amplitude and gusts slower than 1.5 seconds can be easily followed by the 5°/second maximum pitch rate. A similar plot for the 500 KW system indicates a maximum pitch rate requirement of about 3.3 deg/sec due to the lower maximum operating wind speed of 11.2 meters/second.

When the generator is disconnected from the network, the sudden loss of load combined with a gust can cause an overspeed condition. The worst case condition occurs if the WGS is hit with the gust at the instant the breaker is opened, since all of the power originally being delivered to the utility, as well as the gust energy, is now going into accelerating the rotor until the control system can change pitch sufficiently to spill the excess energy. It was decided that, to be compatible with readily available generating equipment, a maximum overspeed of 50% would be allowed under these worst case conditions. It was assumed that the gust was applied as a step function at the time of disconnecting the generator from the line and the gust was held at this level during the analysis. Several values of average wind speed and associated gust amplitudes and pitch angles were investigated using the rotor control characteristics over the speed range from rated to 150% of rated. The most severe overspeeds occurred when the pitch angle started out near zero degrees (flat pitch) and the wind was near rated, because the rotor can overspeed more readily near flat pitch and the largest pitch change is required to reach a stable rpm after the transient has stabilized.

The highest wind speed for flat pitch is rated wind speed. Above rated speed, the average pitch angle is moved toward feather to control power output. Therefore, the worst case potential for overspeed occurs at flat pitch with rated wind and the simultaneous application of the maximum gust corresponding to

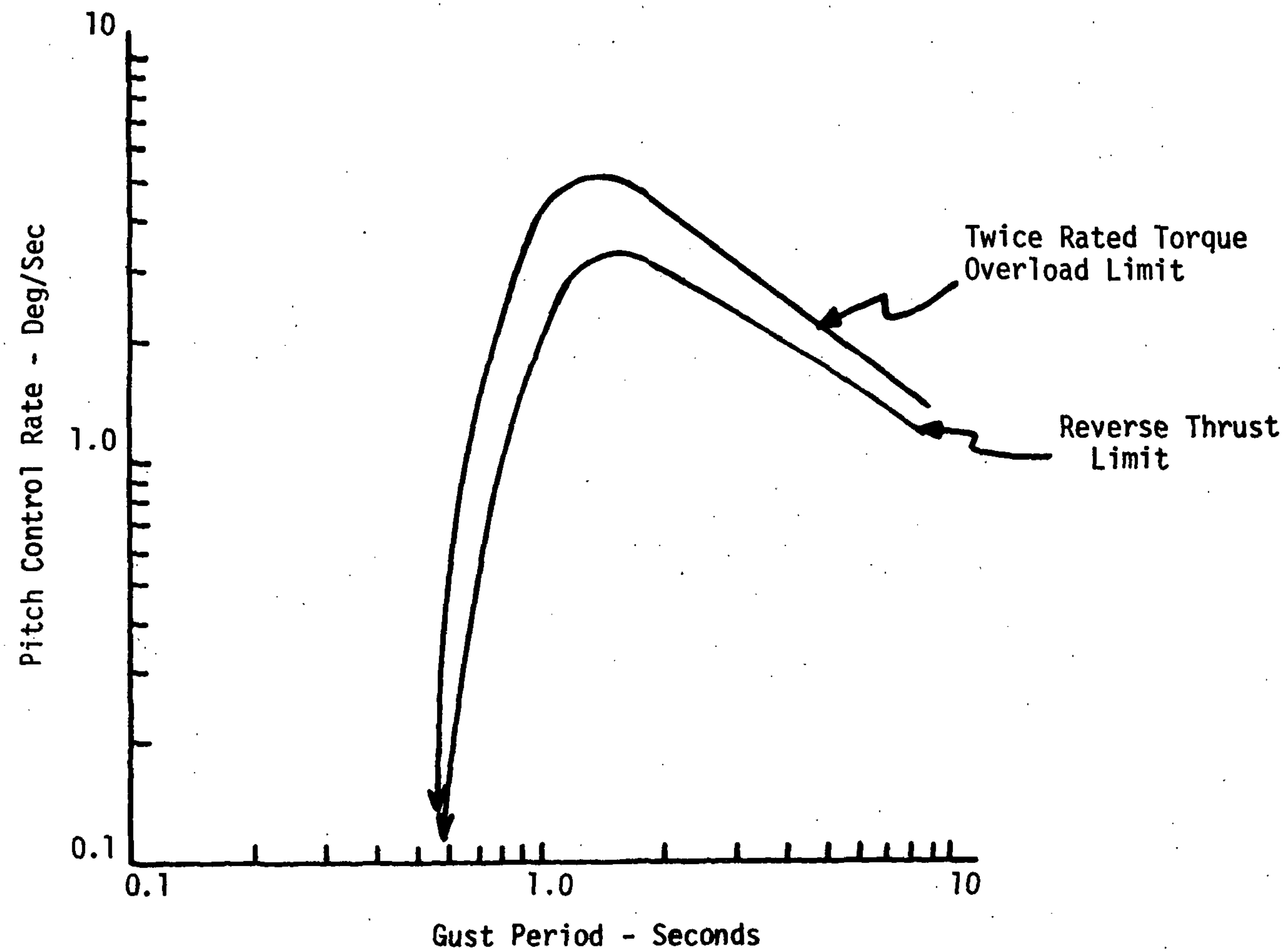


Figure 5-6. Required Control Rates to Prevent Overtorque and Reverse Thrust for 1500 KW System at 20 m/s Average Wind

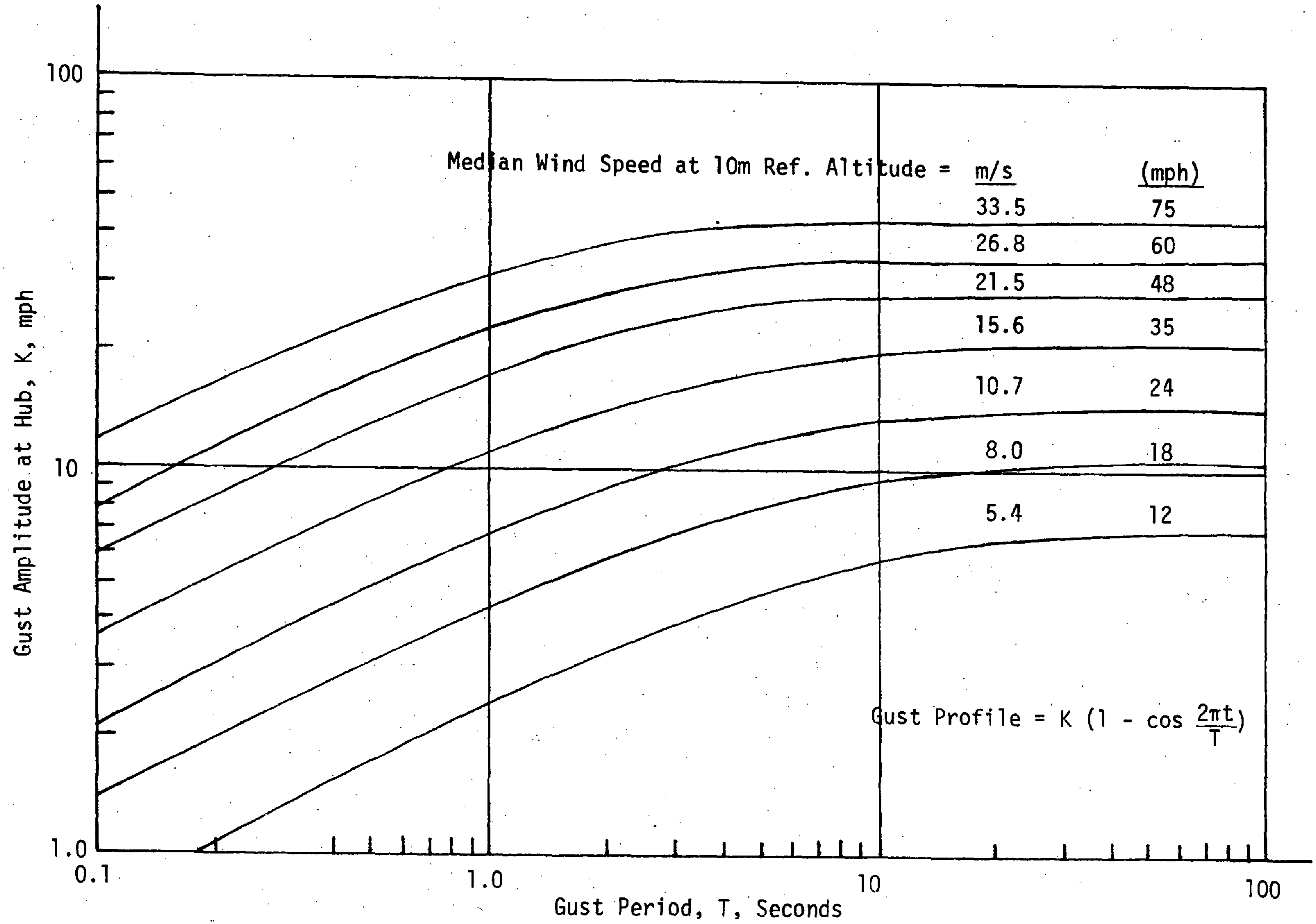


Figure 5-7. Longitudinal Wind Gust Model

trajectories under this condition for both the 1500 KW and 500 KW systems showed a pitch rate requirement for both systems of about 5 degrees/second to limit maximum overspeed to 150% of rated. Maximum speed was reached approximately two seconds after the breaker was opened. Coincidentally, this pitch rate corresponds to the 5 degrees per second previously calculated for the 1500 KW system with the generator connected to the line. Therefore, 5 deg/sec pitch rate was chosen as the preliminary design value for both systems. This control rate was then used in conjunction with the NASA wind gust model to determine the characteristics of the torque impulses for the generator stability analysis given in Section 8, and to size the hydraulic power requirements in Section 7.

It is not desirable to have a pitch control rate capability faster than necessary to perform the desired control functions. Higher rates would increase control system cost and power consumption, and would also increase the potential hazard of damage to other components of the system in the event of pitch control hardover failures. Limiting the pitch rates makes it easier to detect a hardover and take protective action (such as emergency feathering) before damage can occur. This is covered in greater detail under the Failure Modes and Effects Analysis, Appendix C.

Results of the transient analysis (see 8.5.7 and 8.7) indicate that there may be a need to optimize the pitch control system and generator regulator dynamic response to manage the transient line voltage and power variations which occur when the WGS is connected near the end of a distribution feeder. This should be further investigated during the detail design.

5.5.2.2 Weight and Cost Summary

A summary of the estimated weights of the preliminary design of the control system components at the WGS site is given below:

EQUIPMENT ON TOWER	WEIGHT - Kg (Lb)
Sensors (Including Met)	22.7 (50)
Transmitter/Receiver/ Antenna	2.7 (6)
TOTAL ON TOWER	25.4 (56)
EQUIPMENT ON GROUND	
Primary Control Electronics	9.1 (20)
SSC Equipment	31.7 (70)
TOTAL ON GROUND	40.8 (90)

There is no difference in the weights of the above equipments between the 500 KW and 1500 KW systems.

A summary of the costs of the electronic components of the control system is given below, for 1000 units:

	ESTIMATED COST PER UNIT (000 $)
Primary Controls and Sensors	$ 12.2
Supervisory and Sequencing Control (Remote Station Only)	17.7
Subsystem Integration	1.5
TOTAL	$ 31.4

The cost of any master station equipment outside the WGS is not included in the above. The above does include cost allowances for automatic fault monitoring equipment needed to assure safe operation of the WGS as described herein.

5.5.2.3 Reliability

A detailed reliability prediction for the control system equipment is best performed as a part of the final design. However, an estimate of the reliability of the critical portions of the system aids in design tradeoffs during the preliminary design. Provision was made for mean time between failure (MTBF) estimates to be entered on the failure modes and effects analysis sheets of Appendix C. These estimates were to be based on the judgement of each of the individuals filling out the sheets and were used as a further aid in identifying critical failure areas. Although it was not possible to estimate MTBFs for all the sheets, every effort was made to at least estimate MTBFs for those sheets which involved safety critical failure modes. The instructions given to the individuals who filled out the sheets are repeated at the beginning of Appendix C for reference. Although these FMEA sheets are primarily qualitative in nature, they constitute the primary analytical tool on which the control system preliminary design is based. Note that failures are grouped into classes based on the effect which the uncompensated failure would have on the system. This was necessary to keep the level of effort within bounds, consistent with the preliminary design phase.

5.6 Adaptability of Preliminary Design

The preliminary design presented herein for the electrical portion of the Controls Subsystem is applicable to any large horizontal axis Wind Generator in the range of a hundred KW up to several thousand KW. The suggested design is essentially universal in nature, except that for smaller size machines (less than 50 KW), it would be advisable to re-evaluate the design from a simplification cost standpoint. For large machines, where the investment in WGS equipment is large, reliability is a major concern and control system cost can be spread out over a larger base. For small machines, the cost base is reduced and protection of the reduced capital investment may not be as essential. This might allow some of the automatic fault monitoring and redundancy provisions to be reduced or eliminated to reduce Controls Subsystem cost. The WGS must be small enough so that the control system is a significant percentage of the total cost for this to be economically attractive.

5.7 Conclusions and Recommendations

5.7.1 Conclusions

The conclusions reached in this study are summarized below:

1. A microprocessor should be used for sequencing and supervisory functions

2. Critical functions should be continuously monitored and protected from failures

3. A pure mechanical feathering device should be used to eliminate need for a high capacity brake for overspeed and emergency shutdown

4. Near zero pitch angles have highest overspeed potential and greatest hazard

5. Emergency overspeed devices must be tripped at low values of overspeed to be effective

6. WGS design should not require control inputs when shut down in order to protect against high winds

7. Control rates must be limited (5 deg/sec pitch, 1/3 rpm yaw)

8. Startup is critical at low winds (near cut-in velocity)

9. Startup in high winds limited by pitch servo rate

10. RPM and torque control loops should include rate inputs to assure stability and tailor dynamic response.

5.7.2 Recommendations

Also, as a result of this study, it is recommended that the following items should be addressed during the detail design phase:

1. Develop a detailed analysis of rotor rpm variations due to wind gusts when separated from the network in the standby/synchronize mode.

2. Develop detailed failure modes and effects analysis for WGS, in particular for the SSC equipment. This will guide the preparation of the final functional flow diagrams and identify the specific sequences and/or functions of the SSC which require external monitoring.

3. Develop detailed test procedures for acceptance of SSC software to assure integrity of the software package.

4. Develop a detailed dynamic control model for final rotor and controls subsystems which will be used as an input to a detailed stability analysis of the WGS primary control loops. This model should also be combined with the model of the generator/network interface to develop an expanded transient analysis of the WGS generator connected to the utility network (see 8.5.7 and 8.7.2).

5.8 Microprocessor Failure Modes and Effects Analysis (FMEA)

Subsequent to the completion of the main body of work described in this report, an additional task, Microprocessor FMEA, was initiated. This was done partially in response to Recommendation 2 of 5.7.2, above.

This study has provided further evidence that a microprocessor is suitable for use in the sequencing and supervisory contral functions of a wind generator system. The microprocessor can detect and take the necessary corrective action in the event of failures in the remainder of the system that could lead to hazardous conditions. In most cases, the microprocessor can even detect its own failures, or at least their effects that might create hazardous conditions, and then initiate proper corrective action. Microprocessor failure detection techniques to implement this capability generally allow detection of the failure and initiation of corrective action prior to the WGS getting outside its normal operating limits. However, there are some particular failure modes in the microprocessor that may escape detection by these techniques. Therefore, an independent monitor on the possible effects of those particular failure modes should be provided as a backup to microprocessor detection of those failure modes. Critical parameters that should be monitored for this purpose are rotor speed and vibration level.

There is one area in which the microprocessor-based control system is quite different from a control system using conventional techniques, the interface with the operator or maintainer. A control system, using conventional techniques, generally has a number of indicator lamps, annunciators or other directly observable devices that indicate conditions within the control system at all times. The microprocessor-based system, on the other hand, carries out its logical decision making within the microprocessor circuits themselves, where it is invisible to the operator or maintainer, particularly one who is not familiar with computer-control techniques. Therefore, it is concluded that diagnostic aids should be part of the microprocessor-based sequencing and supervisory control system. This not only provides the maintainer with a degree of confidence that the trouble he finds and fixes is, in fact, the one which caused the system to shut down.

The results of this study are generally applicable to all WGS design concepts. However, in applying these results to specific designs, consideration should be given to their particular requirements. Each design will have its own peculiarities and, thus, may have potential hazards which must be identified by careful analysis, and guarded against.

6.0 STRUCTURE SUBSYSTEM

6.1 Description

The WGS Structure Subsystem consists of the tower, the turntable and the foundation. The candidate towers investigated in this study were the steel truss, steel shell, reinforced concrete shell, pre-stressed concrete shell and the guyed pole types. In addition, the use of multiple rotors on a single tower was investigated. The selected towers for the 500 KW and 1500 KW WGS preliminary designs are the steel truss and the pre-stressed concrete shell, and are discussed in paragraph 6.5. Three basic foundations were addressed in this study: mass concrete, friction piles and friction piles with rock anchors. Preliminary design results indicate that, because of the large variations in local soil conditions, determination of the most efficient foundation option must be made for each site. The turntable structure provides the structural link between the hub, main gearbox, generator and nacelle, and the tower top. The studies conducted to analyze and select these subsystem components from the many options examined are presented in this section.

6.2 General Requirements

The tower must support and orient the rotor in the selected wind regime and be capable of reacting the forces imposed by the rotor and by the wind acting on the tower itself for the design conditions. Fatigue strength of the tower must be great enough to withstand the rotor-induced vibratory loads, including the effects of startup, shutdown cycles, gust variations, tower shadow and gravity for a 50 year service life. The stiffness of the tower must be selected so that the resulting tower natural frequencies avoid integral multiples of the operating frequency.

The foundation must provide a firm anchor for the tower structure for all imposed loading conditions and natural (earthquake) imposed conditions.

The turntable structure and associated connecting structure, orientation drive mechanism structure, and protective shrouding must transmit loads developed by the rotor and power conversion machinery to the tower and protect these components from environmental threats (rain, dust, lightning, etc.).

Within these general requirements, many other detailed requirements, specific to each particular WGS concept, size, operating conditions and other factors, were developed and used to guide the tower concept selection and design. These are discussed below in succeeding paragraphs.

6.3 Design Approach

The design approach taken for evaluation of the various tower, foundation and turntable concepts was to minimize cost by utilizing existing commercial quality materials and standard construction techniques. No new materials and construction techniques need be developed for the WGS Structure Subsystem.

The American Institute of Steel Construction (AISC) Steel Handbook was used for choosing standard rolled structural shapes and plate and connections, and for determining the appropriate allowable static and fatigue loads and stresses. The American Concrete Institute (ACI) Handbook was used for determining the design requirements and allowable concrete and reinforcing steel working stress for both the reinforced concrete tower and concrete foundations. The Prestressed Concrete Institute (CPI) Handbook was used in proportioning the design and in determining the allowable stresses in concrete and post-tensioning steel.

The Uniform Building Code and the BOCA Code (Building Officials and Code Administrators International, Inc.) are the source of most local building codes and, as a result, were used in establishing criteria, particularly seismic load requirements.

The Federal Occupational Safety and Health Act (OSHA) applies to facilities which are available to the public or in which employees are required to work or maintain equipment. In the case of the WGS tower, the regulations governing the installation of such items as elevators, stairs, ladders, platforms, railing, ladder cages, etc., were followed for the design.

6.4 Candidate Concepts

A number of tower concepts and configurations were investigated to determine the feasibility and, if attractive, the cost of each tower. General illustrations of these basic types are shown in Figure 6-1. Initially, the steel truss, steel shell and guyed pole were investigated, resulting in deletion of the guyed pole as being impractical for the large Wind Generator System and too costly for the small WGS. A later study examined the use of reinforced concrete as an alternative material for the shell tower. This led to the selection of a pre-cast, post-tensioned concrete shell as being cost competitive with the truss tower on a multi-unit production basis. Multiple rotors per tower were also investigated but did not appear attractive under close analysis.

The initial parametric foundation cost analysis was based upon the use of a mass concrete foundation and resulted in a high cost foundation. As a result, alternate foundation designs were investigated, indicating that friction piles or friction piles with rock anchors would probably be more economical than the mass concrete design, especially for the larger WGS systems.

6.4.1 Alternative Designs for Tower and Foundation

6.4.1.1 Tower Concepts

Truss Tower - The steel truss tower is the most economical of any of the towers investigated. The use of standard steel sections, connections and erecting techniques ensures ready availability throughout the United States. Modifications for instrumentation platforms, strengthening or stiffening can easily be made after construction. Thus, the truss tower is ideal for demonstration installations. There is also some small cost advantage for high quantity production. Disadvantages are aesthetics and the exposed environment of the servicing ladders.

Pre-cast, Post-tensioned Concrete Tower - The pre-cast, post-tensioned concrete tower concept examined is constructed of twelve factory match-cast conical

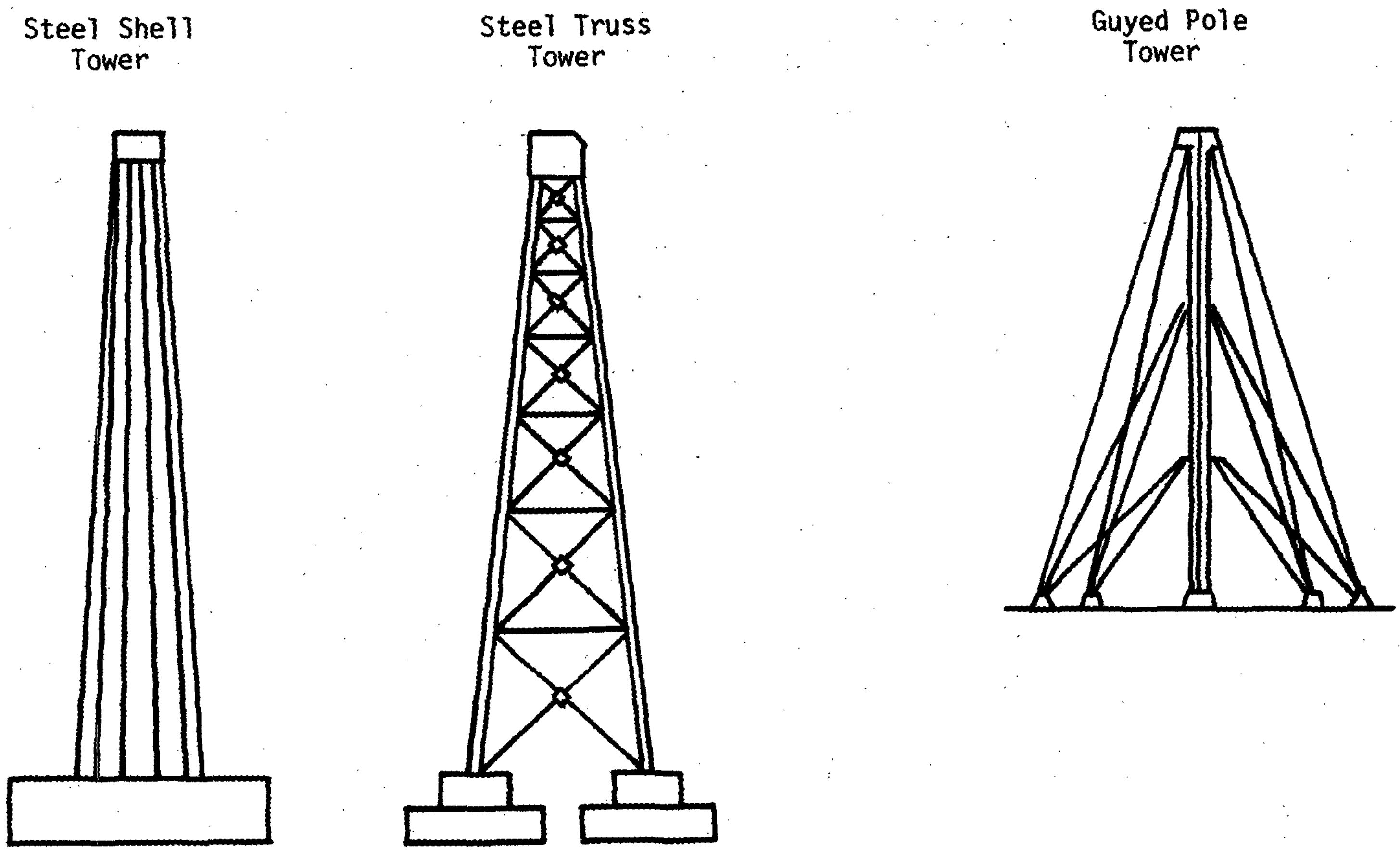


Figure 6-1. Candidate Tower Concepts.

segments. Each of the elements in the upper and lower halves are identical to other elements in the same half to economize on the cost of permanent forms. The detail parts, plus an erection tower, are shipped to the construction site where the erection tower and lower panels are erected. At this time, tensioning of some of the post-tensioning steel is accomplished. Upon erection of the remaining panels, the final post-tensioning is completed. The level of post-tensioning is such that, under the most severe loading condition, the concrete remains in compression.

Pre-cast, post-tensioned concrete permits the use of high strength, high modulus concrete, a minimum amount of reinforcing steel, elimination of concrete cracking due to tensile stresses or shrinkage, and a fully effective section in bending which enhances the bending and torsional stiffness and strength of the tower. Because of the small amount of post-tensioning steel, it does not add to bending strength or stiffness; its function is to keep the concrete in compression.

The pre-cast shell tower provides protection for the ladder and an enclosed storage area. In addition, architectural features can be cast into the segments with little cost increase to enhance the appearance. On a moderately high production basis, the cost approaches that of the steel tower. However, single or low quantity unit costs are very high, due to the cost of the section forms and erection tower. In addition, as with all shell towers, it does not have the adaptability of the truss type tower.

Reinforced Concrete Tower - The reinforced concrete tower is a truncated circular cone cast in place at the construction site using slip forms. Because of the poor tensile properties of concrete, heavy reinforcing steel, both longitudinal and spiral, is required to provide adequate strength and stiffness. As a result, the walls are thick and the tower is heavy and costly. In addition, cracks will develop from applied forces which produce tensile stresses and from shrinkage during curing.

As in the case of other shell towers, the ladder will be sheltered, since access to the nacelle area will be through the inside of the tower, and a sheltered storage area is available. Tower material and labor is readily available for this type of tower.

Steel Shell Tower - The steel shell tower is a fully monocoque, truncated circular cone fabricated from rolled conical segments which are field-welded in position at the site. It has the aesthetics and sheltered area advantages common to all shell towers, but is more costly than either the truss, pre-cast concrete or reinforced concrete towers. In addition, erecting specialists are required for construction.

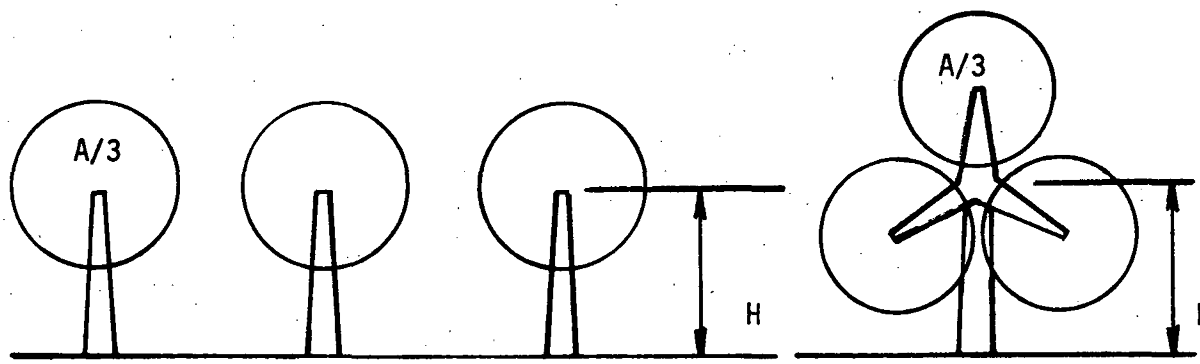
Guyed Pole - The guyed pole tower consists of a circular steel cylinder supported near the top (just below the turntable structure) by steel cables which provide bending stiffness and at intermediate levels to provide column stability. The upper cables are splayed out further from the vertical than the critical angle of truss tower chords. The intermediate cables are fastened to the same anchorage required for the upper cables.

The diameter and wall thickness of the circular cylinder provide the necessary torsional stiffness while bending stiffness is achieved by the redundant relationship between cylinder bending stiffness and the effectiveness of the cable supports. Because of the low efficiency of the cable support system, adequate tower bending stiffness is achieved by adding many heavy upper support cables. Since a large preload in each cable is necessary to keep it effective during operation, very large foundation loads result, particularly for the center support.

As a result, the cost of the guyed pole tower for the low power WGS (100 KW for the initial evaluation) was driven up by the high cost of the foundation. A practical guyed pole tower design was not feasible for the imposed requirements for the high power WGS.

6.4.1.2 Multiple Rotors per Tower

A study was made to evaluate the effect of adding multiple rotors to a single tower. For a basis of comparison, three rotors of equal area were mounted on a single tower and compared with three rotors of equal areas mounted on separate towers. The hub height of the single rotor was made equal to the average hub height of the multiple rotors.



Detailed results of the evaluation are presented in paragraph 6.4.4. A discussion of the major design factors influencing the differences between single and multiple rotor towers is given below to illustrate their effects on the size, weight and cost of the resulting tower designs.

Static Strength - A single tower with three rotors has three times the load of a tower with a single rotor; therefore, the chord (leg) area of the tower is three times higher, resulting in three times the chord weight. The weight of the multiple rotor chords are, therefore, equal to the weight of three single rotor chords.

Rotor hub moments are also three times greater on a single tower with three rotors if all three rotors are in phase. If synchronized 60° out-of-phase, the resultant vibratory hub moment felt by the tower is essentially zero (exactly zero if the hub moments on each rotor were identical). Unsymmetrical loading conditions must also be considered, however, and some weight will be added as a result of increased torsion in the tower.

Fatigue Strength - The synchronous generator prevents differences in operating rpm, except in the start/stop cycle. Vibratory loads on the rotors will, therefore, be essentially the same for both concepts, and local structure near rotor attachments will be the same. As stated above, rotor vibratory loads do cancel in the basic section of the tower, but since fatigue is not a critical consideration in the tower design, little advantage is realized due to this feature.

Bending Frequency - The requirements are equivalent for both tower concepts; three times the mass requires three times the stiffness. Operating frequencies are equal, total weight of rotors, hub and generator are also equal, but the support structure between each hub and turntable will be a large source of flexibility, unless it is very stiff and heavy. Therefore, the weight above the turntable for the multiple rotor concept will be heavier and more flexible requiring more steel in the tower and, hence, more weight and cost.

Torsional Frequency - Operating frequencies of the two concepts are equal, but exciting forces should be significantly reduced for the multiple rotor system, especially if synchronization at 60° is attained. If full synchronization is attained, exciting forces may be low enough to eliminate the requirement that natural frequency in torsion be 2-1/2 times operating frequency. However, flexibility of hub to turntable structure will reduce torsional stiffness, and additional weight may have to be added to the tower to compensate for the loss of stiffness. More significant is the huge increase in mass moment of inertia (polar) of the multiple rotor tower and the resulting reduction in natural frequency, unless large amounts of steel are added to raise the natural frequency by increasing the torsional stiffness.

Synchronization - The canceling effect of multiple rotor hub moments (tower torque and tower overturn) should affect the weight of the tower below the juncture of the three rotors below the turntable (above the turntable the structure weight should be the same). However, electrical synchronization is required before bringing generators on line. This can be done either mechanically with additional interconnecting shafting and gearboxes, or electrically, with additional control and electrical system complication. For either approach, higher cost and reduced reliability will be incurred for the multiple rotor tower system, offsetting any synchronization benefits.

The results given in paragraph 6.4.4 do not account for all of the weight and cost penalties discussed above for the multiple rotor tower case examined. However, on structural grounds alone, the concept appears poor, and these additional factors make the approach even less feasible.

6.4.1.3 Foundations

Mass Concrete - This type of foundation distributes the applied concentrated shear and compression loads, including the weight of the concrete foundation, to the underlying soil, within its bearing capacity. Uplift forces are reacted by the weight of the concrete and overlying soil. This foundation requires no special equipment or construction techniques, but may require large quantities of concrete which will affect cost.

Friction Pile Foundation - Friction pile foundations react compression loads primarily by surface area friction between the pile and the soil, which can vary between 2.4 - 38.3 kPa, (50 - 800 pounds per square foot), depending upon the particular soil conditions. Uplift loads are reacted by both surface friction and the weight of the pile. 15% to 20% of the compression allowable is generally accepted as the tension allowable, with 25% as the upper limit.

The disadvantage of this foundation type is the low tension allowables of each pile, leading to a large number required and the attendant cost. However, this foundation is often the most economical type for certain soil conditions.

Friction Piles With Rock Anchor - This foundation uses the friction piles to react the compression load while a rock anchor reacts the tension or uplift load. To construct a typical anchor in rock, a hole of suitable depth and diameter is drilled, using rotary drilling equipment, through the soil strata into the rock where the boring hole is underreamed. After drilling, the hole is cleaned and an anchor tendon, consisting of high strength strand or bar, is inserted. Cement grout is then pumped into the hole by means of a tube placed at the bottom of the hole. After a curing period, all anchors are tested to the required load. Rock anchors have been installed to capacities of 6.67 meganewtons (750 tons).

Rock anchors are quite economical, even at significant depths. Therefore, this foundation type appears most attractive where suitable bedrock is available. Soil anchors can also be used when bedrock is not available and have been installed to capacities of 1.33 meganewtons (150 tons).

6.4.2 Structural Criteria

The bases for selection of the recommended tower designs were (a) the ability to meet the established structural requirements, (b) economy and (c) aesthetics. The foundation selection involved only (a) and (b).

The structural criteria established included static strength, fatigue strength and stiffness requirement for the tower, and static strength for the foundation.

The static strength requirement consisted of three basic conditions: (1) 53.6 m/sec (120 mph) maximum wind blowover condition; (2) normal operating loads plus seismic force; and (3) maximum operating loads. These are illustrated in Figure 6-2.

For the 53.6 m/sec (120 mph) blowover condition, blades were assumed parked vertically in the feathered position. Yaw control failure of the turntable assembly was assumed allowing the wind to impinge flatwise on the blades and broadside on the nacelle, producing maximum tower torsion in addition to maximum tower shear. Wind velocities were calculated at the center of areas of each rotor blade, the nacelle and the tower, based upon the wind shear relationship given in Section 3.

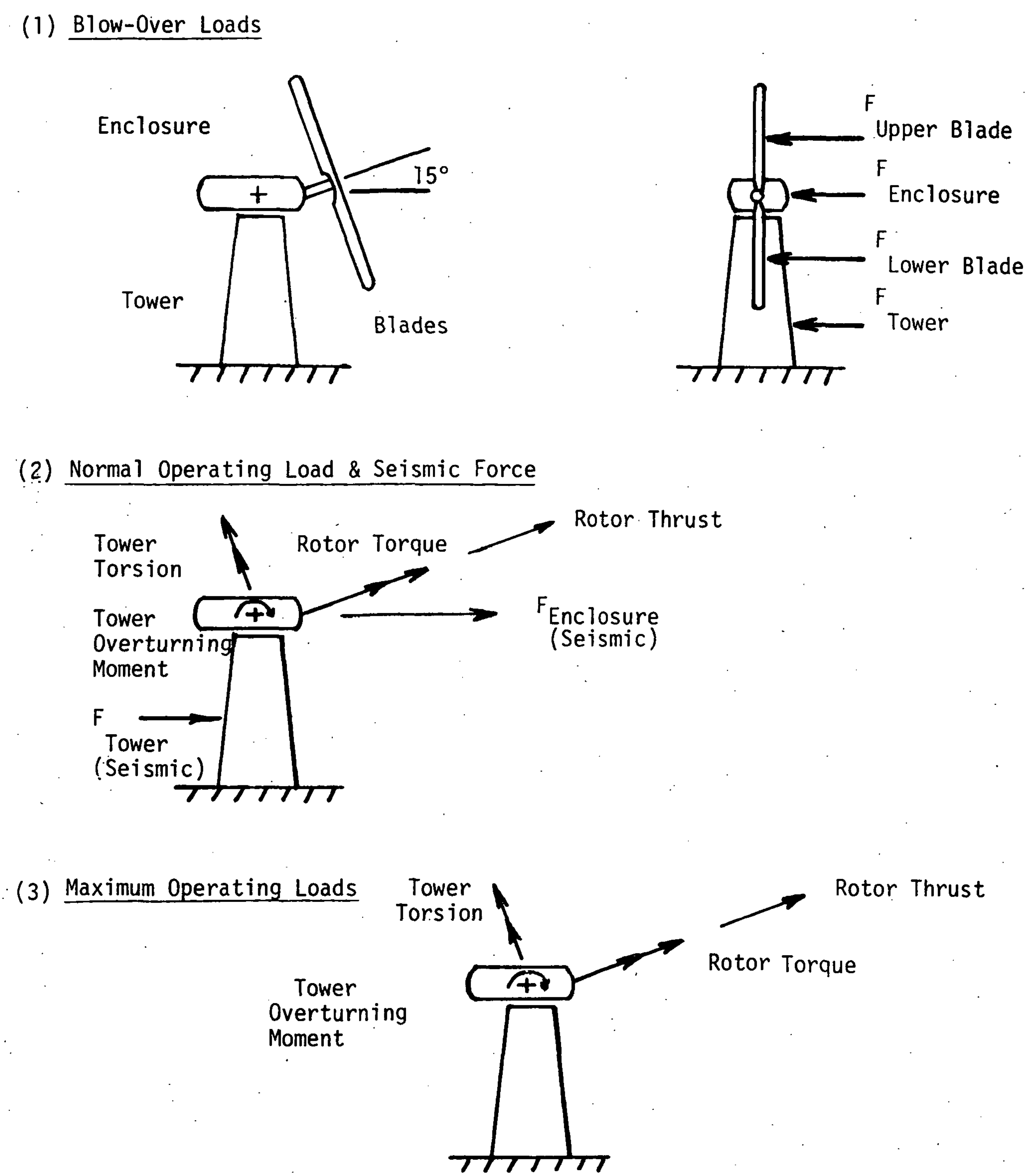


Figure 6-2. Static Tower Loads

Loads were calculated using:

$$F = 1/2 \rho C_D A V^2$$

where:

F is the force acting on a body

ρ is the mass density of air

C_D is the drag coefficient of the body

A is the projected area

V is the velocity of the wind

The drag coefficient (C_D) for the various bodies used in the analysis were:

ITEM	C_D
Blade (Normal to wind)	1.40
Nacelle	0.80
Truss Tower	2.0 on net area of truss
Shell Tower	0.6 on gross area of tower

The normal operating loads plus seismic load condition contains the normal operating loads at rated wind producing maximum thrust, corresponding steady state hub moments and torque, and a seismic load parallel to the direction of rotor thrust. In determining the magnitude of the seismic force, the more conservative lateral load factor to be applied to the total weight of the structure (0.25) was chosen from either the Uniform Building Code (UBC) or the BOCA code for Zone 3 (Zone 3 covers all of the contiguous United States, except for local areas in California, Nevada and Arizona). This condition was not critical in the case of the steel truss, steel shell and guyed pole designs. However, because of the large weight of the concrete towers, a less conservative coefficient was determined to prevent overdesigning for the seismic force. By calculating the natural period of vibration of the structure, a new lateral load factor of 0.16 was determined from the UBC code. (The BOCA code produces 0.12.) The reduction in lateral load factor is shown in paragraph 6.5 for the shell tower.

The maximum operating condition simulates the worst transient loadings, including dynamic response of the tower on the WGS due to gusts, overspeeds and control failures. Factors were applied to the primary load components as follows:

LOAD COMPONENT	FACTOR
Thrust	2.0
Hub Moment (Overturn)	2.0
Hub Moment (Tower Torque)	2.0
Rotor Torque	2.5

This condition was not critical.

Fatigue strength conditions investigated were (1) rotor induced vibratory loads for high cycle loading and (2) startup and shutdown cycle loads for low cycle loadings. Effects of transient loadings from gusts and control failures were not investigated, but based upon the results of the analysis of conditions (1) and (2), it is expected that the transient conditions will also produce low stress levels.

The first condition consists primarily of two components of vibratory hub moments assumed to act in excess of 30 million cycles. Vibratory stress levels in the chords (legs) and diagonal bracing for both the 500 KW and the 1500 KW preliminary design truss towers were less than 10% of the allowable vibratory stress levels presented in the AISC Manual.

The second condition was composed of thrust, overturn moment and rotor torque. The thrust and overturn moment were superimposed and consisted of thrust and moment variations for 0 to 1.5 times maximum steady state loads occurring at rated wind speed, 5 times a day for 50 years. The rotor torque was investigated separately since resulting chord stresses are not additive with the stresses produced by the other load components when the turntable is oriented in the critical position (parallel to the diagonal). The criteria of 1.5 times maximum load and the frequency of occurrence were the same. Resulting maximum chord stresses for the 500 KW and 1500 KW preliminary design truss towers were 26% and 29%, respectively, of the allowable vibratory stress levels presented in the AISC manual.

Vibratory stress levels in the shell designs were correspondingly low. As a result, fatigue requirements are not expected to influence the weight of the tower designs. Of course, attention must be paid, in detail design, to efficient load paths, lack of eccentricities and to the type of joints and connections, particularly in the areas of high concentrated loads.

Bending and torsional stiffness requirements were established for concept selection such that the resulting bending and torsional frequencies were at least 1.5 and 2.5 times, respectively, the operating rotor frequency. In initial analyses, bending stiffness calculations were based upon simple, two-stepped, constant section cantilevers for the shell tower and a simple triangular truss with only the chords contributing to strain energy for the truss tower. A similar simplified structural model neglecting chord contributions was assumed for the torsional stiffness calculations. A flexibility correction factor of two was assumed to account for the effects of the local distortions in the upper portions of the tower. A more rigorous method of determining natural frequencies was used for both the concrete shell and steel truss in the preliminary design stage, as detailed later.

For evaluation of the foundation design, the three static loading conditions were considered. Cost analyses of the three foundation types were performed assuming certain average bearing and friction capacities of the soil and lengths of piles. Costs were calculated and compared for the three types of foundations for both the truss tower and shell tower. Final selection of foundation type, however, must be made for each site because of the large variations in local soil conditions.

6.4.3 Engineering and Cost Data

Besides the basic design criteria described in paragraph 6.4.2, engineering data and unit costs were determined in order to evaluate the various tower and foundation concepts. Some selection studies were made at times when the WGS optimum size was not established and when only preliminary unit costs were available. However, some costs and parameters remained unchanged for all studies. The sources of cost and supplemental engineering data were Mueller Engineering Corporation, Blakeslee Pre-stressed Concrete Corporation, direct contact with vendors and "Means Cost Data", 1975 edition.

6.4.3.1 Engineering Data

Strength criteria for steel from AISC Steel Handbook, for reinforced concrete from ACI Code and for pre-cast, post-tensioned concrete from PCI Design Handbook were used for all analyses:

a. Truss steel - A36 structural grade, F_{ty} = 248 MPa (36 ksi)

b. Steel shell - A36 structural grade, F_{ty} = 248 MPa (36 ksi)

c. Reinforced concrete:

Concrete F'_c = 20.7 MPa (3.0 ksi), E = 20.7 GPa (3×10^3 ksi)

Reinforcing steel, F_{ty} = 276 MPa (40 ksi)

d. Pre-cast, Post-tensioned concrete:

Concrete F'_c = 34.5 MPa (5.0 ksi), E = 30.3 GPa (4.4×10^3 ksi)

Tensioning steel, $F_{t\mu}$ = 1.86 GPa (270 ksi)

e. Assumed average condition for foundation analysis:

Soil bearing pressure = 192 kPa (4000 psf)

Pile friction pressure = 19.2 kPa (400 psf)

Pile length = 12.19 m (40 feet)

Rock anchor depth	= 30.48 m (100 feet)
Pile tension allowable	= 20% compression allowable
Rock anchor	= to 6.67 MN (750 ton)
Soil anchor	= to 1.33 MN (150 ton)

6.4.3.2 Cost Data

Cost data from the sources cited above were as follows:

a. Truss tower structural steel, including material, fabrication, shop drawings, erection, transportation to non-remote site and prime coat:

PRODUCTION UNIT	COST
1	\$ 1212/Mg (\$ 1100/ton)
100	1102 (1000)
1000	1047 (950)
10000	937 (850)

b. Shell tower steel, including structural steel plate, rolled and welded, fabrication, labor, transportation to non-remote site and prime coat:

PRODUCTION UNIT	COST
1	\$ 2590/Mg (\$ 2350/ton)
100	2370 (2150)
1000	2232 (2025)
10000	2150 (1950)

c. Guyed pole tower, includes structural steel plate (rolled and welded), fabrication, labor, transportation and erection; \$ 1874/Mg (\$ 1700/ton).

Guy cables:

.041 m (1-5/8") d. = \$10.50/meter (\$3.20/ft) - 4 instl. in 2 days

.019 m (3/4") d. = \$ 2.92/meter (\$ .89/ft) - 8 instl. in 2 days

Guying labor: 2 days, 4 steelmen, foreman = \$ 1300.

d. Reinforced concrete tower, constructed, (for specific quoted design), 26.2 m (86 ft) high, 0.2 m (8") wall thickness, with 9 Mg (10 ton) spiral and longitudinal steel; $50,000.

Incremental costs:

Steel, in place	\$ 882/Mg (\$ 800/ton)
Concrete, in place	\$ 190/m^3 (\$ 145/yard^3)

e. Post stressed concrete shell tower, erected within 160 Km (100 mi) of casting facility (quoted for specific preliminary designs), including forms:

	PRICE			
UNITS	1	100	1000	10000
1500 KW	\$ 324,000	\$ 63,000	\$ 60,600	\$ 60,300
500 KW	303,000	52,000	49,800	49,600

f. Foundation costs, by increment:

Reinforced concrete, in place \$190/m^3 (\$ 145/yard^3) includes concrete, reinforcing steel, forming and placing.

Excavation	\$1.95/m^3 (\$1.50/yd^3)
Compacted backfill	7.79/m^3 (5.95/yd^3)
Soil testing	3000 - 4000
Mobilization, demobilization	400
Spreading, leveling at dump site	2.19/m^3 (1.63/yd^3)
.61 m (24") square pile	59.06/meter (18/ft)
.36 m (14") square pile	45.93/meter (14/ft)
30.48 m (100 ft) rock anchor	2000

6.4.4 Concept Evaluations

The results of the evaluations for the various conceptual designs are presented on the following pages. Weight and cost comparisons are included, in addition to observations, conclusions and recommendations.

Study 1 - Steel Truss, Steel Shell and Guyed Pole Towers for 100 KW and 1000 KW WGS

This study was the conceptual design study where the steel truss, steel shell and guyed pole were the candidate towers for both a 100 KW (low power) and a 1000 KW (high power) WGS. The design parameters for this study are given in Table 6-1. The results are given in Table 6-2(a), (b), (c) and (d). Table 6-2(a) gives a detailed weight breakdown of the candidate towers. Table 6-2(b) gives a cost breakdown. Tables 6-2(c) and 6-2 (d) show more detailed cost breakdowns.

As a result of the analysis, the truss tower was chosen as the recommended design for optimization over the steel shell tower because of its lower cost. Both designs met all design requirements. The guyed pole was eliminated for the 100 KW design because of the high cost of its foundations for the pole and eight anchorages. High foundation loads resulted from the sum of high cable preload and liveload. A guyed pole design for the 1000 KW WGS was not practical within the imposed limits and selected criteria. Too many large diameter cables were required at a very high splaying angle to produce an acceptable bending natural frequency.

Results of the analysis also indicated that the towers for the 1000 KW WGS were designed by stiffness, were equally influenced by the three static conditions and that fatigue was not a critical condition. The 100 KW system was designed by one strength condition; 53.6 m/sec (120 mph) blowover. Fatigue was not critical for the 100 KW designs, either. It was found that the cost of the mass concrete foundation was approximately equal for the truss and steel shell tower. (Pile foundations were not yet under consideration at this point.)

Table 6-2(d) illustrates the relatively high cost of accessory equipment, such as elevators, hoists and/or jib cranes as part of the tower, per se. These results heavily influenced the choice of a simple tower design, with minimal auxiliary equipment to minimize overall cost. This, in turn, resulted in the installation and erection approach which used general erection equipment brought to the site for erecting the WGS, as well as for major overhaul and maintenance.

Study 2 - Multiple Rotors per Tower - 1000 KW Truss

The use of multiple rotors on a single tower was also considered during the system optimization phase. For a basis of comparison, three rotors of equal area were mounted on a single tower and compared with three rotors of equal areas mounted on separate towers. Details of the approach and major factors were discussed in 6.4.1.2. The design parameters used in the study are given in Table 6-3. The weight and cost results of the comparison are shown in Table 6-4.

The results of the analysis show that the multiple rotor tower has a lower foundation cost, lower site cost and lower vibratory loads in the common portion of the tower structure below the turntable support. However, the accompanying disadvantages are higher total costs, heavier installation, very high mass moment of inertia and lower reliability because of more component parts (total) and higher dynamic loading. Since these results were so dramatic, single rotor towers were recommended over multiple rotor towers for all power levels and tower types.

TABLE 6-1. DESIGN PARAMETERS FOR STUDY 1.

Steel Truss, Steel Shell and Guyed Pole

For 100 Kw and 1000 Kw WGS

Design Parameters

System Rated Power, Kw	100	1000
Rotor Diameter, m (ft)	29.26 (96)	43.28 (142)
Blade Maximum Chord, m (ft)	.88 (2.9)	1.37 (4.5)
Rotor Operating Speed, RPM	40	47
Rated Wind, m/sec (MPH)	8.0 (18)	12.5 (28)
Thrust at Rated Wind, kN(kips)	20.0 (4.5)	118.3 (26.6)
Hub Height, m (ft)	19.20 (63)	26.21 (86)
Rotor Ground Clearance, m (ft)	4.57 (15)	4.57 (15)

TABLE 6-2(a). RESULTS OF STUDY 1.

Structural Weight Summary, Mg (kips)

Item	Truss		Shell		Guyed Pole	
	100 Kw	1000 Kw	100 Kw	1000 Kw	100 Kw	1000 Kw
Chords	3.4 (7.4)	12.7 (27.9)	-	-	-	-
Diagonal Bracing	.8 (1.8)	8.3 (18.3)	-	-	-	-
Horizontal Bracing	.4 (.8)	3.8 (8.3)	-	-	-	-
Top Structure	.5 (1.0)	.9 (1.9)	.5 (1.0)	.9 (1.9)	.7 (1.5)	-
Base Structure	.5 (1.0)	.9 (1.9)	.5 (1.2)	.9 (2.0)	.5 (1.0)	-
Gussets	.5 (1.0)	.9 (1.9)	-	- -	-	-
Misc. Non-structure	.5 (1.0)	.9 (1.9)	.5 (1.0)	.5 (1.1)	.5 (1.0)	-
Shell (4x13x63x0.25)	-	-	8.2 (18.0)	-	-	-
Shell (2x63x0.5)	-	-	-	-	3.7 (8.1)	-
Shell (6x14x86x0.5)	-	-		25.9 (57.0)	-	-
TOTAL	6.4 (14.0)	28.2 (62.1)	9.57(21.1)	28.1 (62.0)	5.3 (11.6)	-

TABLE 6-2(b) RESULTS OF STUDY 1.

Summary of Total Costs, $

	Truss		Shell		Guyed Pole	
	100 Kw	1000 Kw	100 Kw	1000 Kw	100 Kw	1000 Kw
Tower Structural Cost, $	7700	34200	23900	69800	17300	-
Foundation Cost, $	17300	31200	16600	35600	33900	-
Basic Cost, $	25000	65400	40500	105400	51200	-
Auxiliary Provisions,Cost $	13799	23100	13700	23100	14800	-
Total Tower Costs, $	38700	88500	54200	128500	66000	--

TABLE 6-2(c). RESULTS OF STUDY 1.

Basic Tower Cost Summary, $

ITEM	Truss		Shell		Guyed Pole	
	100	1000	100	1000	100	1000
Structural Steel	7700	34200	23900	69800	9900	--
Guy Cables - Material	--	--	--	--	2200	--
Labor	--	--	--	--	5200	--
Equipment						
Ladders	2800	3900	2800	3900	5000	--
Grating	1700	2300	1700	2300	600	
Rails	1900	2600	1900	2600	1900	--
Enclosure	3500	9800	3500	9800	3500	--
Lighting	3800	4500	3800	4500	3800	--
TOWER	21400	57300	37600	92900	32100	
Foundation						
Concrete	11600	24500	12200	30400	24700	
Excavation	600	800	300	400	1500	
Spreading Fill at Dump Site	300	400	200	400	800	
Mobilization & Demobilization	400	400	400	400	400	
Compacted Backfill	1400	1600	500	500	3500	
Soils Testing	3000	3500	3000	3500	3000	
FOUNDATION	17300	31200	16600	35600	33900	
TOTAL	38700	88500	54200	128500	66000	

TABLE 6-2(d). RESULTS OF STUDY 1.

Accessory Equipment Cost, $

Elevator	1500 pound capacity		$ 60000
Hoist	13.6 Mg (15 Ton)		25000
	4.5 Mg (5 Ton)		11500
Jib Crane	(6.10 m (20 foot) reach)		50000
Stairs vs Ladders	Δ Cost	100 Kw	3600
		1000 Kw	4700

TABLE 6-3. DESIGN PARAMETERS FOR STUDY 2

Multiple Rotors per Tower - 1000 KW Truss Tower

PARAMETER	3 ROTORS 3 TOWERS	3 ROTORS 1 TOWER
System Rated Power, KW	3 x 333 = 1000	1000
Rotor Diameter, m (ft)	23.93 (78.5)	23.93 (78.5)
Maximum Chord, m (ft)	.75 (2.46)	.75 (2.45)
Operating Speed, rpm	85.4	85.4
Rated Wind, m/sec (mph)	12.5 (28.0)	12.5 (28.0)
Thrust at Rated Wind, kN (kips)	3 x 40.9 (3 x 9.2)	3 x 40.9 (3 x 9.2)
Hub Height, m (ft)	26.88 (88.2)	26.88 (88.2)
Rotor Clearance, m (ft)	14.94 (49)	5.79 (19)

TABLE 6-4(a). RESULTS OF STUDY 2

Weight Comparison, Mg (kips)

	3 Small Rotors 3 Towers	3 Small Rotors 1 Tower
Rotor	7.6 (16.8)	7.6 (16.8)
Drive System	9.9 (21.9)	9.9 (21.9)
Elec. System on Tower	6.0 (13.3)	6.0 (13.3)
Controls	.7 (1.6)	.7 (1.6)
Pintle Assembly	19.6 (43.2)	46.0 (101.4)
Tower	75.8 (167.0)	298.9 (659.0)
WGS above foundation	119.7 (263.9)	369.2 (814.0)

TABLE 6-4(b). RESULTS OF STUDY 2

Cost Comparisons, $

	3 Small Rotors 3 Towers	3 Small Rotors 1 Tower
Rotor	91,700	91,700
Drive System	81,200	81,200
Elec. System	140,200	97,000
Pintle Assembly	57,400	120,200
Controls	34,100	34,100
Tower (includes install.)	146,700	431,500
Install. (except tower)	53,600	48,200
Site	28,100	21,800
Total direct cost	$ 633,000	$ 925,700

Study 3 - Truss Tower vs Shell Tower for 600 KW, 1000 KW, 3000 KW Systems

A third study also conducted in parallel with the system optimization, was a more detailed analysis of the truss and steel shell tower designs. Three system power levels of 600 KW, 1000 KW ans 3000 KW were investigated for the design parameters and criteria previously described to determine the effect of design power level on the relative merits of the two concepts. The study was based on the design parameters given in Table 6-5. Weight and cost comparison results are given in Table 6-6 and Figure 6-3.

TABLE 6-5. DESIGN PARAMETERS FOR STUDY 3

Truss Tower vs Shell Tower - 600 KW, 1000 KW, 3000 KW

System Rated Power, KW	600	1000	3000
Rotor Diameter, m (ft)	45.72 (150)	42.67 (140)	51.21 (168)
Maximum Chord, m (ft)	-	-	-
Operating Speed, rpm	33.3	46.3	49.2
Rated Wind, m/sec (mph)	8.9 (20)	13.0 (29)	16.1 (36)
Thrust at Rated Wind, kN (kips)	80.1 (18.0)	123.2 (27.7)	270.5 (60.8)
Hub Height, m (ft)	27.43 (90)	26.0 (85.3)	30.21 (99.1)
Rotor Clearance, m (ft)	4.57 (15)	4.57 (15)	4.57 (15)

These results show that the steel shell tower weighs less for sizes above 1000 KW, costs more for all sizes, but that the cost penalty is low for the 3000 KW WGS. The truss is cheaper for all sizes and weighs less for WGS ratings below 1000 KW. At this point, it appeared that for high power systems over 1000 KW, the steel shell tower aesthetic value might be worth the cost difference.

Study 4 - Foundations: Mass Concrete, Piles, Piles With Rock Anchor for 500 KW and 1500 KW Shell and Truss Towers

In the initial conceptual studies, consideration was only given to the mass concrete type foundation. Information became available indicating that a pile type foundation might be more economical under some conditions. As a result, three types of foundations were examined for the 500 KW and 1500 KW system preliminary designs; the mass concrete foundation, the pile foundation and the rock anchor pile foundation. These were described in paragraph 6.4.1.

The design parameters for Study 4 are summarized in Table 6-7. Cost results from the study are summarized in Table 6-8.

The conclusions drawn from these results are influenced by the assumed (standard) soil conditions used in the analysis. As shown, for the 1500 KW size, pile type foundations are more economical for both truss and shell towers. However, for the 500 KW systems, the mass concrete foundation is competitive with

TABLE 6-6(a). STUDY 3 RESULTS

Weight Comparison, Mg (Kips)

Shell Tower Vs Truss Tower

Weight	600 Kw		1000 Kw		3000 Kw	
	Truss	Shell	Truss	Shell	Truss	Shell
Top Structure	.9 (2.0)	.9 (2.0)	.9 (2.0)	.9 (2.0)	2.1 (4.6)	2.1 (4.6)
Chords	6.6 (14.6)	0	12.4 (27.4)	0	43.4 (95.7)	0
Diag. Brac.	2.8 (6.1)	0	6.7 (14.7)	0	37.8 (83.3)	0
Horiz. Brac.	1.3 (2.8)	0	3.0 (6.7)	0	17.1 (37.8)	0
Gussets	.9 (2.0)	0	.9 (2.0)	0	2.1 (4.6)	0
Misc. Struc.	.9 (2.0)	0	.9 (2.0)	0	2.1 (4.6)	0
Base	.9 (2.0)	.9 (2.0)	.9 (2.0)	.9 (2.0)	2.1 (4.6)	2.1 (4.6)
Shell	0	24.0 (53.0)	0	25.4 (56.0)	0	51.7 (113.8)
Struc. Steel (Total)	14.2 (31.3)	25.9 (57.0)	25.7 (56.7)	27.2 (60.0)	106.7 (235.2)	55.8 (123.0)

TABLE 6-6(b). STUDY 3 RESULTS

Cost Comparison, $\$ \times 10^{-3}$

Shell Tower Vs Truss Tower

Cost	600 Kw Truss/Shell		1000/Kw Truss/Shell		3000 Kw Truss/Shell	
Rotor	189		158		267	
Drive System	75		99		225	
Elect. System	53		68		121	
Pintle Assembly	56		62		121	
Controls	11		11		11	
Install. (Excl Tower)	19		21		28	
Site	22		20		27	
System (Minus Tower)	425		439		800	
Tower						
Ladders, Grating	10	10	9	9	11	11
Structural Steel	18	67	32	70	134	146
Foundations	34	34	32	32	43	43
Total Tower Cost	62	111	73	111	188	199
Total System Cost	487	536	512	550	988	999
Tower % of System	12.7%	20.7%	14.3%	20.2%	19.0%	19.9%
Penalty for Shell Tower	10.1%		7.4%		1.1%	

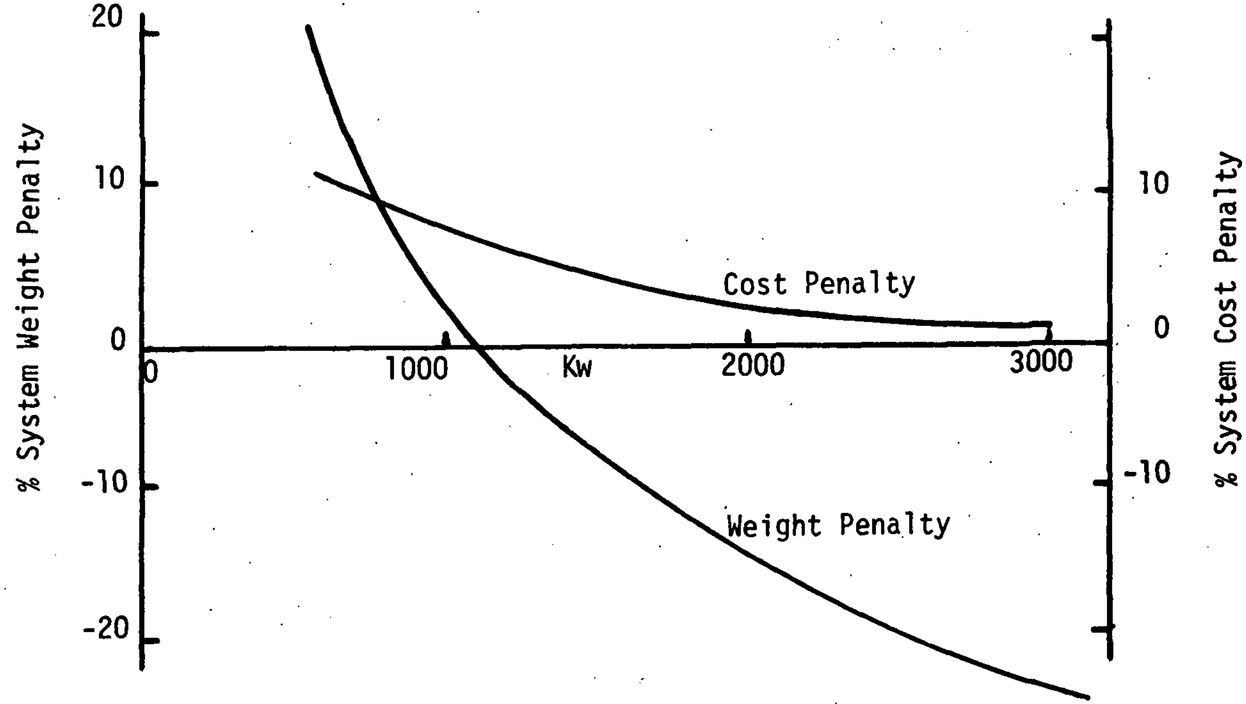


Figure 6-3. Shell Tower Vs Truss Tower Weight and Cost Penalties.

TABLE 6-7. DESIGN PARAMETERS FOR STUDY 4

Foundations - Mass Concrete, Piles, Piles with Rock Anchors for 500 Kw and 1500 Kw Shell and Truss Towers

Parameter	500 Kw	1500 Kw
System Rated Power, Kw	500	1500
Rotor Diameter, m (ft)	45.72 (150)	54.86 (180)
Maximum Chord, m (ft)	1.55 (5.08)	1.87 (6.12)
Operating Speed, RPM	32.3	34.4
Rated Wind, m/sec (MPH)	8.9 (20)	11.2 (25)
Thrust at Rated Wind, kN (kips)	60.9 (13.7)	143.2 (32.2)
Hub Height, m (ft)	37.49 (123)	42.06 (138)
Rotor Clearance, m (ft)	15.24 (50)	15.24 (50)

TABLE 6-8. STUDY 4 RESULTS

Foundation Cost Comparison for 500 and 1500 Kw Systems

	Truss		Shell	
	500 Kw	1500 Kw	500 Kw	1500 Kw
Piles with Rock Anchor	$ 23800	$ 26000	$ 38900	$ 48000
Piles in Soil	24500	35000	33300	48000
Mass Concrete	28900	47000	23500	52000

piles for the truss tower, and less expensive for the shell tower. Because of the lack of consistent conclusions of foundation type for the 500 KW and 1500 KW WGS towers, it was evident that local soil conditions govern the type and cost of each foundation, and that each type should be examined for the particular site under consideration.

The conclusions for the particular soil conditions assumed result in the following:

<u>1500 KW</u> Piles with rock anchors are most economical for the truss tower

Piles and piles with rock anchors are slightly more economical than the mass concrete foundation for the shell tower

Truss tower foundation is more economical than shell tower foundation.

<u>500 KW</u> All three foundations competitive for truss tower

Mass concrete is more economical than piles for shell tower

Truss and shell foundations have equal costs.

<u>Study 5</u> - <u>1500 KW WGS Truss, Steel Shell, Reinforced Concrete, Pre-cast, Post-tensioned Concrete Towers</u>

The initial purpose of this study was to compare the cost of a reinforced concrete shell tower with a steel shell tower for several production levels. In the course of the study, it became apparent that pre-cast, post-tensioned concrete towers could also be competitive, primarily in large production quantities. Thus, the study was broadened to include the steel truss (reference concept), steel shell, reinforced concrete shell and pre-cast, post-tensioned concrete shell towers as candidates for the final preliminary design 500 KW and 1500 KW systems. The comparison was conducted for the 1500 KW system requirements, although it appears that the results would also apply for the 500 KW systems.

Table 6-9 compares the advantages and disadvantages of each tower type. The design parameters used for the study are given in Table 6-10. Table 6-11 gives the summary cost data results of the study. These results show that the pre-cast, post-tensioned concrete tower is competitive with the truss tower in high production quantities, but that the steel shell and reinforced concrete shell are not. Also, it was apparent that the slight cost penalty for the pre-cast concrete shell tower would not outweigh its aesthetic value. Therefore, both the truss and pre-cast, post-tensioned tower concepts were carried through the preliminary design phase.

For a limited number of units, the truss tower remains the most economical choice. In addition, for experimental development systems, the truss tower offers an easily modified structure, whether the modification is for structural

TABLE 6-9. TOWER CONCEPT ADVANTAGES AND DISADVANTAGES

Type	Advantages	Disadvantages
Steel Truss	Most Economical	Aesthetics
Steel Shell	Aesthetics	Expensive, requires erecting specialists
Reinforced concrete shell	Aesthetics Readily available materials and labor.	Expensive. Heavy. No multiple unit cost reduction. Concrete in tension ineffective. Large quantity of reinforcing steel. Cracks develop in concrete.
Pre-cast, post-tensioned concrete shell	Aesthetics, Economical in production quantities. Concrete fully effective. Higher concrete allowable stress. Higher modulus of elasticity. Lighter concrete design. Rapid erection.	High single unit cost. Requires casting facilities.

TABLE 6-10. DESIGN PARAMETERS FOR STUDY 5.

Truss, Steel Shell, Reinforced Concrete, Precast, Post-tensioned Concrete Towers

Design Parameters	
System Rated Power, Kw	1500
Rotor Diameter, m (ft)	54.86 (180)
Maximum Chord, m (ft)	1.87 (6.12)
Operating Speed, RPM	34.4
Rated Wind, m/sec (MPH)	11.2 (25)
Thrust at Rated Wind, kN (kips)	143.2 (32.2)
Hub Height, m (ft)	42.06 (138)
Rotor Clearance, m (ft)	15.24 (50)

TABLE 6-11. STUDY 5 RESULTS

1500 KW Tower Cost Analysis - Dollars

	Unit Production			
	1	100	1000	10000
Truss	63000	63000	55000	49000
Steel Shell	176000	161000	150000	146000
Reinforced Concrete Shell	100000	100000	100000	100000
Precast-Post Tensioned Concrete shell	324000	63000	60600	60300

reasons or for other (utility) reasons. Therefore, the truss tower was recommended for the initial WGS experimental units. However, as discussed in Section 9, wide scale deployment of WGS units would favor the use of the pre-cast concrete shell type tower for public acceptance reasons.

6.5 Selected Concept Preliminary Design and Analysis

Based on the results of the tower concept selection studies summarized in the previous paragraphs, and the overall system optimization analysis, the steel truss and the pre-cast, post-tensioned concrete shell were selected for the 500 KW and 1500 KW WGS preliminary designs.

The three candidate foundations, i.e., mass concrete, piles and piles with rock anchors, are all possible selections for both the 500 KW and 1500 KW WGS. Typical design configurations were developed based upon the assumed soil conditions described in paragraph 6.4.3.

A more detailed description of each of the towers and foundation designs is given below, with a description of the common turntable and nacelle design.

6.5.1 Design Descriptions

1500 KW Steel Truss Tower - The 1500 KW truss tower is a four-sided tower constructed of standard grade structural steel (F_{ty} = 248 mPa [36 ksi]) H-beams and double angles. The width of the top of the tower was dictated by the size bearing required to react the rotor loads and is 2.59 meters (8.5 feet). The tower height is 36.88 meters (121 feet) from the foundation and the base width is 10.67 meters (35 feet). Figure 6-4 presents the tower design details.

Bending strength is primarily provided by the chords, or corner members of the tower. The chords are constructed of .30 meter (12 inch) wide-flange H-beams and each chord is comprised of three sections of a different size H-beam in each section, the heaviest being near the base and the lightest extending to the top of the tower.

The diagonals provide the shear and torsional strength of the tower. They are constructed of various sizes of H-beams and double angles. Again, the heavier members occur near the base of the tower. The purpose of the horizontal braces is to stabilize the diagonals and reduce their effective column length. The horizontal braces carry no primary loads.

500 KW Steel Truss Tower - The 500 KW truss tower is similar to the 1500 KW tower. The 500 KW is shorter; 33.53 meters (110 feet) compared to 36.88 meters (121 feet) and has one less panel. The width of the top is the same to accommodate the same bearing as the 1500 KW and the slopes of the sides are identical, giving a base width of 9.75 meters (32 feet) for the 500 KW tower. The design is shown in Figure 6-5.

Structurally, the major difference between the two towers is the size of the chords. The chords of the 500 KW tower are .20 meter (8 inch) wide-flange H-beams, rather than the .30 meter (12 inch) H-beams of the 1500 KW tower. Each chord is comprised of three sections and the heaviest is at the base while the lightest extends to the top.

1500 KW Pre-cast, Post-tensioned Concrete Tower - The pre-cast concrete tower for the 1500 KW design is constructed of twelve factory match-cast segments, six upper and six lower. Figure 6-6 shows the tower design. The lower six segments are identical, except for a doorway in one, while each of the upper six are identical. The upper and lower segments are each 18.90 meters (62 ft) in length and, when assembled, produce a tower whose top is 37.80 meters (124 feet) above the foundation. Maximum size of each segment is determined by shipping considerations of weight and dimension. The tower is a truncated circular cone with a constant cone angle and constant wall thickness of .15 meters (6 in). The top diameter is 3.05 meters (10 feet) while the base diameter is 7.32 meters (24 feet). The .15 meter (6 in) wall is the minimum recommended by a pre-cast concrete manufacturer for this construction.

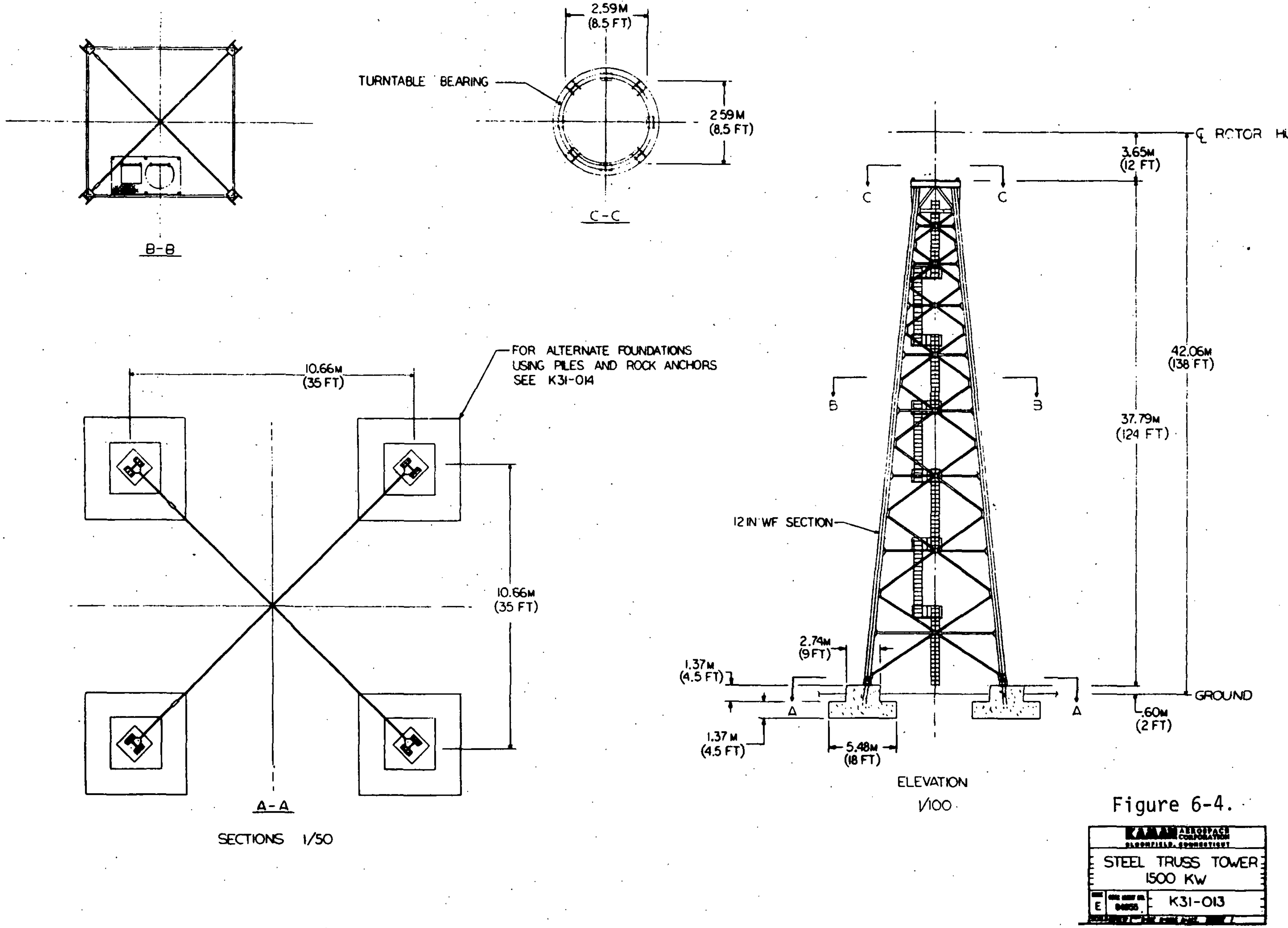
Post-tensioning is accomplished during erection of the tower in two stages; the first post-tensioning is applied after the lower half is erected, while the second post-tensioning is applied after erection of the upper half. Post-tensioning strands are positioned at twelve locations around the circumference of the shell and within the wall thickness which is thickened locally to provide protection to the strands. Shear ties between the segments are required along vertical joints to provide structural continuity. Horizontal ties may not be required if the post-tensioning load is large enough.

Access to the tower top is provided by caged ladders in the shell interior.

500 KW Pre-cast, Post-tensioned Concrete Tower - The 500 KW pre-cast, post-tensioned concrete tower is identical to the 1500 KW tower, except for the height of the lower segments. The tower design is shown in Figure 6-7. The total tower height is 33.53 meters (110 feet) above the foundation vs 37.80 meters (124 feet) for the 1500 KW tower, but the top diameter is 3.05 meters (10 feet). Consequently, the upper segments are 18.90 meters (62 feet) in length while the lower segments are 14.63 meters (48 feet). The resulting base diameter is 6.83 meters (22.42 feet). The minimum wall thickness of .15 meters (6 in) was held throughout. The upper segments can be interchangeable between the 500 KW and 1500 KW systems, while the lower segment is similar but shorter. A single form can be used for the upper segments of both towers and, possibly, for the lower segments, also.

Mass Concrete Foundation - The mass concrete foundation for the truss tower designs consists of four stepped blocks of concrete at each of the four chords (legs) of the tower. For the 1500 KW design, the maximum dimensions of the block were 5.49 x 5.49 x 2.74 meters (18 x 18 x 9 feet). The 500 KW design had a block 4.57 x 4.57 x 2.29 meters (15 x 15 x 7.5 feet). The foundation designs are shown in Figures 6-4 and 6-5.

The mass concrete foundation for the shell tower design consists of one large stepped, cylindrical block of concrete. The dimensions for the 1500 KW system are 14.63 meters (48 feet) OD, 6.10 meters (20 feet) ID and 2.44 meters (8 feet) deep. For the 500 KW design, the corresponding dimensions are 10.97 meters (36 feet), 4.89 meters (16 feet) and 1.83 meters (6 feet). The designs are shown in Figures 6-8 and 6-9.



B-B
TURNTABLE BEARING
2.59M
(8.5 FT)
2.59M
(8.5 FT)
C-C
FOR ALTERNATE FOUNDATIONS
USING PILES AND ROCK ANCHORS
SEE K31-014
10.66M
(35 FT)
10.66M
(35 FT)
A-A
SECTIONS 1/50
C
C
B
B
12 IN WF SECTION
3.65M
(12 FT)
42.06M
(138 FT)
37.79M
(124 FT)
2.74M
(9 FT)
1.37M
(4.5 FT)
1.37M
(4.5 FT)
A
A
5.48M
(18 FT)
.60M
(2 FT)
GROUND
ELEVATION
1/100
KAMAN AEROSPACE CORPORATION
BLOOMFIELD, CONNECTICUT
STEEL TRUSS TOWER
1500 KW
E
K31-013

Figure 6-4.

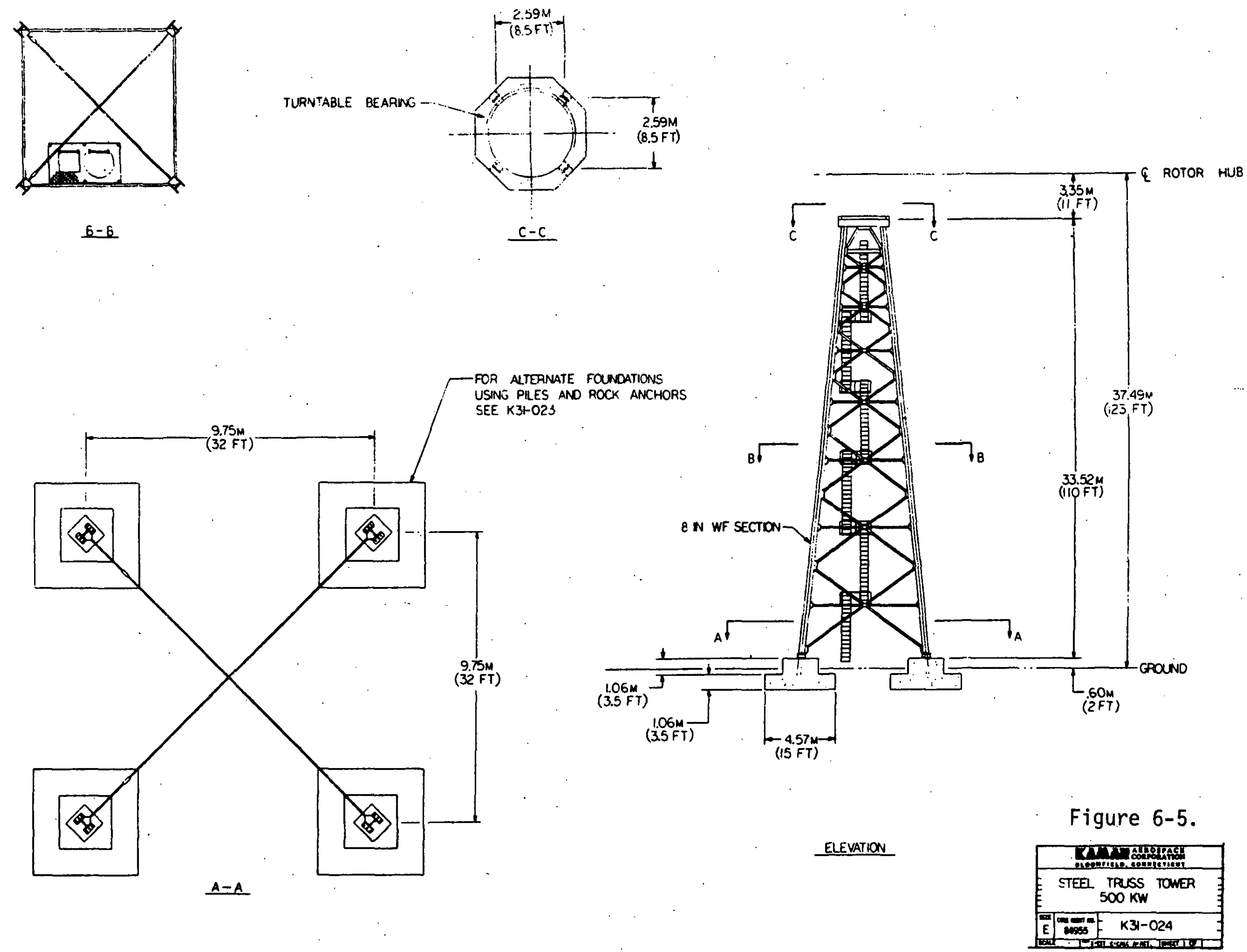

6-6
TURNTABLE BEARING
2.59M (8.5 FT)
2.59M (8.5 FT)
C-C
FOR ALTERNATE FOUNDATIONS USING PILES AND ROCK ANCHORS SEE K31-023
9.75M (32 FT)
9.75M (32 FT)
A-A
℄ ROTOR HUB
3.35M (11 FT)
C
C
37.49M (125 FT)
33.52M (110 FT)
B
B
8 IN WF SECTION
A
A
GROUND
1.06M (3.5 FT)
1.06M (3.5 FT)
4.57M (15 FT)
.60M (2 FT)
ELEVATION
KAMAN AEROSPACE CORPORATION
BLOOMFIELD, CONNECTICUT
STEEL TRUSS TOWER
500 KW
E
94955
K31-024

Figure 6-5.

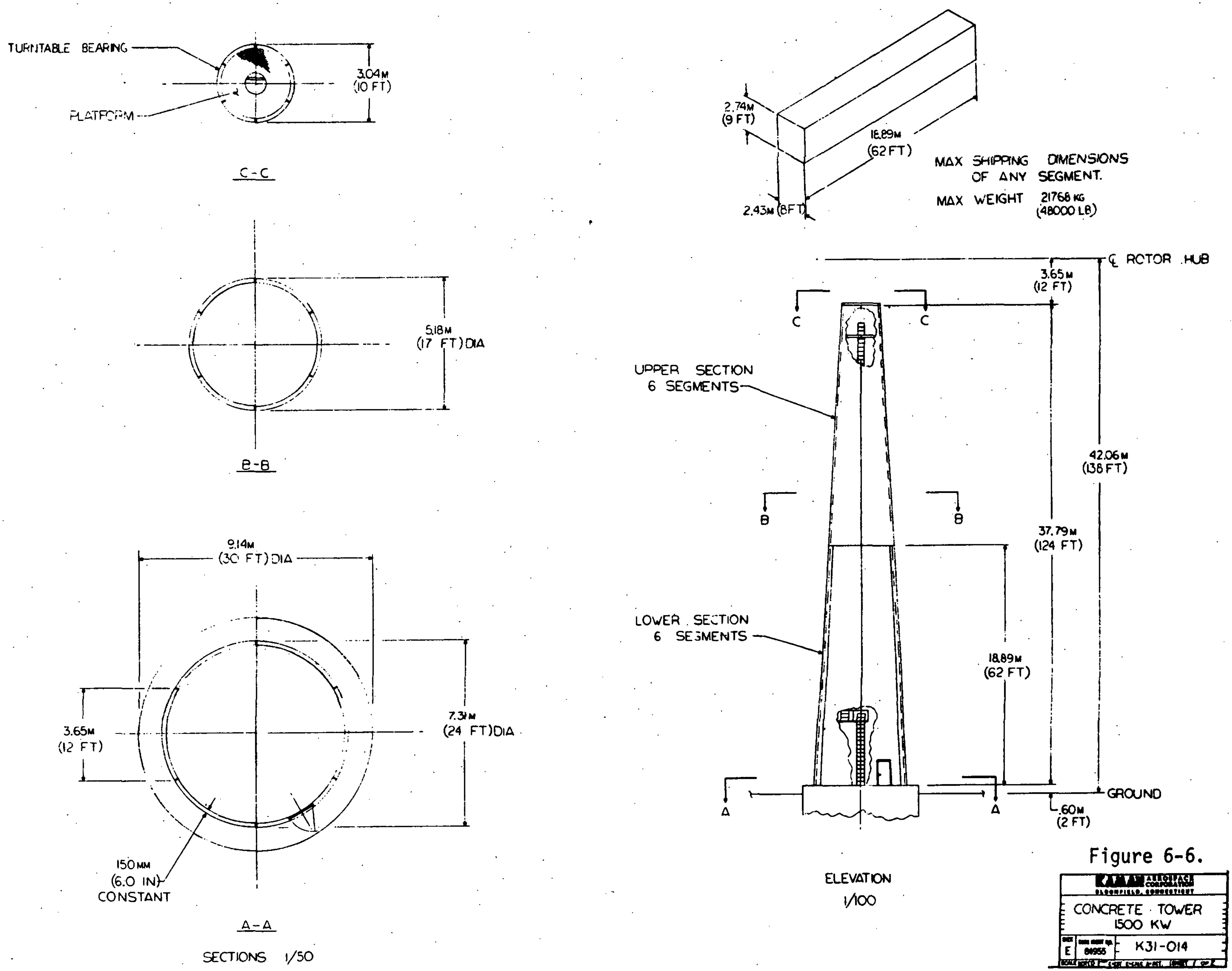
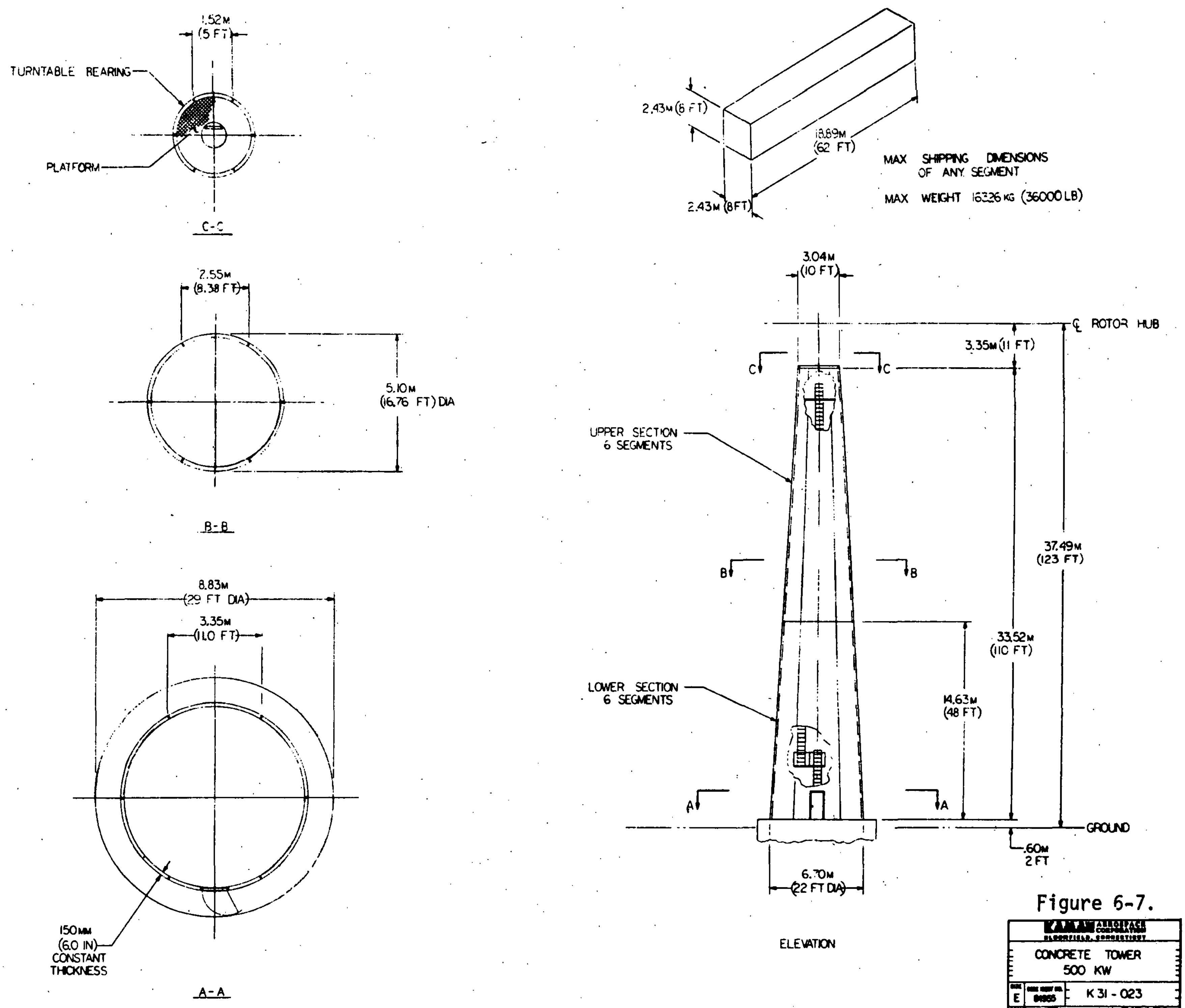

TURNTABLE BEARING
PLATFORM
3.04M (10 FT)
C-C
5.18M (17 FT) DIA
B-B
9.14M (30 FT) DIA
3.65M (12 FT)
7.31M (24 FT) DIA
150MM (6.0 IN) CONSTANT
A-A
SECTIONS 1/50
2.74M (9 FT)
18.89M (62 FT)
2.43M (8 FT)
MAX SHIPPING DIMENSIONS OF ANY SEGMENT.
MAX WEIGHT 21768 KG (48000 LB)
℄ ROTOR HUB
3.65M (12 FT)
UPPER SECTION 6 SEGMENTS
LOWER SECTION 6 SEGMENTS
42.06M (138 FT)
37.79M (124 FT)
18.89M (62 FT)
GROUND
.60M (2 FT)
ELEVATION 1/100
KAMAN AEROSPACE CORPORATION
BLOOMFIELD, CONNECTICUT
CONCRETE TOWER
1500 KW
E
K31-014

Figure 6-6.



1.52M
(5 FT)
TURNTABLE BEARING
PLATFORM
C-C
2.55M
(8.38 FT)
5.10M
(16.76 FT) DIA
B-B
8.83M
(29 FT DIA)
3.35M
(11.0 FT)
150MM
(6.0 IN)
CONSTANT
THICKNESS
A-A
2.43M (6 FT)
18.89M
(62 FT)
2.43M (8FT)
MAX SHIPPING DIMENSIONS
OF ANY SEGMENT
MAX WEIGHT 16326 KG (36000 LB)
3.04M
(10 FT)
℄ ROTOR HUB
3.35M (11 FT)
C
C
UPPER SECTION
6 SEGMENTS
B
B
37.49M
(123 FT)
33.52M
(110 FT)
LOWER SECTION
6 SEGMENTS
14.63M
(48 FT)
A
A
GROUND
.60M
2 FT
6.70M
(22 FT DIA)
ELEVATION
CONCRETE TOWER
500 KW
E
K 31 - 023

Figure 6-7.

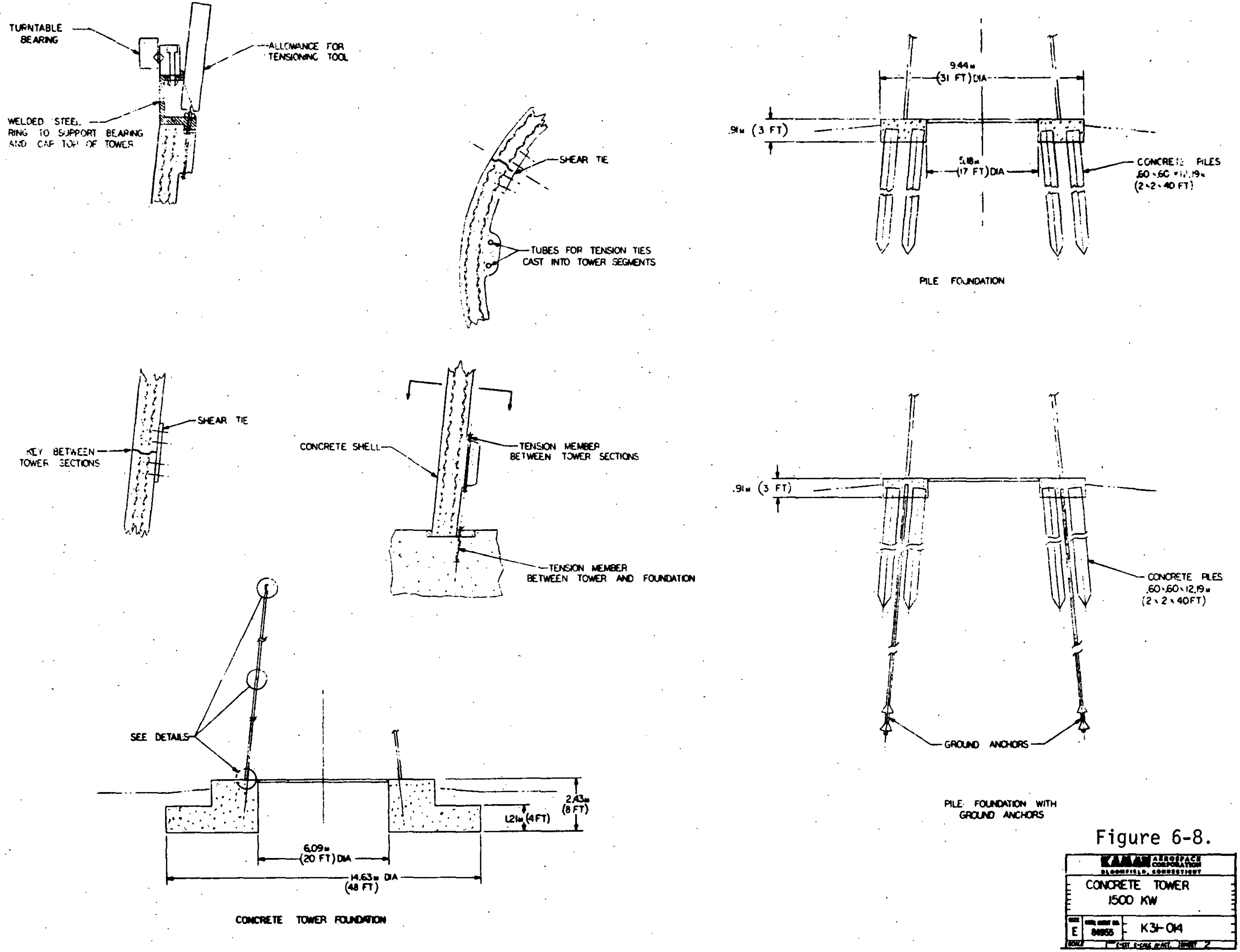
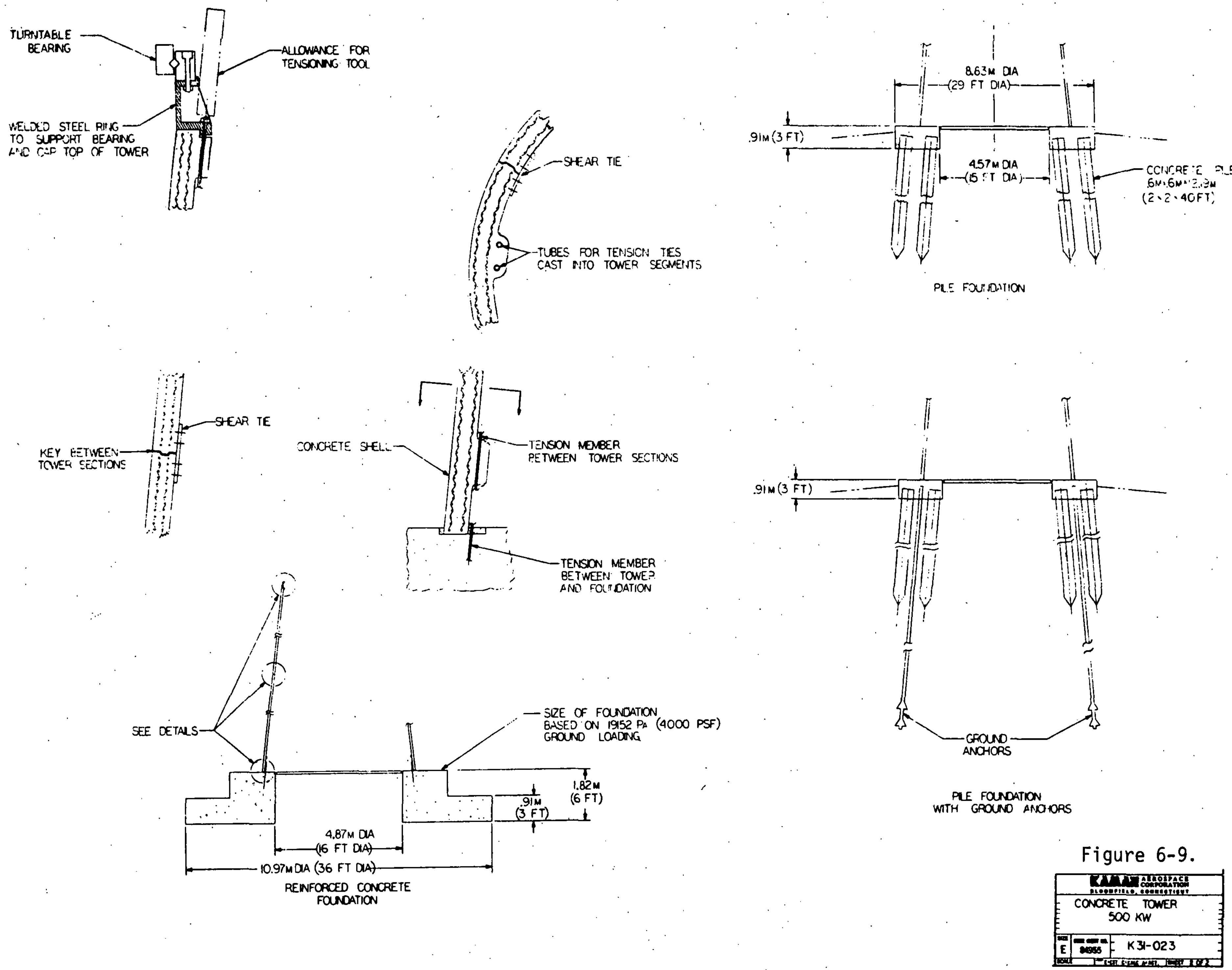

TURNTABLE BEARING
ALLOWANCE FOR TENSIONING TOOL
WELDED STEEL RING TO SUPPORT BEARING AND CAP TOP OF TOWER
SHEAR TIE
TUBES FOR TENSION TIES CAST INTO TOWER SEGMENTS
9.44m (31 FT) DIA
.91m (3 FT)
5.18m (17 FT) DIA
CONCRETE PILES .60×.60×12.19m (2×2×40 FT)
PILE FOUNDATION
SHEAR TIE
KEY BETWEEN TOWER SECTIONS
CONCRETE SHELL
TENSION MEMBER BETWEEN TOWER SECTIONS
TENSION MEMBER BETWEEN TOWER AND FOUNDATION
.91m (3 FT)
CONCRETE PILES .60×.60×12.19m (2×2×40FT)
GROUND ANCHORS
PILE FOUNDATION WITH GROUND ANCHORS
SEE DETAILS
2.43m (8 FT)
1.21m (4 FT)
6.09m (20 FT) DIA
14.63m DIA (48 FT)
CONCRETE TOWER FOUNDATION
KAMAN AEROSPACE CORPORATION BLOOMFIELD, CONNECTICUT
CONCRETE TOWER 1500 KW
E
K31-014

Figure 6-8.



TURNTABLE BEARING
ALLOWANCE FOR TENSIONING TOOL
WELDED STEEL RING TO SUPPORT BEARING AND CAP TOP OF TOWER
SHEAR TIE
TUBES FOR TENSION TIES CAST INTO TOWER SEGMENTS
8.63M DIA (29 FT DIA)
.91M (3 FT)
4.57M DIA (15 FT DIA)
PILE FOUNDATION
SHEAR TIE
KEY BETWEEN TOWER SECTIONS
CONCRETE SHELL
TENSION MEMBER BETWEEN TOWER SECTIONS
TENSION MEMBER BETWEEN TOWER AND FOUNDATION
.91M (3 FT)
SEE DETAILS
SIZE OF FOUNDATION BASED ON 19152 PA (4000 PSF) GROUND LOADING
1.82M (6 FT)
.91M (3 FT)
4.87M DIA (16 FT DIA)
10.97M DIA (36 FT DIA)
REINFORCED CONCRETE FOUNDATION
GROUND ANCHORS
PILE FOUNDATION WITH GROUND ANCHORS
KAMAN AEROSPACE CORPORATION BLOOMFIELD, CONNECTICUT
CONCRETE TOWER 500 KW
E
K31-023

Figure 6-9.

Pile Foundation - For the 1500 KW truss tower pile foundation, seven .61 meter (24 inch) square piles, 12.19 meters (40 feet) in length were necessary to react the tension (uplift) load at each of the four legs for the assumed conditions defined in paragraph 6.4.3. Four .61 meter (24 inch) square piles, 12.19 meters (40 feet) in length were required at each of the four legs of the 500 KW system.

The 1500 KW system shell tower required forty-seven .61 meter (24 inch) square piles at a spacing of .49 meters (1.6 feet) around the base diameter of 7.32 meters (24 feet). The 500 KW system required twenty-seven .61 meter (24 inch) square piles at a spacing of .80 meters (2.63 feet) around the base diameter of 6.83 meters (22.42 feet). The pile foundation designs are shown in Figures 6-8 and 6-9.

Pile Foundation With Rock Anchors - The 1500 KW truss tower design required three .61 (24 inch) square piles plus one rock anchor at each of the four chords. The 500 KW truss tower design required three .36 meter (14 inch) square piles plus one rock anchor at each of the four chords.

The 1500 KW shell tower foundation required twenty-seven .61 meter (24 inch) square piles and seven rock anchors around the 7.32 meter (24 foot) base diameter. The 500 KW shell tower foundation required nineteen .61 meter (24 inch) square piles and six rock anchors around the 6.83 meter (22.42 foot) base diameter. These designs are also shown in Figures 6-8 and 6-9.

6.5.2 Design Analysis

The analyses performed to support the preliminary design for both of the selected tower concepts included load analysis, strength analysis and vibration analysis. A description and the results of the analyses performed are presented on the following pages, and are based on the preliminary design tower design parameters presented in Table 6-12.

6.5.2.1 Load Analysis

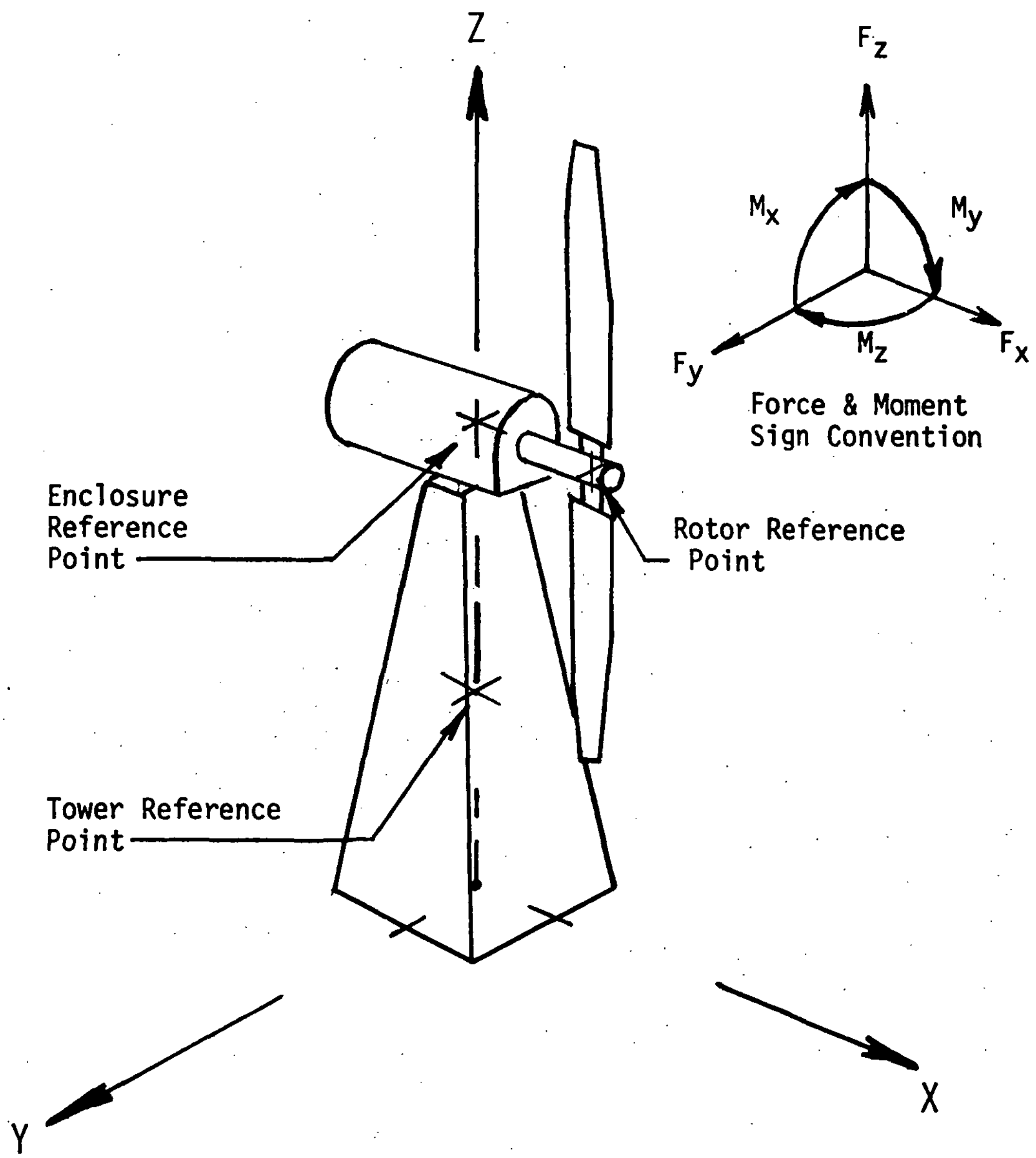
The applied loads were derived from the statement of the design criteria. Paragraph 6.4.2 presents these criteria and a schematic showing the loads on the tower, enclosure and rotor. Figure 6-10 shows the load sign convention and reference points.

The loads for each condition are given in Table 6-13 for both the 500 KW and 1500 KW WGS truss and shell towers. The loads were all transferred to convenient reference points for presentation. In the actual analysis, the loads were not always applied at these points. The wind and seismic loads on the tower, for instance, were applied as distributed loads. The tower weight was also applied as a distributed load for the strength analysis of the towers. The tower weight and weight of the turntable assembly were, of course, constant for each load condition, varying only for the different size tower and tower concept.

In addition to the shear load produced by the 53.6 m/sec (120 mph) wind impinging on the blades in the upright position, there is a rotor torque produced because of the difference in the wind force on each blade. This difference was

TABLE 6-12. PRELIMINARY DESIGN TOWER PARAMETERS

	TRUSS		SHELL	
	500 Kw	1500 Kw	500 Kw	1500 Kw
HEIGHT, m (ft)				
HUB Above Ground	37.49 (123)	42.06 (138)	37.49 (123)	42.06 (138)
HUB Above Foundation	36.88 (121)	41.45 (136)	36.88 (121)	41.45 (136)
TOWER Above Ground	34.14 (112)	38.40 (126)	34.14 (112)	38.40 (126)
TOWER Above Foundation	33.53 (110)	37.40 (124)	33.53 (110)	37.40 (124)
TOP DIMENSION, m(ft)	2.59 (8.5) Sq.	2.59 (8.5) Sq.	3.05 (10) Dia.	3.05 (10) Dia.
BASE DIMENSION, m (ft)	9.75 (32) Sq.	10.67 (35) Sq.	6.83 (22.4) Dia.	7.32 (24) Dia.
HUB OFFSET, m (ft)	2.13 (7)	2.44 (8)	2.13 (7)	2.44 (8)
ROTOR INCLINATION, DEGREES	15	15	15	15
WEIGHT, mg (kips)				
Rotating Assembly	31.8 (70)	66.7 (147)	31.8 (70)	66.7 (147)
Tower	29.5 (65)	40.8 (90)	200.5 (442)	237.2 (523)
MASS MOMENT OF INERTIA, kg-m^2 (Slug-ft^2)				
Rotating Assembly	73,214 (54000)	352,512 (260000)	73,214 (54000)	352,512 (260000)



	500 Kw						1500 Kw					
	x (Ft.)	x m	y (Ft.)	y m	z (Ft.)	z m	x (Ft.)	x m	y (Ft.)	y m	z (Ft.)	z m
Rotor Reference Point	7.0	2.13	0	0	123.0	37.49	8.0	2.44	0	0	138.0	42.06
Enclosure Reference Point	0	0	0	0	123.0	37.49	0	0	0	0	138.0	42.06
Tower Reference Point	0	0	0	0	56.0	17.07	0	0	0	0	63.0	19.20

Figure 6-10. Sign Convention and Reference Points.

TABLE 6-13(a). SUMMARY OF APPLIED LOADS

500 Kw Truss Tower

Load Condition	Load Reference Point	F_x (kips)	F_x kN	F_y (kips)	F_y kN	F_z (kips)	F_z kN	M_x (ft-kips)	M_x kN.m	M_y (ft-kips)	M_y kN.m	M_z (ft-kips)	M_z kN.m
(1) Blow-Over	Rotor			52.4*	233.1*			-172.0*	-233.2*			-46.0*	-62.4*
	Enclosure			8.6*	38.3*	-70.5	-313.6					-60.4*	-81.9*
	Tower			50.0	222.4	-65.0	-289.1						
(2) Normal Operation Plus Earthquake	Rotor	13.8	61.4			3.7	16.5	150.0	203.4	101.4	137.5	-88.0	-119.3
	Enclosure	17.6*	78.3*			-70.5	-313.6						
	Tower	16.3*	72.5*			-65.0	-289.1						
(3) Maximum Operation	Rotor	27.6	122.8			7.4	32.9	358.2	485.7	202.8	275.0	-160.0	-216.9
	Enclosure					-70.5	-313.6						
	Tower					-65.0	-289.1						

* The applied load used in the Truss Tower Analysis is .75 of this value, as the AISC Manual of Steel Construction, Pg. 5-22, Para. 1.5.6, allows the allowable stresses to be increased one-third for loads due to wind or seismic loading. It was decided to reduce the load rather than increase the allowable.

TABLE 6-13(b). SUMMARY OF APPLIED LOADS

1500 Kw Truss Tower

Load Condition	Load Reference Point	F_x		F_y		F_z		M_x		M_y		M_z	
		(kips)	kN	(kips)	kN	(kips)	kN	(ft-kips)	kN.m	(ft-kips)	kN.m	(ft-kips)	kN.m
Blow-Over	Rotor			73.1*	325.2			-346.5*	-469.8			-92.9*	-126.0
	Enclosure			19.4*	86.3	-146.6	-652.1					-135.8*	-184.1
	Tower			64.9	288.7	-89.7	-399.0						
Normal Operation Plus Earthquake	Rotor	32.4	144.1			8.7	38.7	371.1	503.1	145.8	197.7	-84.8	-115.0
	Enclosure	36.7*	163.2			-146.6	-652.1						
	Tower	22.4*	99.6			-89.7	-399.0						
Maximum Operation	Rotor	64.6	287.4			17.3	77.0	905.0	1227.0	291.6	395.4	-136.0	-184.4
	Enclosure					-146.6	-652.1						
	Tower					-89.7	-399.0						

*Refer to the Note on the previous page.

TABLE 6-13(c). SUMMARY OF APPLIED LOADS

500 Kw Shell Tower

Load Condition	Load Reference Point	F_x (kips)	F_x kN	F_y (kips)	F_y kN	F_z (kips)	F_z kN	M_x (ft-kips)	M_x kNm	M_y (ft-kips)	M_y kNm	M_z (ft-kips)	M_z kNm
Blow-Over	Rotor			52.4	233.1			-172.0	-233.2			-46.0	-62.4
	Enclosure			8.6	38.3	-70.5	-313.6					-60.4	-81.9
	Tower			43.3	192.6	-442	-1966.1						
Normal Operation Plus Earthquake	Rotor	13.8	61.4			3.7	16.5	150.0	203.4	101.4	137.5	-88.0	-119.3
	Enclosure	11.3*	50.3			-70.5	-313.6						
	Tower	70.7*	314.5			-442	-1966.1						
Maximum Operation	Rotor	27.6	122.8			7.4	32.9	358.2	485.7	202.8	275.0	-160.0	-216.9
	Enclosure					-70.5	-313.6						
	Tower					-442	-1966.1						

* Seismic force load factor = 0.16 Vs. 0.25 factor used on truss tower, Ref. Section 6.4.1.

TABLE 6-13(d). SUMMARY OF APPLIED LOADS

1500 Kw Shell Tower

Load Condition	Load Reference Point	F_x		F_y		F_z		M_x		M_y		M_z	
		(kips)	kN	(kips)	kN	(kips)	kN	(ft-kips)	kNm	(ft-kips)	kNm	(ft-kips)	kNm
Blow-Over	Rotor			73.1	325.2			-346.5	-469.8			-92.9	-126.0
	Enclosure			19.4	86.3	-146.6	-652.1					-135.8	-184.1
	Tower			55.4	246.4	-523	-2326.4						
Normal Operation Plus Earthquake	Rotor	32.4	144.1			8.70	38.7	371.1	503.1	145.8	197.7	-84.8	-115.0
	Enclosure	23.5*	104.5			-146.6	-652.1						
	Tower	83.7*	372.3			-523	-2326.4						
Maximum Operation	Rotor	64.6	287.4			17.3	77.0	905	1227.0	292	395.9	-136.0	-184.4
	Enclosure					-146.6	-652.1						
	Tower					-523	-2326.4						

* Seismic force load factor = 0.16 Vs. 0.25 factor used on truss tower, Ref. Section 6.4.1.

due to the higher wind velocity occuring at the greater height on the upper blade. Because the axis of rotation of the blades is 15° to the horizontal plane, this rotor torque is shown as two components (M_x and M_z), an overturning moment on the tower plus a tower torsion. The 53.6 m/sec (120 mph) wind striking the enclosure also produces a torsion on the tower because the center of area of the enclosure does not coincide with the centerline of the tower. For the strength analysis of the towers, the wind loads were reduced to .75 of these values to take into account the higher allowables that are available for this type of load (reference AISC Manual, Pg. 5-22, paragraph 1.5.6).

The wind loads for the normal operating plus earthquake loads were applied to the enclosure and to the tower. Except for the weights (tower and turntable assembly), the other loads were transformed to the tower coordinate system. Again, the earthquake loads, like the wind loads were reduced to .75 of the value shown for analysis.

Wind loads were negligible for the maximum operating condition as well. Except for the weights of the tower and turntable assembly, the only applied loads were the maximum rotor loads. Because of the rotor shaft being inclined at 15° to the horizontal plane, these loads were transformed to the tower coordinate system.

The foundation loads were derived by transferring all the applied loads to the base of each tower. These loads are summarized in Table 6-14.

6.5.2.2 Truss Tower Strength and Fatigue Analysis

The truss tower strength analysis was accomplished with the use of a computerized three-dimensional finite element model. The model contained the primary load carrying members of the truss towers and enabled accurate internal load and deflection calculations for all loading conditions. The deflections were important in the calculation of the natural frequency.

Results of the strength and fatigue analyses for the 500 KW and 1500 KW truss towers are summarized in Tables 6-15 and 6-16, using the nomenclature and panel definitions shown in Figure 6-11 for the 1500 KW truss tower. Nomenclature for the 500 KW tower is the same, except that the smaller tower has one less panel.

The sizing of the truss towers was an iterative process. A configuration was chosen with a particular base size, number of panels and cross-sectional area for each truss member. The resultant internal loads from the model were checked against allowable loads from the AISC manual. Deflections from unit load conditions from the model run were then used to determine the natural frequencies of the tower. This process was continued; changing configuration and truss member cross-sectional area until acceptable load levels were achieved, along with acceptable natural frequencies. The preliminary designs evolved by this process, while not totally optimized, have near optimum configurations, adequate strength and acceptable natural frequencies.

Because it was convenient to reduce the applied load (by a factor of .75) due to wind and seismic forces to take into account the increased allowables for load conditions due to these forces (reference AISC Manual, Pg. 5-22), it should

TABLE 6-14. SUMMARY OF FOUNDATION LOADS

500 KW AND 1500 KW

TRUSS AND SHELL TOWER

Tower	Basic Loading Condition	F_x (kips)	F_x kN	F_y (kips)	F_y kN	F_z (kips)	F_z kN	M_x (ft-kips)	M_x MNm	M_y (ft-kips)	M_y MNm	M_z (ft-kips)	M_z MNm
500 KW Truss	1	-		111	493.8	-136	-605.0	-10300	-13.965	-	-	260	.353
	2	47.7	212.2	-	-	-132	-587.2	150	.203	4800	6.508	-90	-.122
	3	27.6	122.8	-	-	-128	-569.4	360	.488	3540	4.800	-160	-.217
500 KW Shell	1	-		104	462.6	-512	-2277.5	-9930	-13.463	0	0	260	.353
	2	95.8	426.1	-	-	-509	-2264.1	150	.203	7030	9.531	-90	-.122
	3	27.6	122.8	-	-	-505	-2246.4	360	.488	3540	4.800	-160	-.217
1500 KW Truss	1	-		157	698.4	-236	-1049.8	-17000	-23.049	0	0	360	.488
	2	91.5	407.0	-	-	-228	-1014.2	370	.502	10900	14.778	-90	-.122
	3	64.6	287.4	-	-	-219	-974.2	900	1.220	9080	12.311	-140	-.190
1500 KW Shell	1	-		150	667.2	-670	-2980.3	-16400	-22.235	0	0	360	.488
	2	140	622.8	-	-	-661	-2940.3	370	.502	12900	17.490	-90	-.122
	3	64.6	287.4	-	-	-652	-2900.2	900	1.220	9080	12.311	-140	-.190

Conditions:
1 53.6 m/sec (120 mph) Blowover Condition.
2 Normal Operating Plus Seismic Force.
3 Maximum Operating Condition.

TABLE 6-15(a). STRENGTH ANALYSIS RESULTS

500 Kw Truss Tower

Panel	Truss Member	Section Designation	Area (in^2)	Area mm^2	Effective Length (ft)	Effective Length m	Load** (kips)	Load** kN	Maximum** Stress (ksi)	Maximum** Stress MPa	L/ρ	Allowable Load (kips)	Allowable Load kN
1	C	W8 x 24	7.06	4.55	7.9	2.41	-85	-378.1	-11.94	-82.32	58.8	-123	-547
	D	3-1/2 x 3 x 13.2	3.9	2.52	8.4	2.56	-20.7	-92.1	-5.30	-36.54	91.6	-46	-205
	B	W6 x 8.5	2.51	1.62	11.1	3.38	0	0	0	0	55.0	N.A.	N.A.
2	C	W8 x 24	7.06	4.55	9.3	2.83	-97	-431.5	-13.73	-94.67	69.3	-113	-503
	D	3-1/2 x 3 x 13.2	3.9	2.52	8.1	2.47	-13.3	-59.2	-3.40	-23.44	88.4	-46	-205
	B	W6 x 8.5	2.51	1.62	12.1	3.69	0	0	0	0	60.0	N.A.	N.A.
3	C	W8 x 24	7.06	4.55	10.5	3.20	-105	-467.1	-14.85	-102.39	78.2	-107	-476
	D	3-1/2 x 3 x 13.2	3.9	2.52	9.4	2.87	-12.9	-57.4	-3.30	-22.75	103.0	-46	-205
	B	W6 x 8.5	2.51	1.62	13.9	4.24	0	0	0	0	68.8	N.A.	N.A.
4	C	W8 x 40	11.8	7.61	11.4	3.47	-116	-516.0	-9.82	-67.71	67.0	-193	-859
	D	4 x 3 x 14.4	4.2	2.71	10.7	3.26	-12.1	-53.8	-2.87	-19.79	101.0	-47	-209
	B	W6 x 15.5	4.6	2.97	16.2	4.94	0	0	0	0	75.7	N.A.	N.A.
5	C	W8 x 40	11.8	7.61	13.8	4.21	-122	-542.7	-10.30	-71.02	81.2	-178	-792
	D	W6 x 15.5	4.6	2.97	12.5	3.81	-11.9	-52.9	-2.58	-17.79	103.0	-55	-245
	B	W6 x 15.5	4.6	2.97	18.8	5.73	0	0	0	0	87.8	N.A.	N.A.
6	C	W8 x 40	11.8	7.61	15.9	4.85	-132	-587.2	-11.20	-77.22	93.5	-162	-721
	D	W6 x 20	5.9	3.81	14.5	4.42	-13.2	-58.7	-2.24	-15.44	115.0	-68	-302
	B	W6 x 15.5	4.6	2.97	21.8	6.64	0	0	0	0	102.0	N.A.	N.A.
7	C	W8 x 58	17.1	11.03	18.2	5.55	-135	-600.5	-7.90	-54.47	104.2	-203	-903
	D	W8 x 24	7.06	4.55	16.7	5.09	-12.8	-56.9	-1.81	-12.48	125.0	-66	-294
	B	W8 x 24	7.06	4.55	25.3	7.71	0	0	0	0	89.0	N.A.	N.A.
8	C	W8 x 58	17.1	11.03	21.0*	6.40	-147	-653.9	-8.60	-59.24	120.0	-162	-721
	D	W8 x 31	9.1	5.87	19.5	5.94	-28.4	-126.3	-3.12	-21.51	117.0	-95	-423
	B	W8 x 24	7.06	4.55	29.2	8.90	0	0	0	0	103.0	N.A.	N.A.

* The effective length here was considered to be .90 actual length, taking advantage of some end fixity of the joint. N.A. -Not Applicable

** These loads and stresses are due to applied loads reduced by a .75 factor (Ref. discussion Section 6.5.2.2).

TABLE 6-15(b). STRENGTH ANALYSIS RESULTS

1500 Kw Truss Tower

Panel	Truss Member	Section Designation	Area (in^2)	Area mm^2	Effective Length (ft)	Effective Length m	Load** (kips)	Load** kN	Maximum** Stress (ksi)	Maximum** Stress MPa	L/ρ	Allowable Load (kips)	Allowable Load kN
1	C	W12 x 50	14.7	9.48	6.8	2.07	-141.0	-627.2	-9.60	-66.19	41.7	-279	-1241
	D	3-1/2 x 3 x 13.2	3.9	2.52	8.3	2.53	-30.6	-136.1	-7.85	-54.12	90.6	-46	-205
	B	W6 x 8.5	2.51	1.62	10.7	3.26	0	0	0	0	52.9	N.A.	N.A.
2	C	W12 x 50	14.7	9.48	8.7	2.65	-152.0	-676.1	-10.34	-71.29	48.4	-271	-1205
	D	3-1/2 x 3 x 13.2	3.9	2.52	7.3	2.23	-18.3	-81.4	-4.69	-32.34	79.6	-56	-249
	B	W6 x 8.5	2.51	1.62	11.4	3.47	0	0	0	0	56.3	N.A.	N.A.
3	C	W12 x 50	14.7	9.48	9.7	2.96	-168.0	-747.3	-11.42	-78.74	57.0	-254	-1130
	D	3-1/2 x 3 x 13.2	3.9	2.52	8.5	2.59	-18.0	-80.1	-4.62	-31.85	92.7	-46	-205
	B	W6 x 8.5	2.51	1.62	13.4	4.08	0	0	0	0	66.2	N.A.	N.A.
4	C	W12 x 50	14.7	9.48	11.3	3.44	-185.0	-822.9	-12.59	-86.80	66.8	-246	-1094
	D	3-1/2 x 3 x 13.2	3.9	2.52	9.9	3.02	-15.1	-67.2	-3.88	-26.75	108.0	-46	-205
	B	W6 x 8.5	2.51	1.62	15.5	4.72	0	0	0	0	76.6	N.A.	N.A.
5	C	W12 x 72	21.2	13.68	13.2	4.02	-203.0	-903.0	-9.55	-65.84	50.2	-387	-1721
	D	4 x 3 x 14.4	4.2	2.71	11.6	3.54	-15.1	-67.2	-3.60	-24.82	109.6	-47	-209
	B	W6 x 15.5	4.56	2.94	18.0	5.49	0	0	0	0	84.1	N.A.	N.A.
6	C	W12 x 72	21.2	13.68	15.1	4.60	-220.0	-978.6	-10.38	-71.57	58.8	-371	-1650
	D	W6 x 15.5	4.6	2.97	13.5	4.11	-12.5	-55.6	-2.71	-18.68	111.0	-50	-222
	B	W6 x 15.5	4.56	2.94	20.9	6.37	0	0	0	0	97.5	N.A.	N.A.
7	C	W12 x 72	21.2	13.68	17.4	5.30	-241.0	-1072.0	-11.34	-78.19	68.7	-346	-1539
	D	W6 x 20	5.9	3.81	15.8	4.82	-14.4	-64.1	-2.44	-16.82	126.0	-54	-240
	B	W6 x 15.5	4.56	2.94	24.2	7.38	0	0	0	0	113.0	N.A.	N.A.
8	C	W12 x 99	29.1	18.77	20.2	6.16	-263.0	-1169.9	-9.01	-62.12	79.3	-429	-1908
	D	W8 x 24	7.06	4.55	18.5	5.64	-12.7	-56.5	-1.80	-12.41	138.0	-53	-236
	B	W8 x 24	7.06	4.55	28.3	8.63	0	0	0	0	99.5	N.A.	N.A.
9	C	W12 x 99	29.1	18.77	23.0	7.01	-285.0	-1267.7	-9.79	-67.50	92.5	-402	-1788
	D	W8 x 31	9.1	5.87	20.0	6.10	-18.0	-80.1	-1.97	-13.58	120.0	-95	-423
	B	W8 x 24	7.06	4.55	32.5	9.91	0	0	0	0	114.0	N.A.	N.A.

N.A. - Not Applicable

** Refer to note on preceding page.

TABLE 6-16(a). FATIGUE ANALYSIS RESULTS

500 KW TRUSS TOWER

Load Condition	Cycles	Vibratory Load or Moment (Kips) (Ft-Kips)	Vibratory Load or Moment kN kNm	Truss Member (Ref P6)	Maximum Vibratory Stress (ksi)	Maximum Vibratory Stress mPa	Type of Joint	Allowable Vibratory Stress[1] (ksi)	Allowable Vibratory Stress[1] mPa
High Cycle Rotor-Induced Vibratory Hub Moments	30×10^6	$M_z = \pm 62$	±84.0	Diagonals	±.52	±3.59	Single Shear	±7.5	±51.71
							Double Shear	±12.0	±82.74
		$M_y = \pm 51$	±69.1	Chords	±.61	±4.21	Single Shear	±7.5	±51.71
							Double Shear	±12.0	±82.74
							Column With Welded Gussets	±6.0	±41.37
							Butt-Weld (No Inspection)	±6.0	±41.37
Start-Up & Shut-Down	90×10^3	M_y+Thrust $0 \leq$ Thrust ≤ 21.5	95.6	Chords	±3.65	±25.17	Single Shear	±16.5	±113.76
		$0 \leq M_y \leq 152$	206.1				Double Shear	±20.0	±137.90
							Column With Welded Gussets	±14.0	±96.53
							Butt-Weld (No Inspection)	±14.0	±96.53
		$0 \leq M_x \leq 183$	248.1	Chords	±1.1	±7.59	Single Shear	±16.5	±113.76
							Double Shear	±20.0	±137.90
							Column With Welded Gussets	±14.0	±96.53
							Butt-Weld (No Inspection)	±14.0	±96.53

(1) Ref. AISC Manual for allowable stresses, Pgs 5-107 to 5-113.

TABLE 6-16(b). FATIGUE ANALYSIS RESULTS

1500 KW TRUSS TOWER

Load Condition	Cycles	Vibratory Load or Moment (Kips) (Ft-Kips)	Vibratory Load or Moment N kNm	Truss Member Ref P6)	Maximum Vibratory Stress (ksi)	Maximum Vibratory Stress mPa	Type of Joint	Allowable Vibratory Stress[1] (ksi)	Allowable Vibratory Stress[1] mPa
High Cycle Rotor-Induced Vibratory Hub Moments	30×10^6	$M_z = \pm 89$	± 120.7	Diagonals	$\pm .75$	± 5.17	Single Shear Double Shear	± 7.5 ± 12.0	± 51.71 ± 82.74
		$M_y = \pm 73$	± 99.0	Chords	$\pm .44$	± 3.03	Single Shear Double Shear Column With Welded Gussets Butt-Weld (No Inspection	± 7.5 ± 12.0 ± 6.0 ± 6.0	± 51.71 ± 82.74 ± 41.37 ± 41.37
Start-Up & Shut-Down	90×10^3	M_y+Thrust 0 < Thrust < 50.3 $0 \leq M_y \leq 219$	223.7 296.9	Chords	± 4.05	± 27.92	Single Shear Double Shear Column With Welded Gussets Butt-Weld (No Inspection)	± 16.5 ± 20.0 ± 14.0 ± 14.0	± 113.76 ± 137.90 ± 96.53 ± 96.53
		$0 \leq M_x \leq 505$	684.7	Chords	$\pm .91$	± 6.27	Single Shear Double Shear Column With Welded Gussets Butt-Weld (No Inspection)	± 16.5 ± 20.0 ± 14.0 ± 14.0	± 113.76 ± 137.90 ± 96.53 ± 96.53

(1) Ref. AISC Manual for allowable stresses, Pgs 5-107 to 5-113.

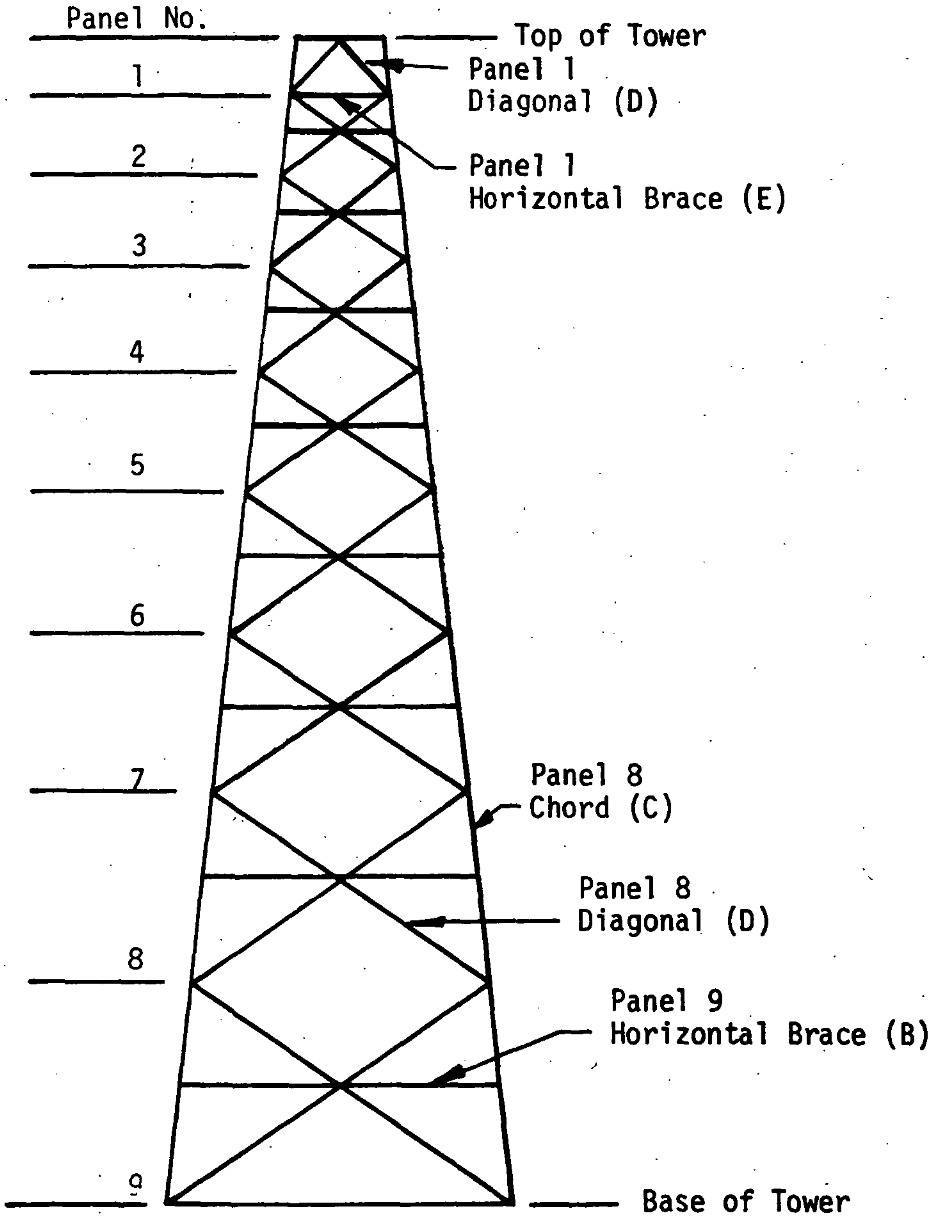


Figure 6-11. Truss Tower Nomenclature

be noted that the actual stresses are higher than those shown on the tables. The stress analysis, however, was based on the stresses shown. Allowable column loads were obtained from the AISC Manual.

The final truss tower configurations are similar for both the 500 KW and 1500 KW systems. The truss towers are four-sided towers constructed by conventional means from standard grade structural steel. The load carrying structure is made up of H-beams and double angles. The chords, the corner members of the truss towers, are the members which provide the primary bending strength of the truss towers. They are the only sections of the towers which were designed primarily by strength. The chords are stability critical and the effective column length was considered to be that total length existing between the intersections of the diagonals and the chords. The diagonals were considered to be the only stabilizing members of the chords, even though the horizontal braces also appear to stabilize the chords.

The horizontal braces are H-beams whose only function is to stabilize the diagonals; that is, to provide a hard point at the intersection of the two diagonals in each panel to allow the effective column length of the diagonals to be that length extending from the intersection at the chord to the intersection of the two diagonals and the horizontal brace. The effectiveness of these hard points at the intersection of the diagonals and horizontal braces is dependent upon the bending stiffness of the horizontal braces. It is recognized that a more efficient manner of stabilizing the diagonals might be possible. Instead of horizontal braces, members could be placed at each diagonal intersection connecting to the two adjacent diagonal intersections. This would result in a square shape that connected all of the diagonal intersections at any particular height. This possible alternative design dictated the effective column lengths of the chords that were used in the analysis. This change in the method of stabilizing the diagonals would have no effect on the rest of the tower design. The natural frequency would be unaffected.

As the design evolved, the required natural frequency became the dominating design factor. If the tower base were made much smaller, the chords would have to be heavier than that dictated by strength in order to provide the necessary natural frequency in bending. Reducing the size of the base affects the natural frequency in torsion similarly, even though the diagonals are not now strength designed.

The fatigue analysis of the two truss towers included two basic fatigue loadings. One loading included those loads produced by the oscillatory motion of the rotor during normal operation. The components of this loading were vibratory hub moments in two directions, producing bending and torsion on the tower. This was a high cycle loading (30×10^6 cycles) and required a stress level below the endurance limit. The other loading that occurred during startup and shutdown operations was assumed to occur five times a day for 50 years. The loading produced by this cycle was a rotor torque and a rotor thrust combined with an overturning hub moment. The load was assumed to vary from 0 to 1.5 times the maximum steady state operating load. Paragraph 6.4.2 gives a more detailed description of the fatigue load conditions. The maximum stresses due to the fatigue conditions were determined from unit load conditions run on the computerized model. These stresses, along with the allowable fatigue stresses, are shown in Table 6-16.

6.5.2.3 Pre-cast, Post-tensioned Concrete Tower Strength and Fatigue Analysis

The 500 KW and 1500 KW pre-cast, post-tensioned concrete towers were analyzed for bending at three sections; the top section where the x-roller bearing transmits the load to the tower; at the midsection; and at the base. Using the same coordinate system and nomenclature as for the truss tower analysis, the internal loads for the concrete towers are given in Table 6-17. Longitudinal stresses due to axial load and bending were calculated and combined. The results are presented in Table 6-18.

The desired pre-load in the concrete was determined at the mid-height and the base so that the concrete remains in compression for the highest loading condition. Resulting concrete pre-stress at the top of the tower and at the mid-height where the combined pre-load is acting is also presented in Table 6-18. The maximum and minimum compression stresses at all three sections due to the application of the post-tensioning load, the dead load and the most critical liveload, are presented in the last two columns of Table 6-18.

It was found that the fatigue loading condition produces low stress levels and is not a critical factor.

6.5.2.4 Tower Vibration Analysis

It is important to prevent tower natural frequencies from coinciding with a rotor operating frequency. Such a coincidence can result in large amplitudes of vibration, high stresses and possible tower structural failure. To accurately determine the tower natural frequencies, a computer program, known as RASA was used. RASA is normally used in the determination of natural frequencies and normal modes of non-uniform helicopter rotors and has the capability of determining the natural frequencies for six types of motion coupling; including fully coupled torsional-flatwise-edgewise motion (relative to the rotor blade plane); partially coupled torsional-flatwise motion or flatwise-edgewise motion; and uncoupled torsional motion, flatwise motion, or edgewise motion.

Input to the program is in the form of a segmented beam with stiffness and mass distribution properties for each segment, including EIs about both lateral axes, GJs about the longitudinal axis, and mass moments of inertia about the lateral axis and longitudinal axis. EIs and GJs for the truss towers were determined based on deflections of the truss towers for unit load conditions from the math models. Using Newmark's numerical method of calculating the deflections and moments, the required EIs and GJs were determined to match the unit load condition deflections from the math models for each joint along the truss towers. EIs, GJs and mass properties of the shell tower were calculated for each segment.

Figure 6-12 presents copies of the input pages for the program used in calculating the tower natural frequencies. All properties are listed, including the structural simulation of the turntable structure and all the mass mounted on top of the tower. Results of the analysis, giving the first four natural frequencies in torsion, longitudinal bending and lateral bending for the four selected towers is presented in Table 6-19.

TABLE 6-17. SUMMARY OF SHELL TOWER INTERNAL LOADS

500 Kw and 1500 Kw

Tower	Section	Basic Loading Condition	F_x (kips)	F_x (kN)	F_y (kips)	F_y (kN)	F_z (kips)	F_z (kN)	M_x (ft-kips)	M_x MNm	M_y (ft-kips)	M_y MNm	M_z (ft-kips)	M_z MNm
500 Kw	Top 110 Ft Up From Foundation	1	-	-	61.0	271.3	-70.5	-313.6	-840	-1.139	0	0	260	.353
		2	25.1	111.7	-	-	-66.8	-297.1	150	.203	380	.515	-90	-.122
		3	27.6	122.8	-	-	-63.1	-280.7	360	.488	510	.691	-160	-.217
	Center 55 Feet Up From Foundation	1	-	-	82.6	367.4	-292	-1298.9	-5390	-7.308	0	0	260	.353
		2	60.5	269.1	-	-	-288	-1281.1	150	.203	3700	5.017	-90	-.122
		3	27.6	122.8	-	-	-284	-1263.3	360	.488	2020	2.739	-160	-.217
1500 Kw	Top 124 Ft Up From Foundation	1	-	-	92.5	411.5	-147	-653.9	-1460	-1.979	0	0	360	.488
		2	55.9	248.7	-	-	-140	-622.8	370	.502	820	1.112	-90	-.122
		3	64.6	287.4	-	-	-129	-573.8	900	1.220	1070	1.451	-140	-.190
	Center 62 Feet Up From Foundation	1	-	-	120.2	534.7	-409	-1819.3	-9090	-12.324	0	0	360	.488
		2	97.8	435.0	-	-	-402	-1788.2	370	.502	6990	9.477	-90	-.122
		3	64.6	287.4	-	-	-391	-1739.3	900	1.220	5200	7.050	-140	-.190

TABLE 6-18(a). STRESSES IN PRECAST, POST-TENSIONED CONCRETE SHELL TOWER.

500 kW TOWER

Section	Elevation Above Foundation		Outside Diameter		Applied Stress with Zero Post-Tensioning				Post Tensioning Stress		Post-Tensioning Plus Applied Stress			
					Max. C		Max. T				Max. C		Min. C	
	(Ft)	(m)	(Ft)	(m)	(Ksi)	mPa	(Ksi)	mPa	(Ksi)	mPa	(ksi)	mPa	(Ksi)	mPa
Top	110	33.53	10	3.05	-.206	-1.42	+.140	+0.97	-.580	-4.00	-.786	-5.42	-.440	-3.03
Center	55	16.76	16.21	4.94	-.480	-3.31	+.316	+2.18	-.350 -.487	-2.41 -3.36	-.830 -.967	-5.72 -6.67	-.034 -.171	-0.23 -1.18
Base	0	0	22.42	6.83	-.477	-3.29	+.270	+1.86	-.350	-2.41	-.827	-5.70	-.080	-0.55

TABLE 6-18(b). STRESSES IN PRECAST, POST-TENSIONED CONCRETE SHELL TOWER.

1500 kW TOWER

Section	Elevation Above Foundation		Outside Diameter		Applied Stress with Zero Post-Tensioning				Post Tensioning Stress		Post-Tensioning Plus Applied Stress			
					Max. C		Max. T				Max. C		Min. C	
	(Ft)	(m)	(Ft)	(m)	(Ksi)	mPa	(Ksi)	mPa	(Ksi)	mPa	(ksi)	mPa	(Ksi)	mPa
Top	124	37.80	10	3.05	-.369	-2.54	+.231	+1.59	-.869	-5.99	-1.238	-8.54	-.638	-4.40
Center	62	18.90	17	5.18	-.717	-4.94	+.498	+3.43	-.500 -.712	-3.45 -4.91	-1.217 -1.429	-8.39 -9.85	-.002 -.214	-0.01 -1.48
Base	0	0	24	7.32	-.665	-4.59	+.410	+2.83	-.500	-3.45	-1.165	-8.03	-.090	-0.62

BLADE FREQUENCY PROGRAM

BLADE MODEL AND OPERATING CONDITION PARAMETERS

WINDMILL TOWER NATURAL FREQUENCIES 500KW,H=123 Truss 7/29/75

INFINITE STIFFNESS IN THE Y DIRECTION
INFINITE STIFFNESS IN THE Z DIRECTION

NUMBER OF SECTIONS = 11
ROTATIONAL SPEED CAP OMEGA = 0.0 RADIANS/SEC
CONTROL ANGLE SI ZERO = 0.0 DEGREES
CONTROL ANGLE THETA C = 0.0 DEGREES
CONTROL OFFSET = 0.0 FEET
RADIAL DISTANCE TO FIRST BLADE ELEMENT = 0.0

I	LENGTH FEET	MASS LB-SEC2 PER-FT	EIX LB-FT2	EIY LB-FT2	EIZ LB-FT2	IX LB-SEC2-FEET	IY LB-SEC2-FEET	PHI DEGREES	EPSILON FEET	OFFSET FEET
1	0.1240D 02	0.2500D 03	-0.2020D 12	0.3200D 12	0.3200D 12	0.5920D 05	0.3030D 05	0.0	0.0	0.0
2	0.1240D 02	0.2400D 03	0.1320D 12	0.4300D 12	0.4300D 12	0.4840D 05	0.2730D 05	0.0	0.0	0.0
3	0.1240D 02	0.2300D 03	0.9300D 11	0.3140D 12	0.3140D 12	0.3960D 05	0.2270D 05	0.0	0.0	0.0
4	0.1240D 02	0.2200D 03	C.8500D 11	0.1650D 12	0.1650D 12	0.3120D 05	0.1840D 05	0.0	0.0	0.0
5	0.1240D 02	0.2100D 03	0.6330D 11	0.2320D 12	0.2320D 12	0.2410D 05	0.1480D 05	0.0	0.0	0.0
6	0.1240D 02	0.2000D 03	0.4200D 11	0.7300D 11	0.7300D 11	0.1800D 05	0.1150D 05	0.0	0.0	0.0
7	0.1240D 02	0.1900D 03	C.2560D 11	0.1130D 12	0.1130D 12	0.1330D 05	0.9100D 04	0.0	0.0	0.0
8	0.1240D 02	0.1800D 03	0.1950D 11	0.1800D 11	0.1800D 11	0.9300D 04	C.7000D 04	0.0	0.0	0.0
9	0.1240D 02	0.1700D 03	0.1110D 11	-0.2990D 11	-0.2990D 11	0.6100D 04	0.5200D 04	0.0	0.0	0.0
10	0.1240D 02	0.1600D 03	C.7500D 10	0.7140D 10	0.7140D 10	0.3800D 04	0.4000D 04	0.0	0.0	0.0
11	0.1000D 00	0.2188D 04	C.1000D 20	0.1000D 20	0.1000D 20	0.5400D 05	0.5400D 05	0.0	0.0	0.0

NOTE: The coordinate system for this program is different from the coordinate system assumed previously for the towers (see Fig. 6-10). Y & Z are the lateral axes for the program and x is the longitudinal axis.

Figure 6-12. WGS Frequency Analysis Program Input.

BLADE FREQUENCY PROGRAM

BLADE MODEL AND OPERATING CONDITION PARAMETERS

WINDMILL TOWER NATURAL FREQUENCIES 1500KW,H=140 Truss 7/18/75

INFINITE STIFFNESS IN THE Y DIRECTION
INFINITE STIFFNESS IN THE Z DIRECTION

NUMBER OF SECTIONS = 11
ROTATIONAL SPEED CAP OMEGA = 0.0 RADIANS/SEC
CONTROL ANGLE SI ZERO = 0.0 DEGREES
CONTROL ANGLE THETA C = 0.0 DEGREES
CONTROL OFFSET = 0.0 FEET
RADIAL DISTANCE TO FIRST BLADE ELEMENT = 0.0

I	LENGTH FEET	MASS LB-SEC2 PER-FT	EIX LB-FT2	EIY LB-FT2	EIZ LB-FT2	IX LB-SEC2-FEET	IY LB-SEC2-FEET	PHI DEGREES	EPSILON FEET	OFFSET FEET
1	0.13900 02	0.37000 03	-0.22400 12	0.30700 12	0.30700 12	0.10300 06	0.57000 05	0.0	0.0	0.0
2	0.13900 02	0.35000 03	0.16400 12	0.36300 13	0.36300 13	0.82000 05	0.47000 05	0.0	0.0	0.0
3	0.13900 02	0.33000 03	0.18100 12	0.46800 12	0.46800 12	0.63000 05	0.37000 05	0.0	0.0	0.0
4	0.13900 02	0.31000 03	0.10200 12	0.45100 12	0.45100 12	0.47000 05	0.28000 05	0.0	0.0	0.0
5	0.13900 02	0.29000 03	0.76000 11	0.40700 12	0.40700 12	0.34000 05	0.22000 05	0.0	0.0	0.0
6	0.13900 02	0.27000 03	0.51000 11	0.22800 12	0.22800 12	0.23000 05	0.16000 05	0.0	0.0	0.0
7	0.13900 02	0.25000 03	0.32000 11	0.13700 12	0.13700 12	0.15000 05	0.12000 05	0.0	0.0	0.0
8	0.13900 02	0.23000 03	0.21000 10	0.90900 11	0.90900 11	0.90000 04	0.80000 04	0.0	0.0	0.0
9	0.13900 02	0.21000 03	0.14000 11	0.50800 11	0.50800 11	0.50000 04	0.60000 04	0.0	0.0	0.0
10	0.13900 02	0.19000 03	0.74000 10	0.30600 11	0.30600 11	0.20000 04	0.40000 04	0.0	0.0	0.0
11	0.10000 00	0.45510 04	0.10000 20	0.10000 20	0.10000 20	0.26000 06	0.26000 06	0.0	0.0	0.0

NOTE: The coordinate system for this program is different from the coordinate system assumed previously for the towers (see Fig. 6-10). Y & Z are the lateral axes for the program and X is the longitudinal axis.

Figure 6-12 (cont.)

BLADE FREQUENCY PROGRAM

BLADE MODEL AND OPERATING CONDITION PARAMETERS

WINDMILL TOWER NATURAL FREQUENCIES 500KW,H=121 Concrete 7/31/75

INFINITE STIFFNESS IN THE Y DIRECTION
INFINITE STIFFNESS IN THE Z DIRECTION

NUMBER OF SECTIONS = 13
ROTATIONAL SPEED CAP OMEGA = 0.0 RADIANS/SEC
CONTROL ANGLE SI ZERO = 0.0 DEGREES
CONTROL ANGLE THETA C = 0.0 DEGREES
CONTROL OFFSET = 0.0 FEET
RADIAL DISTANCE TO FIRST BLADE ELEMENT = 0.0

I	LENGTH FEET	MASS LB-SEC2 PER-FT	EIX LB-FT2	EIY LB-FT2	EIZ LB-FT2	IX LB-SEC2-FEET	IY LB-SEC2-FEET	PHI DEGREES	EPSILON FEET	OFFSET FEET
1	0.10000 02	0.15620 04	0.94700 12	0.11840 13	0.11840 13	0.17820 06	0.10420 06	0.0	0.0	0.0
2	0.10000 02	0.14800 04	0.80500 12	0.10060 13	0.10060 13	0.15140 06	0.89900 05	0.0	0.0	0.0
3	0.10000 02	0.13570 04	0.67700 12	0.84700 12	0.84700 12	0.12740 06	0.77000 05	0.0	0.0	0.0
4	0.10000 02	0.13150 04	0.56400 12	0.70500 12	0.70500 12	0.10610 06	0.65500 05	0.0	0.0	0.0
5	0.10000 02	0.12320 04	0.46400 12	0.58100 12	0.58100 12	0.87300 05	0.55300 05	0.0	0.0	0.0
6	0.10000 02	0.11500 04	0.37700 12	0.47100 12	0.47100 12	0.70900 05	0.46200 05	0.0	0.0	0.0
7	0.10000 02	0.10670 04	0.30200 12	0.37700 12	0.37700 12	0.56700 05	0.33200 05	0.0	0.0	0.0
8	0.10000 02	0.98400 03	0.23700 12	0.29600 12	0.29600 12	0.44500 05	0.31300 05	0.0	0.0	0.0
9	0.10000 02	0.90200 03	0.18200 12	0.22800 12	0.22800 12	0.34200 05	0.25300 05	0.0	0.0	0.0
10	0.10000 02	0.81900 03	0.13600 12	0.17100 12	0.17100 12	0.25700 05	0.20200 05	0.0	0.0	0.0
11	0.10000 02	0.73600 03	0.99000 11	0.12400 12	0.12400 12	0.18600 05	0.15900 05	0.0	0.0	0.0
12	0.11000 02	0.76000 03	0.41500 11	0.98000 09	0.98000 09	0.16900 05	0.16600 05	0.0	0.0	0.0
13	0.10000 00	0.21680 04	0.10000 20	0.10000 20	0.10000 20	0.54000 05	0.54000 05	0.0	0.0	0.0

Note: The coordinate system for this program is different from the coordinate system assumed previously for the towers (see Fig. 6-10). Y & Z are the lateral axes for the program and X is the longitudinal axis.

Figure 6-12 (cont.)

BLADE FREQUENCY PROGRAM

BLADE MODEL AND OPERATING CONDITION PARAMETERS

WINDMILL TOWER NATURAL FREQUENCIES 1500KW,H=140 Concrete 7/12/75

INFINITE STIFFNESS IN THE Y DIRECTION
INFINITE STIFFNESS IN THE Z DIRECTION

NUMBER OF SECTIONS = 15
ROTATIONAL SPEED CAP OMEGA = 0.0 RADIANS/SEC
CONTROL ANGLE SI ZERO = 0.0 DEGREES
CONTROL ANGLE THETA C = 0.0 DEGREES
CONTROL OFFSET = 0.0 FEET
RADIAL DISTANCE TO FIRST BLADE ELEMENT = 0.0

I	LENGTH FEET	MASS LB-SEC2 PER-FT	EIX LB-FT2	EIY LB-FT2	EIZ LB-FT2	IX LB-SEC2-FEET	IY LB-SEC2-FEET	PHI DEGREES	EPSILON FEET	OFFSET FEET
1	0.10000 02	0.1681D 04	0.1179D 13	0.1473D 13	0.1473D 13	0.2217D 06	0.1273D 06	0.0	0.0	0.0
2	0.10000 02	0.1603D 04	0.1023D 13	0.1299D 13	0.1299D 13	0.1924D 06	0.1118D 06	0.0	0.0	0.0
3	0.10000 02	0.1526D 04	0.8810D 12	0.1102D 13	0.1102D 13	0.1658D 06	0.9760D 05	0.0	0.0	0.0
4	0.10000 02	0.1448D 04	0.7540D 12	0.9420D 12	0.9420D 12	0.1417D 06	0.8470D 05	0.0	0.0	0.0
5	0.10000 02	0.1370D 04	0.6390D 12	0.7990D 12	0.7990D 12	0.1201D 06	0.7310D 05	0.0	0.0	0.0
6	0.10000 02	0.1293D 04	0.5360D 12	0.6700D 12	0.6700D 12	0.1009D 06	0.6270D 05	0.0	0.0	0.0
7	0.10000 02	0.1215D 04	0.4450D 12	0.5570D 12	0.5570D 12	0.8360D 05	0.5330D 05	0.0	0.0	0.0
8	0.10000 02	0.1138D 04	0.3650D 12	0.4570D 12	0.4570D 12	0.6870D 05	0.4500D 05	0.0	0.0	0.0
9	0.10000 02	0.1060D 04	0.2960D 12	0.3690D 12	0.3690D 12	0.5560D 05	0.3760D 05	0.0	0.0	0.0
10	0.10000 02	0.9820D 03	0.2350D 12	0.2940D 12	0.2940D 12	0.4430D 05	0.3120D 05	0.0	0.0	0.0
11	0.10000 02	0.9050D 03	0.1840D 12	0.2300D 12	0.2300D 12	0.3460D 05	0.2550D 05	0.0	0.0	0.0
12	0.10000 02	0.8270D 03	0.1400D 12	0.1760D 12	0.1760D 12	0.2640D 05	0.2070D 05	0.0	0.0	0.0
13	0.10000 02	0.7500D 03	0.1050D 12	0.1310D 12	0.1310D 12	0.1970D 05	0.1660D 05	0.0	0.0	0.0
14	0.10000 02	0.6720D 03	0.4000D 11	0.5800D 09	0.5800D 09	0.1420D 05	0.1310D 05	0.0	0.0	0.0
15	0.10000 00	0.4551D 04	0.1000D 20	0.1000D 20	0.1000D 20	0.2600D 06	0.2600D 06	0.0	0.0	0.0

NOTE: The coordinate system for this program is different from the coordinate system assumed previously for the towers (see Fig. 6-10). Y & Z are the lateral axes for the program and X is the longitudinal axis.

Figure 6-12 (cont.)

TABLE 6-19. RESULTS OF TOWER VIBRATION ANALYSIS

		NATURAL FREQUENCY				
		MODE				
		1		2	3	4
		RAD/SEC	CYCLES/REV	RAD/SEC	RAD/SEC	RAD/SEC
500 Kw Truss	Torsion	57.4	17	161	287	419
	Bending 1	11.0	3.25	88	279	535
	Bending 2	11.0	3.25	76	138	297
1500 Kw Truss	Torsion	23.9	6.64	113.6	213	307
	Bending 1	8.20	2.28	75.6	233	487
	Bending 2	8.16	2.27	58.4	119	277
500 Kw Shell	Torsion	115.6	32.1	210	358	517
	Bending 1	14.9	4.41	46.2	126.8	305
	Bending 2	14.5	4.29	32.8	84.1	145
1500 Kw Shell	Torsion	68.7	19.1	162	303	446
	Bending 1	9.98	2.77	35.2	98.6	235
	Bending 2	9.16	2.55	17.0	54.9	111

NOTES: Bending 1 is bending about an axis parallel to the rotor shaft.

Bending 2 is bending about an axis perpendicular to the rotor shaft.

The difference is due to the greater mass moment of inertia of the turntable assembly perpendicular to the rotor shaft.

The fundamental natural frequencies in torsion and bending for all four towers are above 1.5 times the operating frequency in bending and 2.5 times the operating frequency in torsion. In addition, all the fundamental frequencies avoid even multiples of the operating frequency. Therefore, substantial safety margins exist for all vibratory excitations of the tower designs.

6.5.2.5 Turntable Design

The turntable structure sits between the tower top and the rotor hub. Its function is to support and orient the rotor and provide support for the main gearbox, generator, nacelle, miscellaneous accessory equipment and the yaw control mechanism. The lower base of the turntable fastens to the upper race of the x-roller bearing which is mounted on top of the tower.

The turntable structure is a weldment made from A36 structural grade steel. The main features, as shown in Figure 6-13, include the box structure supporting the rotor hub, the end plate structure supporting the box structure which distributes rotor loads from the box structure to the x-roller bearing support structure, and the side rails which provide bending and torsional strength for reacting loads imposed by the main gearbox, generators, accessories and the nacelle. The upper and lower plates which fasten to the side rails are not shown on the sketch for clarity. Emphasis was placed on providing short, direct and stiff load paths between both the rotor hub support point and main gearbox support and the x-roller bearing support structure.

6.5.2.6 Structural Weight and Cost Summary

Structural weight and costs summary data are presented for the 500 KW and 1500 KW WGS towers in Tables 6-20 and 6-21, respectively. Due to the more rigorous analyses conducted in the preliminary design phase, the weight of the steel truss tower is higher than that originally estimated in the conceptual design and optimization phases. However, the cost prediction for the tower is very close to the original forecast.

6.6 Conclusions and Recommendations

6.6.1 Conclusions

Results of the program studies have shown that the steel truss tower, for both the 500 KW and 1500 KW WGS, is the most economical tower design. The pre-cast, post-tensioned concrete shell tower, however, is competitive for multi-unit production and has the advantage of more acceptable visual appearance. Both tower types meet the selected strength and stiffness criteria. Both the steel shell and reinforced concrete towers are more costly for multi-unit production while the reinforced concrete tower has the disadvantage of developing cracks caused by curing or tensile stresses.

The three foundation types are all possible selections for the truss and shell towers for both the 500 KW and 1500 KW WGS. Local soil conditions at the site will determine the most desirable foundation.

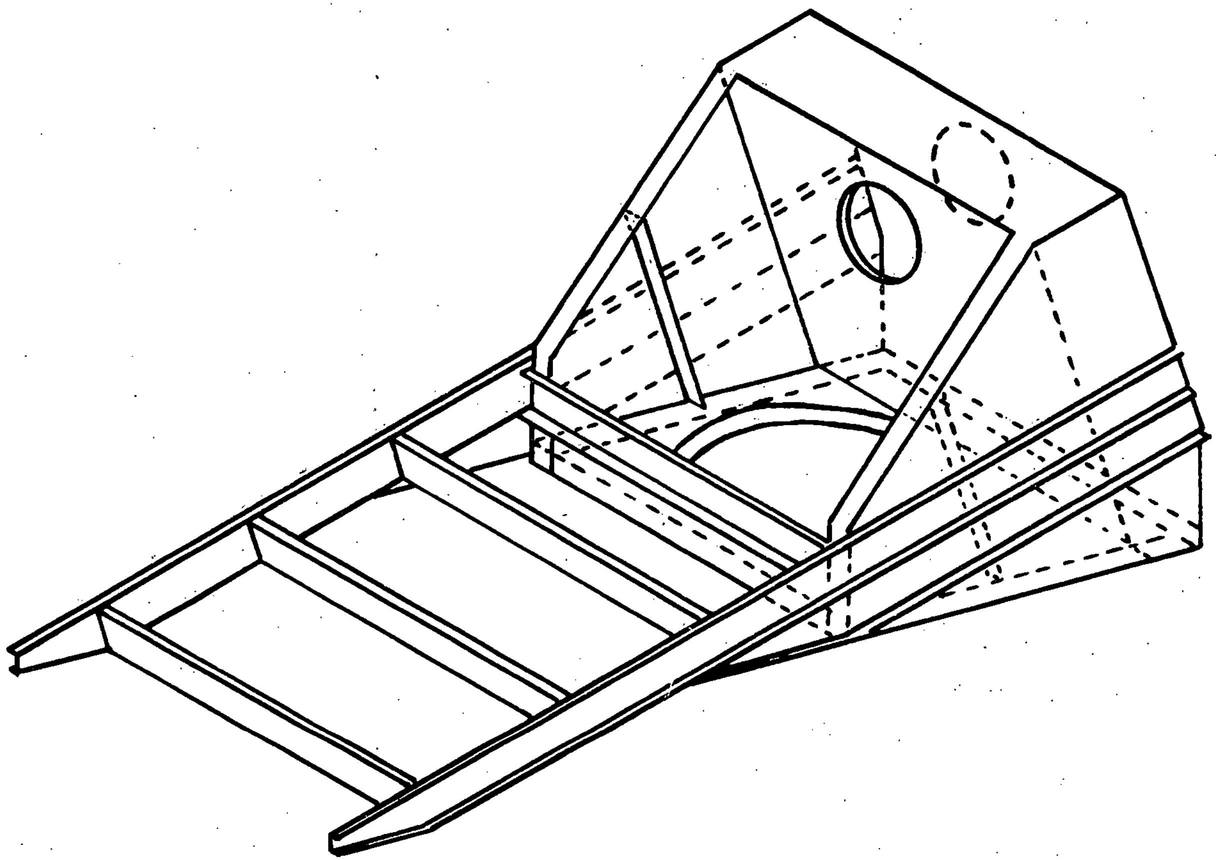


Figure 6-13. Basic Structure of Turntable

TABLE 6-20. STRUCTURAL SYSTEM WEIGHT SUMMARY, MG (KIPS)

	TRUSS		CONCRETE SHELL	
	500 kW	1500 kW	500 kW	1500 kW
TRUSS TOWER				
Chords	8.0 (17.6)	16.4 (36.2)	-	-
Diagonal Bracing	14.2 (31.3)	15.9 (35.1)	-	-
Horizontal Bracing	5.0 (11.0)	5.6 (12.3)	-	-
Gussets	2.7 (6.0)	3.8 (8.3)	-	-
Bearing Support Structure	6.9 (15.2)	9.7 (21.3)	-	-
Base Support	.3 (.7)	.4 (.9)	-	-
Miscellaneous	.3 (.7)	.5 (1.1)	-	-
	37.5 (82.7)	52.2 (115.1)	200.5 (442.0)	237.2 (523.0)
PRE-CAST CONCRETE TOWER				
Common Structure				
Turntable	4.7 (10.4)	13.1 (28.9)	4.7 (10.4)	13.1 (28.9)
Walkways and Steps	1.3 (2.9)	1.8 (4.0)	1.3 (2.9)	1.8 (4.0)
Nacelle	1.3 (2.9)	2.0 (4.4)	1.3 (2.9)	2.0 (4.4)
Lighting	.2 (.4)	.2 (.4)	.2 (.4)	.2 (.4)
	7.5 (16.5)	17.1 (37.7)	7.5 (16.5)	17.1 (37.7)
TOTAL STRUCTURE	45.0 (99.2)	69.4 (153.0)	208.0 (458.6)	254.4 (560.7)

TABLE 6-21. STRUCTURAL SYSTEM COST SUMMARY - DOLLARS

(1000 Unit Basis)

Items	Truss		Precase - Concrete	
	500 Kw	1500 Kw	500 Kw	1500 Kw
Structural Steel	39000	55000	-	-
Precast Concrete Shell	-	-	49800	60600
Turntable & Enclosure	13900	30000	13900	30000
Miscellaneous Equipment (Ladders, Walkways, Lighting)	19000	21000	19000	21000
TOTAL STRUCTURAL SYSTEM	71900	106000	82700	111600
Foundation				
Pile, Rock Anchor	23800	26000	38900	48000
Pile	24500	35000	33300	48000
Mass Concrete	28900	47000	23500	52000

6.6.2 Recommendations

Because of the cost and adaptability to modifications required for instrumentation and possible strengthening and stiffening, the truss tower is recommended for the experimental system. For the production installation, the pre-cast, post-tensioned concrete shell is recommended because of the small cost penalty required for the large gain in visual acceptability.

7.0 DRIVE SUBSYSTEM

The Drive Subsystem consists primarily of the machinery train which transmits the rotor torque to the electrical generator at the proper rotational speed. As defined in this study, it also includes the control system power supply, the mechanical portions of the rotor controls and the turntable orientation mechanism. The electronic and electrical portions of the control system, and the control system design concepts were presented in Section 5, Controls Subsystem.

7.1 Requirements

The Drive Subsystem is made up of the components which support the rotor, transmit its useful power to the generator, and provide actuation of turntable and rotor controls. A list of the prime functions of the Drive Subsystem, and typical components which provide these functions, is given in Table 7-1.

Meeting these functional requirements results in major design parameters dictated by the optimized integration of this subsystem and other subsystems of the WGS. A list of these parameters is given in Table 7-2, with typical numerical values representative of the high power and low power WGS units.

7.2 Design Approach

In order to minimize cost, maximize reliability and provide long drive train component life, an approach emphasizing simplicity and design ruggedness was used. All Drive Subsystem concepts considered only off-the-shelf components operating well within their strength and performance capacities. Safe failure modes were designed into the Drive Subsystem where possible; e.g., feathering the blades and locking the turntable in the case of servo system failure.

Simple field assembly tasks were assured by assigning critical fitting tasks to shop fabrication steps and by allowing latitude for misalignment between field-assembled units. Maintainability was emphasized by the simple design approach, by designing good accessibility provisions and by including strategically placed diagnostic sensors.

In short, the entire design approach to the Drive Subsystem was geared to achieve low initial and operating costs through the use of rugged components and straightforward, proven component integration techniques.

7.3 Available Components and Technology

7.3.1 Main Drive Line

Off-the-shelf speed changers, bearings, couplings, and auxiliary equipment are available to perform all of the primary functions required in the main drive train for both the low power and high power WGS units. In some instances, components of varied design are available. Table 7-3 summarizes the characteristics of available components applicable to the WGS.

Design and analytical techniques and the large body of existing usage experience with these components can provide an accurate prediction of life and

TABLE 7-1. PRIME FUNCTIONS OF THE DRIVE SUBSYSTEM COMPONENTS

COMPONENT	FUNCTION
Rotor Spindle or Shaft	Supports the rotor, allows its rotation
Input Drive	Transmits rotor torque to the gearbox
Gearbox	Provides rotor to generator speed conversion
	Drives auxiliary equipment
Parking Brake	Stops the rotor from low rpm and locks it
Inching Drive	Repositions the parked rotor
Pitch Control	Controls pitch setting of rotor blades
Turntable Control	Orients rotor into the wind
Turntable Bearing	Provides pivot between turntable and tower
Hydraulic System	Provides power to pitch and orientation controls

TABLE 7-2. PRINCIPAL DRIVE SUBSYSTEM DESIGN PARAMETERS

POWER		500 KW	1500 KW
Input	Speed	32.3 RPM	34.4 RPM
	Torque	165 kNm (122 ft KIPS)	461 kNm (340 ft KIPS)
Output	Speed	1800 RPM	1800 RPM
	Torque	3 kNm (2.2 ft KIPS)	8.8 kNm (6.5 ft KIPS)
ROTOR LOADS			
Thrust		64 kN (14.3 KIPS)	149 kN (33.5 KIPS)
Weight		107 kN (24 KIPS)	218 kN (49 KIPS)
Cyclic Torque (Rated)		± 41 kNm (± 30 ft KIPS)	± 114 kNm (± 84 ft KIPS)
Gyroscopic Moment (1/3 RPM)		45 kNm (33 ft KIPS)	108 kNm (80 ft KIPS)
TURNTABLE LOADS			
Normal Operation		115 kNm (85 ft KIPS)	244 kNm (180 ft KIPS)
Maximum Load		576 kNm (425 ft KIPS)	949 kNm (700 ft KIPS)
LIFE-DYNAMIC COMPONENTS		30 years at 71% operating fraction	
TEMPERATURE		- 51°C (- 60°F) to + 49°C (+ 120°F)	
OTHER ENVIRONMENT		Rain, Hail, Dust, Lightning	
PITCH RATE		5° per second Normal	
		15° per second Emergency Feathering	
TURNTABLE REORIENTATION RATE		1/3 RPM	

TABLE 7-3. COMPONENT AND TECHNOLOGY AVAILABILITY

COMPONENT	LIMITATIONS	AVAILABILITY	REMARKS
Gearbox			
Fixed Ratio Gearbox	Suited to full range	Fully catalogued	Well developed, good quality
Belt Drive	6500 ft-lb torque limit	Fully catalogued	Not durable or efficient
Chain Drive	30000 ft-lb torque limit	Fully catalogued	Not as efficient as gears
Hydrostatic	2000 ft-lb torque limit	Special	Inefficient
Bearings			
Ball	Lowest load capacity	Fully catalogued	Lowest friction
Straight Roller	No axial load capacity	Fully catalogued	Low friction
Tapered Roller	Close tolerance mounting	Fully catalogued	Provides stiffness
Spherical Roller	High cost, modest capacity	Fully catalogued	Tolerates deflection
Crossed Roller	Stiff backup required	Fully catalogued	Available with ring gear raceway
Flexible Couplings			
Gear	Periodic maintenance required	Fully catalogued	
Flex Plate	Limited flexibility	Fully catalogued	
Brake			
Caliper/Disc/Hydr.	15000 ft-lb	Catalogued	Fail released
Caliper/Disc/Spring	7500 ft-lb	Catalogued	Automatic fail-on
Multiple Disc Wet	10000 ft-lb	Catalogued	Repeatable torque, durable
Miscellaneous			
Auxiliary Drives			
Worm Gear	Inefficient	Catalogued	Can be irreversible
Hydraulic Motors		Catalogued	High power density
Hydraulic Cylinders	Column strength limit	Catalogued	High power density

efficiency if the operating parameters are well defined. Fabrication practices are well enough advanced and quality sufficiently maintained in practice to assure consistent results from component to component.

The speed change units, which are available in the required power regime and somewhat beyond, are fixed ratio, parallel shaft geared units. Belt drive, chain drive and field coupled units are not available to handle power levels over a few hundred kilowatts.

Optional lubrication systems offered with these gearboxes will satisfactorily control oil temperatures over a moderate temperature range, but for extremely low temperatures, modifications would be required. Startup from sub-zero (Fahrenheit) temperatures would require a heated starting oil supply, together with control system software modifications which would permit low power level operation for some warm-up period. The hardware and technology required to design and build these modifications is available from several industries.

Bearings required for rotor blade and hub support for the drive line and for the turntable are available in ball, straight roller, tapered roller, and spherical roller configurations. Computer programs to predict bearing life and performance are adaptable to any of these bearing types. The matter of choice of bearing configuration can be resolved in most cases on the basis of installed system cost.

7.3.2 Other Mechanical Components

All of the other mechanical components required for the Drive Subsystem are available as off-the-shelf articles. These components, using the preliminary design system for example, include the wet clutch used in the parking brake, the worm and gear of the inching drive and the worm gearbox for turntable orientation. In some cases, adaptation of the hardware is required for efficient packaging, but this does not affect performance or durability.

Consideration was given to use of a brake of sufficient capacity to control and stop the rotor in very high winds. The technology to perform this task is not lacking, but the energy absorption and torque capacity needed are well beyond normal braking requirements. A primary brake, capable of stopping the rotor from an overspeed condition (due to extreme gusts, network disconnect, etc.), was found to be impractical for the WGS. Brake size becomes unreasonably large when installed directly on the low speed rotor shaft, while gearbox strength and brake energy dissipation capability come into question with the brake installed on the generator high speed shaft. This problem was circumvented by providing a direct, fail-safe mechanical method for slowing the rotor by blade feathering, actuated by rotor inertia when overspeed exceeds safe limits. After slowing, the stop is completed with a small parking brake.

It was recognized in the conceptual phase that some type of clutch and/or slippage device might be required between the rotor and the generator to limit torque or to augment system stability during wind gusts. This potential need could not be confirmed until the preliminary design had reached the point where the results of the stability analysis and the utility interface requirements were known. The use of clutches and/or slippage devices was not considered desirable from a Drive Subsystem standpoint, since they would add extra complexity to the drive train and potentially reduce system efficiency. Therefore, until a specific need for such a device has been established, it was decided not to include a slippage device in the design. A tabulation of the available devices, with some of their salient features is given in Table 7-4. Even if one of the more sophisticated slippage devices is not required, it might still be desirable to provide a simple shear pin or weak link to protect expensive components in the event of a mechanical jam.

7.3.3 Hydraulic Components

For the selected preliminary designs, normal control system functions are handled hydraulically, and an abundance of catalogue hardware is available for system build-up and for control actuators. Good reliability and long life can be predicted for these components by using them well below their rated pressure and by maintaining fluid purity and cleanliness. Hydraulic seals are presently available which suffer no apparent aging damage, so that seal life of at least ten to fifteen years and perhaps longer should be achievable. Demand type pumps can be provided to minimize power wastage and fluid wear.

7.4 Design Concept Selections

Because of the abundance of candidate components and the variety of possible arrangements for the WGS, the preferred Drive Subsystem configuration selection process consisted of several steps leading to a general concept selection, followed by a narrower selection of individual components suited to the specific subsystem design requirements.

7.4.1 Concept Selection Criteria

The primary consideration in selecting the Drive Subsystem concept was cost. Factors affecting this cost are power capacity, efficiency, weight, and durability and maintenance requirements. Comparisons of candidate concepts were made on the basis of these factors in the conceptual design phase leading to a common concept for both low and high powered systems. Specific selections of components and arrangements were made after system optimization, based on comparisons guided by the following criteria:

1. Commercial Availability
2. Simplicity and Ruggedness
3. Safe Failure Modes
4. Ease of Maintenance
5. Minimum Assembly Requirements

7.4.2 Alternative Configurations

A variety of Drive Subsystem candidates were evaluated for the WGS application. A summary of the variants considered is given in Table 7-5. A discussion of the

TABLE 7-4. SALIENT FEATURES OF CLUTCHES AND SLIPPAGE DEVICES

	ELECTRIC EDDY CURRENT CLUTCH	MAGNETIC PARTICLE CLUTCH	HYDRAULIC SLIP COUPLING	MECHANICAL PLATE OR DRUM CLUTCH	SHEAR PIN[1]	MULTIPLE V BELTS[2]
Torque limit repeatability (in expected environment)	Excellent	Excellent	Good	Fair	Poor	Poor
Transient behavior	Excellent	Excellent	Excellent	Fair	Excellent	Poor
Efficiency (full load)	94 - 97%	99%	94 - 97%	100%	100%	96 - 98%
Reliability	Excellent	Good	Excellent	Good	Excellent	Fair
Life/Maintenance	Excellent	Good	Good	Good	Good	Fair
Maximum Power	250 KW	50 KW	1500 KW		Unlimited	1500 KW
Energy Absorption Capability	Low	Low	Low	Moderate	None	Moderate
Weight/KW	Moderate	High	Low	Low	Very Low	Low
Cost/KW	Moderate	High	Low	Low	Very Low	Low

NOTES:
1. Shear pin can only be used as ultimate protective device to prevent secondary damage to other components in the event of mechanical jams or impact loads. Features for shear pin are given assuming system is designed so pin does not break under normal operating transient loads.
2. Unequal load sharing between belts and belt stretch cause slip characteristics to be unpredictable for V belt drive.

TABLE 7-5. DRIVE SYSTEM CANDIDATES

COMPONENT	CANDIDATES
Transmission	Belt Drive
	Chain Drive
	Hydrostatic Drive
	Fixed Ratio Gearbox
	Combinations
Rotor Support	Overhung Rotating Shaft
	Fixed Spindle and Quill Shaft
Pitch Control	Mechanical (Rotor Powered)
	Electro Mechanical
	Hydraulic (Motor or Cylinder)
	Push Pull Linkage
	Bevel Gear Drive
	Rack and Pinion Drive
Orientation Control	Wind-powered (Self-powered)
	Electric
	Hydraulic
Turntable Mounting	Multiple Bearing Stack
	Single Combination Bearing
Parking Brake	Brake Input Shaft
	Brake Output Shaft
	Full Energy Brake
	Brake for Parking
Inching Drive	Drive thru Clutch
	Drive thru Brake

major features of these candidates, their advantages and limitations, and the factors leading to the selection of the preferred concept is given below.

Transmission - In the conceptual design phase, consideration was given to the use of variable ratio gearboxes driving the generator. This presented one method for using a variable speed rotor to drive a constant speed generator. The principal difficulty in this approach was a lack of hardware of suitable size.

Rotor torque for the low and high power systems is greater than 135 kNm (100 ft KIPS). Hydrostatic drives, which offer the highest torque capacity for variable speed transmissions, are limited to 2.7 kNm (2000 ft lb). Multi-ratio gearboxes are limited to 23.8 kNm (17,600 ft lb). Belt and chain drives are torque-limited to 8.8 kNm (6500 ft lbs), and 40.7 kNm (30,000 ft lbs), respectively. Thus, it is apparent that all these devices fall far short of the torque capacity required for the WGS first stage. An added disadvantage to all but the multi-ratio gearbox is inferior efficiency ranging from 65% to 85%. This compares to 96% for a gearbox.

Thus, only fixed ratio gearboxes are capable of operation at torque levels typical of the WGS rotor output. Gearboxes are available which cover a wide range of requirements from well below to well above the WGS torque range. Suitable gear ratios are available, and efficiency is superior to all other speed change systems.

Since the first stage of any WGS speed conversion system must be a gearbox, economy and reliability dictate completion of all speed conversion in a single component. It was, therefore, decided to accomplish speed conversion in a triple mesh gearbox.

Rotor Support - A stationary support spindle was shown to be superior when compared to an overhung rotor shaft on pillow blocks. Bearing loads are much lower and the bearings required are lighter and less expensive when mounted in the hub. The support shaft, when rotating, experiences gyroscopic, weight and wind shear loads, reversing once each revolution, whereas these are non-reversing static loads in the spindle. This dictates a heavier shaft design to handle the higher fatigue stress content in the rotating shaft design.

Pitch Control - Of the various pitch control systems studied, the push-pull linkage was judged superior in simplicity, ruggedness, ease of maintenance and cost. Other schemes, which were examined, included a push-pull rack and pinion system and a barrel cam, bevel gear system. Each of these alternatives contains more parts, introduces backlash problems and is difficult to assemble and inspect.

Comparison of various pitch control actuation systems is complex since the method of employment of each can affect reliability, complexity, control accuracy, control response rate, efficiency and cost. However, the principal factors which determined the choice of actuation concept were evaluated and are discussed below.

Rotor-powered, mechanically-actuated control systems have the ability to change pitch in the absence of outside power sources and with minimal power usage during quiescent periods. Large amounts of control power can be brought to bear, when needed, but if not required, excessive power is not dissipated. Reliability, durability and complexity of pure mechanical systems is very dependent on the actuation mechanism employed, which will depend, in turn, on the type of control needed. Unless a bang-bang or step change system can be tolerated, continuously slipping clutches must be used which will waste power and be subject to wear.

Hydraulically-powered control systems offer the greatest abundance of existing hardware and experience for solution of position control servo problems. While much of the hardware is complex and sophisticated, development work has brought about good reliability. Servo system characteristics can be readily tailored, as needed, to provide control stability. Power usage in a hydraulic control system suffers from some inefficiency. While it is possible to use a demand system which pumps fluid only as required to change pitch, the pressure output is dictated by the highest expected control load plus system pressure drop. When control loads are light, this pressure dissipates.

Electrically-powered control systems offer the advantage of availability of stored power in batteries. Actuation hardware is available, but generally requires considerably more space than equivalent capacity hydraulic actuators. Continuous operation, near stall, can be destructive to electrical equipment, and control of overrunning loads is difficult.

Pneumatically-powered control systems have many advantages but, for the WGS application, also have two serious disadvantages; low stiffness and freezing problems at low temperatures.

Orientation Control - Similar considerations apply to the turntable orientation servo system as to the blade pitch control system. An additional concept, considered for this application, was a side wheel wind-powered system which offered the advantage of simple, automatic orientation control with no outside power requirement. Disadvantages were the possibility of sidewheel icing (since the rotor would not always be turning) and lack of control stiffness. An additional difficulty with this type of system is its inability to occasionally make several turntable revolutions in one direction for cable untwist (cable twist control is discussed in Section 8).

Two turntable support configurations were considered. In one configuration, a spindle projecting downward from the turntable would mount a pair of bearings to carry the moment loads from turntable to tower, and thrust would either be handled by a third bearing or by one of the pair. This approach offers good stiffness, but bearing cost is high and installation difficult. The second approach uses one large combination bearing, probably of crossed roller configuration. Cost of the single large bearing is slightly higher than the cost of the three bearing stack, but installation is much simpler and cheaper. Strength and stiffness are adequate, and the large bearing permits an access passageway from inside the tower to the turntable. Stiff backup structure is required to develop the full strength of these bearings, but both turntable and tower top are characterized by heavy steel structure in this area anyway.

Parking Brake - In the event that the pitch control system fails to feather the rotor in response to an overspeed condition, an emergency braking or feathering system is required. Both approaches were evaluated and the emergency feathering system was judged to be preferable from the standpoint of cost and complexity.

The hypothesized emergency situation supposes that a severe gust has occurred, increasing rotor torque 2-1/2 times beyond rated torque. At that point, the generator will trip off the line, allowing all rotor torque to be available for rotor acceleration. Failure to change blade pitch control towards feather through the normal pitch control mechanism would presumably be discovered at that moment. Conceivably, the gust might continue to build, perhaps to 3 or 4 times normal rated power momentarily, requiring a brake torque equivalent to that level merely to prevent overspeed.

In order to cause positive deceleration of the rotor at 2-1/2 times rated torque, brake torque would have to exceed this with some positive margin (3-1/2 times rated torque); otherwise the brake will quickly overheat and become ineffective and a potential fire hazard. The total energy which the brake must absorb is calculated to exceed 1.1×10^6 kg-meters (8.4×10^6 ft-lb) and 4.2×10^5 kg-meters (3.1×10^6 ft-lb) for the 1500 KW and 500 KW systems, respectively. Typical brake disc energy capacity is 20,422 kg-meters/kg (67,000 ft-lb/pound), which would require discs weighing 57 kg (126 lbs) and 21 kg (47 lbs) for the 1500 KW and 500 KW systems, respectively. Brake discs of this size present no apparent problems. However, the largest spring-applied, hydraulically-released caliper available from commercial sources has a maximum load capacity of 4,355 kg (9600 lb). Assuming a 1.52 meter (15 ft) diameter disc, and a 25% margin, 62 calipers and 22 calipers would be required for the 1500 KW and 500 KW systems, respectively. A brake unit of this complexity would be very costly and would require frequent inspection and maintenance.

It was decided, therefore, to opt for the emergency feathering system, utilizing a small parking brake to bring the rotor to a full stop after feathering has slowed it. The emergency feathering system, described in paragraph 7.5.1, is inherently simpler than the brake, employing a mechanical actuating system which can be made highly redundant. The system proposed is believed to be as dependable as, and significantly less costly than, a rotor brake of the required capacity, although a brake could be provided if considered desirable.

7.4.3 Selected Concept for Preliminary Design

After estimating weights, costs and reliability of each of the candidate Drive Subsystem elements, and after reviewing results of the optimization study, the preferred concept for the optimum preliminary design configuration was selected. A summary of these selected elements is given below:

Stationary rotor spindle

Quill shaft drive

Fixed ratio triple mesh gearbox

Multiple wet disk parking brake

Worm gear inching drive

Push-pull pitch link control system

Hydraulic cylinder pitch actuator

Hydraulic motor turntable orientation mechanism

Crossed roller turntable bearing

These elements, then, formed the Drive Subsystem concept embodied in the preliminary design. A clutch or slippage device was not provided in the drive train for the reasons given in paragraphs 8.4.4.1 and 8.5.3.2.

7.5 Drive Subsystem Preliminary Design and Analysis

With the Drive Subsystem concept selected for both the high power and low power WGS units, the preliminary design of the subsystem for the specific power levels (1500 KW and 500 KW, respectively) was initiated. As the design effort progressed, close coordination with the evolving rotor hub design and control system design and analysis was maintained to insure an integrated drive train and control system configuration evolved. The final design and the supporting analyses are presented in this section.

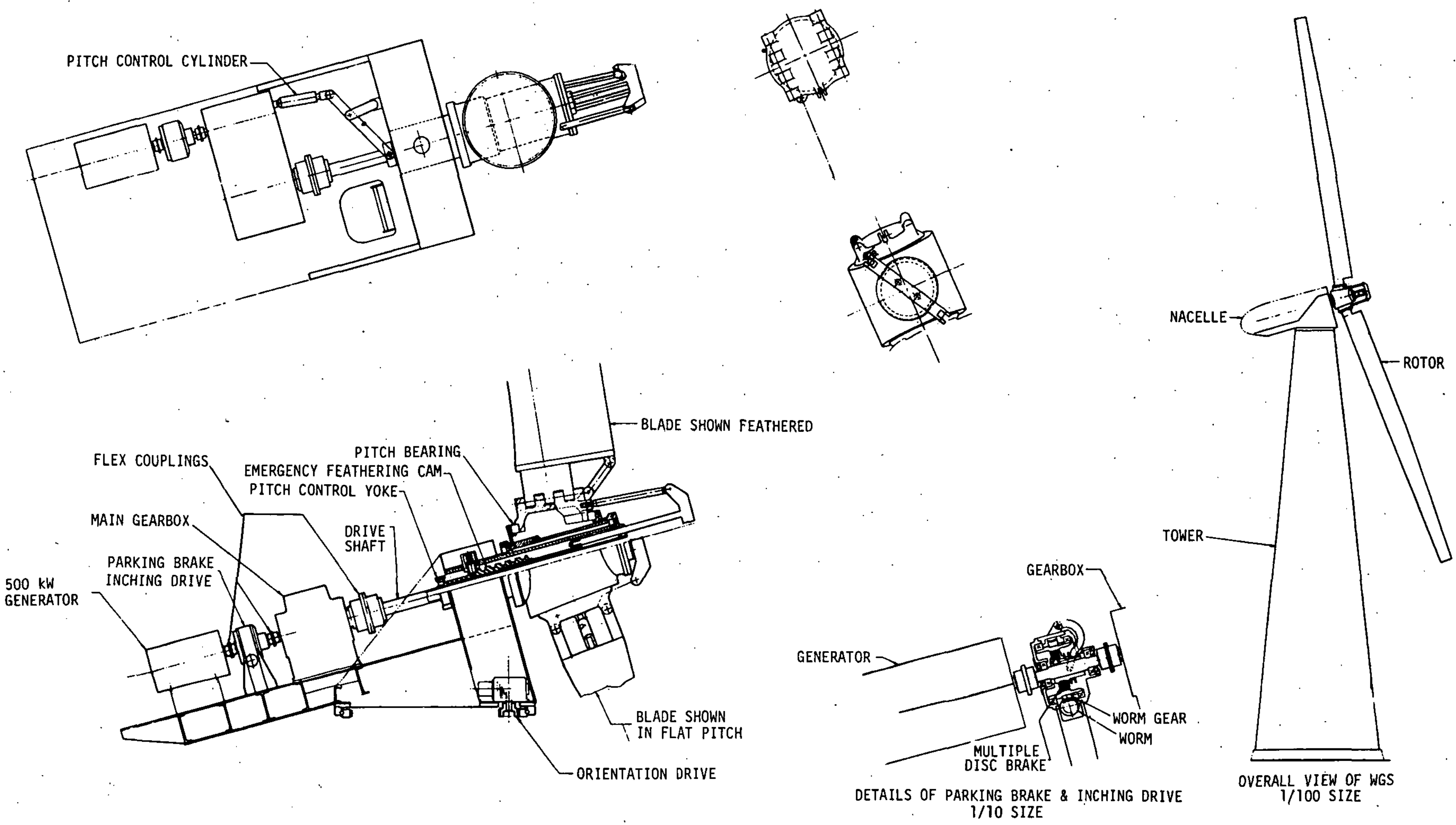


Figure 7.1. Drive Subsystem Components
500 kW WGS

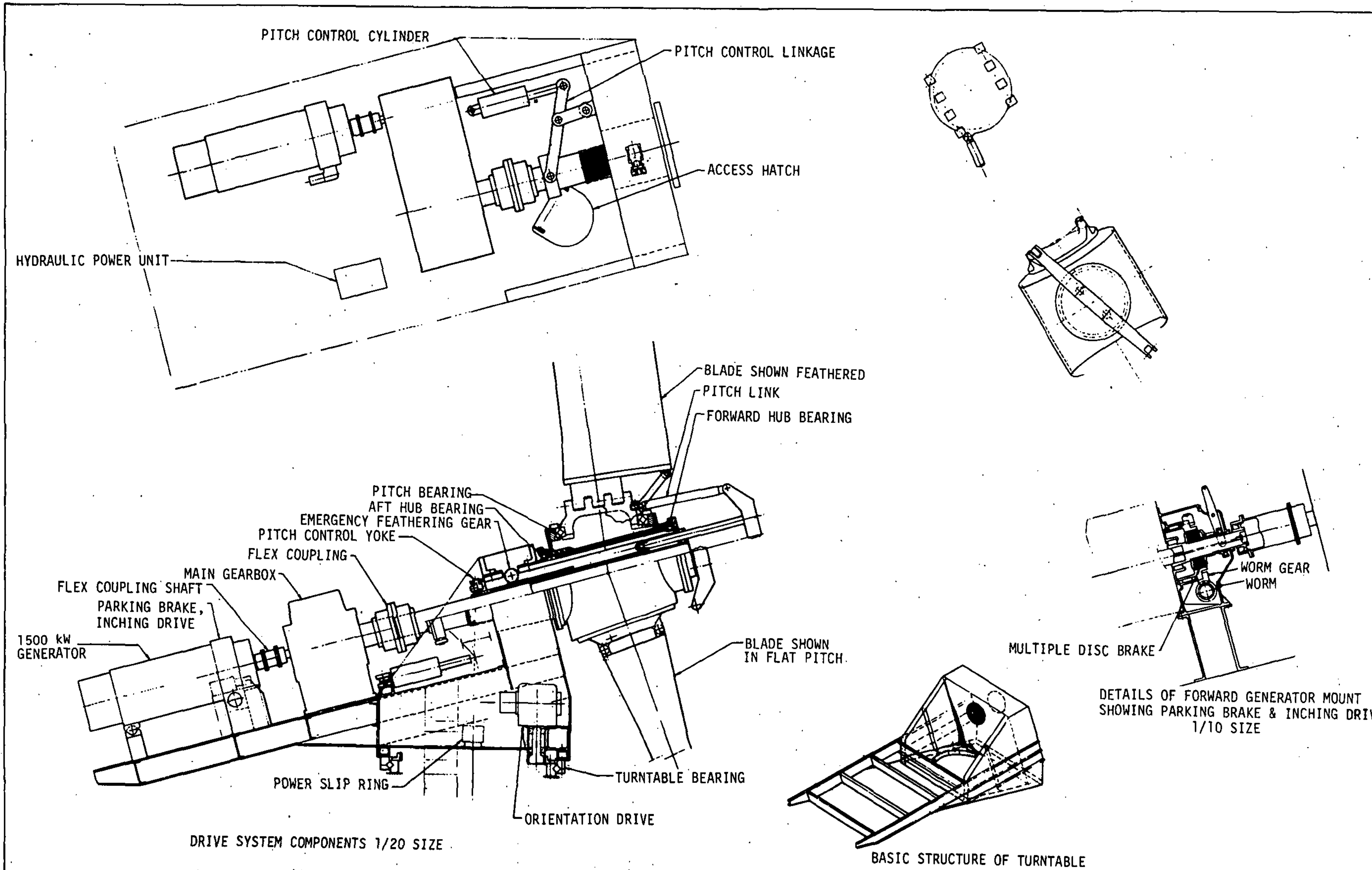


Figure 7.2. Drive Subsystem Components, 1500 kW WGS

7.5.1 Drive Subsystem Preliminary Design

The Drive Subsystems of the 500 KW and 1500 KW WGS preliminary designs are shown in Figures 7-1 and 7-2, respectively. Each of the components is further described below:

Rotor Spindle - The rotor is supported on a tubular spindle which is fitted to a socket in the turntable structure (shown in Figure 7-2). Attachment of the spindle to structure is by means of shear bolts. Stress levels in the spindle are sufficiently low to permit its consideration as permanent structure with a life of 50 years. The spindle provides mounting for a pair of tapered roller bearings which support the rotor hub.

Quill Shaft - The driving connection between the hub and gearbox is a quill shaft which transmits torque only, since all other rotor loads are carried by the stationary spindle. In the 500 KW WGS, this tubular steel shaft is .2 m (8 in.) in diameter with a .9 cm (.35 in) wall. Shaft for the 1500 KW system is .406 m diameter with a .6 cm wall thickness. A crowned gear type flexible coupling connects the quill shaft to the gearbox, allowing for installation misalignment. Candidate couplings are Zurn type F-113 for the 1500 KW WGS and type F-109 for the 500 KW WGS.

Gearbox - Rotor to generator speed conversion is by means of a triple mesh, parallel shaft gearbox. Candidate gearboxes are Philadelphia Gear 18HP3 and 22HP3 models for the 500 KW and 1500 KW WGS, respectively. These gearboxes are equipped with anti-friction bearings throughout, and use case-hardened and shaved or ground gears.

Pitch Control - The rotor pitch control system utilizes a central push-pull control column which is keyed to the rotating quill shaft. A rigid cross beam on the end of this column is linked directly to each of the two blade grip fittings so that the push-pull motion is transformed into rotation of the blades about their pitch axes.

The pitch column can be actuated by two independent means. Normal actuation is by means of a single, large hydraulic cylinder which is linked to a yoke on the inboard end of the column. A thrust bearing in the yoke allows transfer of the linear motion from the cylinder linkage to the rotating column. Motion of the hydraulic cylinder is controlled by an electro-hydraulic servo valve, and cylinder position fed back by a linear displacement transducer.

Emergency actuation of the pitch column is accomplished when cam follower plungers drop into engagement with a helical cam slot in the column. Since a single plunger will actuate the feathering mechanism, multiple, independently acting plungers can be used to provide a high degree of redundancy. Rotor shaft rotation will then move the column, very forcibly, in the feather direction. A deeper annular slot at the end of travel prevents motion beyond feather, even if the shaft continues to rotate, and also prevents inadvertent reversing. The cam follower plungers are spring loaded to the engaged position, and hydraulically disengaged. In the absence of hydraulic pressure, either through malfunction or actuation of a selector valve, the followers will engage and feathering will take place.

Orientation Control - Turntable orientation is accomplished by means of a hydraulic motor driving through a gear train. The motor is mounted on, and drives through, a single stage vertical shaft worm gearbox of 60 to 1 ratio. Output of this gearbox drives a pinion gear which meshes with a large diameter internal gear. This large gear is integral with the inner race of the turntable bearing, which is secured to the tower. Ratio of the pinion-ring gear is 10 to 1.

The worm gearbox is irreversible, so that the turntable will be rigidly held against wind load, unless the wind load is assisting the hydraulic motor. Speed of the motor will be regulated by hydraulic flow control valves, so that turntable rotation will be held to 1/3 rpm , even with assisting wind. This is required to limit rotor gyroscopic forces.

Turntable Bearing - The turntable bearing is a single, large diameter bearing of crossed roller configuration. This type of bearing can withstand radial, thrust and overturning moment loads, separately or simultaneously. Balance of the complete turntable assembly (with all equipment and nacelle) is slightly offset with the rotor on the light side. With wind thrust, there is a constant moment tending to pitch the rotor down. The bearing is sized to handle all loads, including those generated by the maximum wind blowing broadside on the nacelle and feathered blades.

Parking Brake/Inching Drive - A parking brake/inching drive has been designed as a single unit. For this arrangement, the brake connects the generator shaft to a motor driven worm gear, which is the inching drive. The purpose of this arrangement is to eliminate the need for a secondary clutch and actuator, and to bypass a hazardous sequencing step where the brake may be released when the inching drive is engaged. When the parking brake is engaged, it locks the generator shaft to the worm drive, which is irreversible to torque applied to the gear. The worm is driven by a gear motor so that the shaft can be slowly rotated to a selected parking position for stowing or maintenance. At no time is the brake released, until it is desired to start the WGS.

7.5.2 Design Analyses

The preliminary design process led to definition of rotor diameter, solidity, speed, and torque. Estimates of blade weight, inertia and stiffness were also derived and the rotor operating characteristics were further defined. With these values determined, the Drive Subsystem requirements were compared with the capabilities of commercially available hardware, and the most economically suitable parts were chosen. Table 7-2 lists the principal parameters which determine the Drive Subsystem design.

The analysis concentrated on critical design areas, particularly the rotor shaft, torque tube and bearings, pintle or turntable bearings, and orientation drive gear. Analyses of these and the other subsystem components are presented below.

7.5.2.1 Rotor Spindle, Quill Shaft and Bearings

During the conceptual design phase, effort focused on the design of a conventional overhung rotor shaft mounted on pillow blocks. This type of arrangement is illustrated in Figure 7-3 (a). The weight and cost of this arrangement is quite high for several reasons:

1. Since the support bearings are offset from the rotor hub, the moment arm from the support to the rotor weight vector is large, increasing loads in both the shaft and bearings.

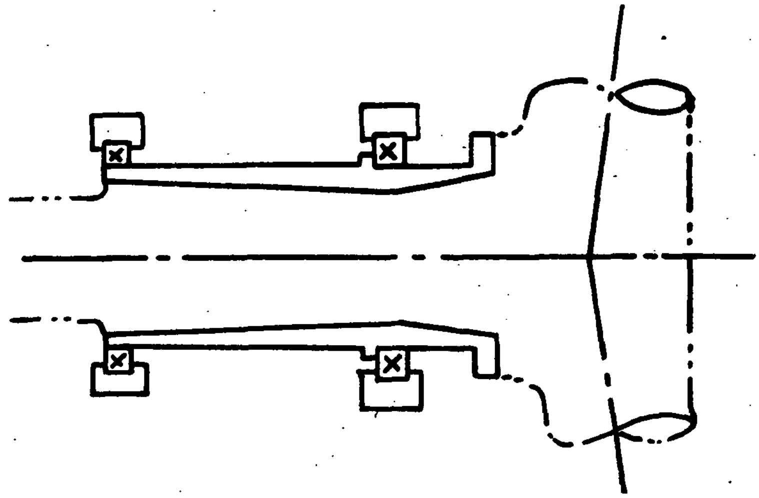
2. In a rotating shaft design, stresses due to weight, wind shear, and gyroscopic moments, are reversing cyclic (fatigue) stresses. Allowable fatigue stress levels are much lower than static stress levels and hence, heavier sections must be used to achieve the required design life.

In an effort to save weight and cost, the design of Figure 7-3 (b) was adopted. This design places the support bearings astride the hub loads, reducing the bearing reaction forces. It also significantly reduces the fatigue content of the spindle stresses. Figures 7-3 and 7-4 compare the weight and cost for each configuration. These estimates include the weight and cost of the shaft, bearings, bearing supports, and the quill shaft of configuration (b).

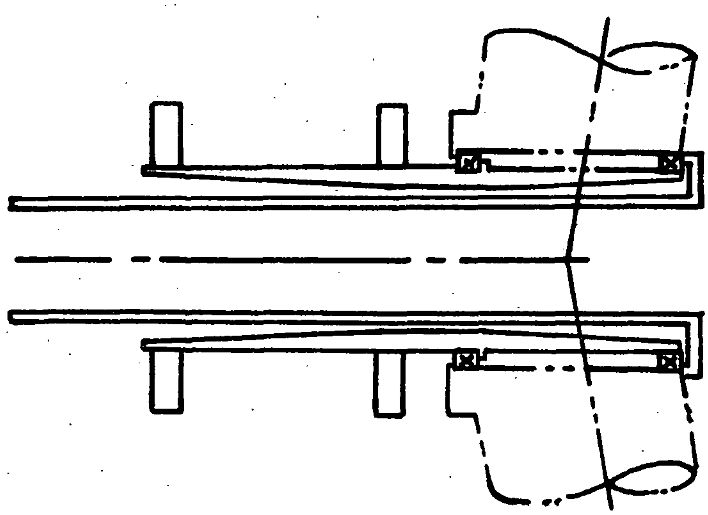
Analysis of the spindle, quill shaft, and hub bearings for the 1500 KW WGS indicated bearing lives of approximately 30 years or longer. Stress levels are 7100 kPa (1030 psi) in the spindle, and 83,430 kPa (12,100 psi) in the quill shaft. In a detail design, a better balance of stress levels and bearing lives could be achieved by readjusting sizes with no significant effect on overall cost or weight.

7.5.2.2 Gearbox.

The WGS gearbox is primarily sized by input (rotor) torque, including both steady and cyclic variations. Steady input torque was determined by power level, and the cyclic component caused by wind variations through the rotor disk was estimated at $\pm$ 25% of the steady. Critical elements in determining



(a)	500 KW	1500 KW
WEIGHT	4180 kg (9200 lb)	8630 kg (19000 lb)
COST	$ 28,175	$ 58,187



(b)	500 KW	1500 KW
WEIGHT	3090 kg (6800 lb)	5000 kg (11000 lb)
COST	$ 14,945	$ 24,090

Figure 7-3. Rotor Shaft Configurations.

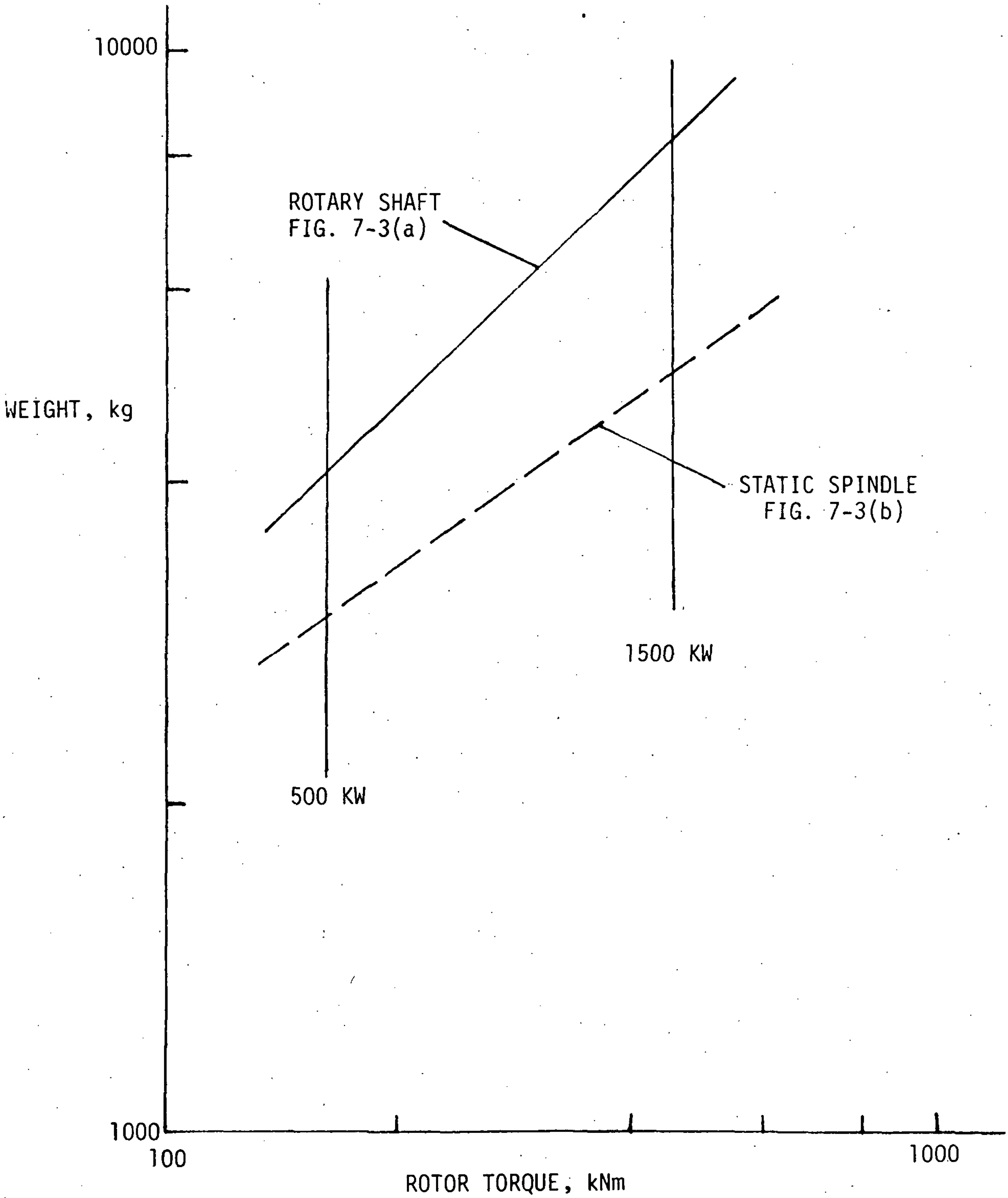


Figure 7-4. Rotor Shaft and Bearing Weight Trends.

gearbox capacity are gear tooth stresses and bearing life. Transmitted power has an effect on the heat generated and, therefore, determines the heat removal requirements for the lubrication system. WGS power levels indicate the need for positive circulation and heat removal. Table 7-6 summarizes the gearbox requirements and the characteristics of the selected gearboxes.

The final design gearbox weight is higher than that estimated in the conceptual design phase. There are two reasons for this:

1. Initially, a service factor close to 1.0 was assumed, which would be adequate if input torque was a very steady value. In view of the final estimate of $\pm$ 25% cyclic torque variations, an increase of service factor to 1.25 - 1.50 was deemed necessary.

2. In addition, it was assumed that quality improvements could be incorporated to upgrade commercial gearbox performance. Actually, significant upgrading can only be achieved through special design and fabrication steps, which were evaluated as not cost effective.

7.5.2.3 Pitch Control

The WGS pitch control system is designed with dual operating modes. A hydraulic cylinder provides a stiff reaction point for blade feedback loads and has sufficient power to handle all anticipated operating loads quickly enough to follow all normal gusts. A mechanical cam provides a very strong and quick system to feather the blades, if normal control must be bypassed for any reason. Events which might require use of the backup system include: loss of hydraulic power; rare gusts, exceeding normal control response capabilities; jammed pitch bearing; and malfunction of the servo system.

Based on results of the Rotor and Control Subsystem analyses, and analyses of the Drive Subsystem internal requirements, the pitch control elements were sized as shown in Table 7-7.

7.5.2.4 Orientation Mechanism

Wind loads will normally assist the orientation drive as it attempts to face the rotor into the wind. Three critical conditions were investigated to determine the orientation mechanism requirements:

1. Static load capacity to withstand 53.6 m/sec (120 mph) winds broadside to the rotor blades

2. Dynamic load capacity to move against friction at 1/3 rpm

3. Static load capacity to hold against rotor cyclic moments.

The candidate motors selected for the 500 KW and 1500 KW orientation mechanisms are the Sundstrand MH-10 and MH-21 hydraulic motors, respectively. These motors

TABLE 7-6. GEARBOX DESIGN DATA

	500 KW	1500 KW
Input RPM	32.3	34.4
Output RPM	1800	1800
Ratio	55.72:1	52.33:1
Transmitted Power (Rated), KW (HP)	560 (751)	1650 (2209)
Service Factor	1.25 to 1.50	1.25 to 1.50
Corrected Power, KW (HP)	700 (938) to 840 (1126)	2060 (2761) to 2475 (3314)
Candidate Gearbox (Philadelphia Gear)	18HP3	22HP3
Mechanical Power Capacity, KW (HP)	810 (1086)	2560 (3424)
Thermal Power Limit, KW (HP)	350 (472)	530 (715)
Gearbox Weight, kg (kips)	9888 (22)	20865 (46)
Oil Capacity, Liters (Gal)	568 (150)	1892 (500)

TABLE 7-7. PITCH CONTROL ELEMENT DATA		
	500 KW	1500 KW
Pitch Travel, degrees	90	90
Pitch Moment, Nm (ft-lb)	12326 (9090)	34438 (25400)
Pitch Rate, Normal, °/sec	5	5
Pitch Rate, Feather, °/sec	15	15
Pitch Column Travel, m (ft)	.91 (3)	1.14 (3.75)
Cylinder Stroke, m (ft)	.46 (1.5)	.64 (2.1)
Operating Load, N (lb)	42091 (9460)	84524 (19000)
Min. Stall Load, N (lb)	63136 (14190)	126786 (28500)
Cylinder Diameter, m (ft)	.152 (.5)	.254 (.83)
Cylinder Force, N, (lb)	62886 (14130)	146784 (33000)
Cylinder Velocity, m/min (ft/min)	1.53 (5)	2.13 (7)
Flow, liter/min (gpm)	28 (7.4)	108 (28.5)
Cylinder Weight, kg (lb)	36 (79)	141 (310)

provide more than adequate capacity for all requirements. A summary of the orientation mechanism data is given in Table 7-8.

7.5.2.5 Turntable Bearing

In the conceptual design phase of the program, the turntable pivot approach was selected as an axle protruding from the turntable, provided with a triple bearing stack and fitted to a socket atop the tower. That system proved costly, as mentioned before, due primarily to the bearing support structure and the axle itself. It also added the requirement for an external tower catwalk and turntable hatch.

In the preliminary design, crossed roller bearings have been selected for this function. Crossed roller bearings, as turntable bearings, were found satisfactory to meet thrust and moment loads imposed by:

1. Maximum wind (120 mph) conditions

2. Maximum operating conditions with maximum superimposed gusts

3. Steady operating (rated) conditions.

Calculated bearing life was 40 years or more.

Another possibility for the turntable bearing is a very large, single row, 4 point contact ball bearing. Calculations indicated that such a bearing is viable, with a calculated life of 50 years. The ball bearing would be less expensive than the roller bearing and would offer less frictional resistance, but might be more prone to fretting failure during long term operation with the turntable stationary. In the detail design effort, consideration should be given to use of a ball bearing for the turntable support.

7.5.2.6 Parking Brake/Inching Drive

As previously discussed, cost effectiveness considerations led to the decision to restrict the rotor brake task to stopping the rotor from low speed, subsequent to blade feathering. This choice was based on minimization of cost, weight and risk, comparing a backup feathering system with an extremely powerful and sophisticated rotor brake. The brake requirements analyzed, and the characteristics of the brake selected to meet these requirements, are given in Table 7-9.

For the inching drive, a 500 RPM, 1.5 KW (2 HP) motor was selected; 1.5 KW (2HP) being consistent with the wiring capacity furnished for standby power. Different motor speeds are available at this power level so that inching drive torque can be traded off against drive speed. The selected motor yields a .2 rpm drive speed with 26,790 Nm (19,760 ft-lb) available torque. This is based on a 50:1 worm drive of 45% efficiency between the motor and the generator shaft.

TABLE 7-8. ORIENTATION MECHANISM DATA

	500 KW	1500 KW
Operating Torque, NM (ft-lb)	83.5 (61.6)	176 (130)
Motor Torque @ 500 psi, Nm (ft-lb)	92.2 (68)	188 (139)
Maximum Static Load, Nm (ft-lb)	337 (249)	504 (372)
Motor Static Cap, Nm (ft-lb)	737 (544)	1508 (1112)
Gearbox Cap, Nm (ft-lb)	490 (362)	691 (510)
Gearbox Weight, kg (lb)	689 (1520)	1340 (2955)
Gearbox Model	8000 C SV	8000 C SV
Motor Weight, kg (lb)	38.14 (84)	65.80 (145)

TABLE 7-9. BRAKE DATA

	500 KW	1500 KW
Rotor Speed at Engagement, RPM	5	5
Rotor Inertia, kgm^2 (KIP ft^2)	375896 (8912)	881335 (20900)
Residual Torque, Nm (ft KIPS)	687.4* (.5)	2202* (1.6)
Design Brake Torque, Nm (ft KIPS)	3390 (2.5)	8813 (6.5)
Inertia Energy, kgm (ft KIPS)	5249 (38)	7874 (57)
Stopping Time, sec	22.2	21.3
Brake Energy, Nm (ft KIPS)	19702	49144
Candidate Brake (Twin Disc)	Model EH124	Model EH324
Brake Weight, kg (lb)	209 (459.8)	317 (679.4)

*Conservatively assumed at 25% of rated torque.

7.5.3 Drive Subsystem Weight and Cost

Drive Subsystem weight and cost data are given in Tables 7-10 and 7-11 for the 500 KW and 1500 KW preliminary designs. The cost data include subsystem integration and installation costs and are based on a production level of 1000 units of each power class machine.

7.6 Design Adaptability

All of the Drive Subsystem components are available in larger sizes. A gearbox size larger than that selected for the 1500 KW WGS is available, with the capacity of handling 2000 KW. Bearings, brakes, hydraulic motors and cylinders are all available in larger sizes so that a 2000 KW WGS could be designed in the same general arrangement of the preliminary design.

Predicting the upper limit of a WGS capacity, using off-the-shelf commercial components, is complicated by such factors as the need to reduce rotor speed as capacity is increased and the non-linear effects of rotor loads. It does appear that available Drive Subsystem components should be able to cover systems to 5000 KW ratings, and use configurations similar to those for the 500 KW and 1500 KW system designs.

7.7 Conclusions and Recommendations

7.7.1 Conclusions

For both the 500 KW and 1500 KW WGS units, and probably for higher rated units as well, the following principal conclusions were reached in the study:

1. Standard commercial components are available for all Drive Subsystem elements

2. A gear type transmission is the only economically feasible speed increaser available for the WGS application

3. A primary brake, capable of stopping the rotor under all operating and failure conditions, is impractical.

7.7.2 Recommendations

Based on these conclusions, and other results and conclusions previously discussed, the following recommendations are offered:

1. The WGS detail designs should generally follow the concepts embodied in the preliminary designs for the 500 KW and 1500 KW WGS units

2. The WGS detail designs should examine economically competitive bearing concepts for the orientation system support

3. Emergency rotor shutdown provisions should focus on blade pitch control as the primary means of reducing rotor torque and speed to safe levels.

TABLE 7-10. DRIVE SUBSYSTEM WEIGHT

COMPONENT	WEIGHT, kg (lb) 500 KW	WEIGHT, kg (lb) 1500 KW
Rotor Spindle	2210 (4865)	3270 (7201)
Rotor Drive Shaft Assembly	1030 (2277)	2770 (6094)
Gearbox	9900 (21800)	20890 (46000)
Generator Drive Shaft Assembly	680 (1490)	860 (1891)
Pitch Control Actuation Assembly	620 (1362)	1550 (3418)
Turntable Bearing	1080 (2375)	3150 (6947)
Turntable Drive Train	820 (1800)	2060 (4547)
Hydraulic System	480 (1035)	660 (1461)
TOTAL	16810 (37004)	35210 (77559)

TABLE 7-11. DRIVE SUBSYSTEM PRODUCTION COSTS

	COST (000 $)	
	500 KW WGS	1500 KW WGS
Transmission	$ 38.6	$ 96.4
Drive Line Components	16.5	33.4
Parking Drive Components	1.9	7.1
Turntable Bearing and Drive Mechanism	15.1	30.3
Hydraulic Components	2.2	5.2
Subsystem Integration	3.7	8.6
TOTAL	$ 78.0	$181.0

8.0 ELECTRICAL SUBSYSTEM

This section describes the development of the Electrical Subsystem for the WGS. Several types of electrical generation schemes were studied during the concept selection process, including variable shaft RPM and fixed RPM schemes. Two fixed RPM concepts, using either an induction or synchronous generator were selected for the WGS preliminary design since these concepts provided the highest overall system efficiency with the lowest overall cost and complexity. The Electrical Subsystem preliminary design includes the switchgear and protective relaying equipment, the generator controls and indicators, the main stepup transformer, the lightning protection equipment, and station service and emergency power supplies for the WGS.

A stability analysis of the WGS connected to a typical utility distribution system was completed by Northeast Utilities and indicates the system to be stable for the combinations of conditions investigated.

8.1 Requirements

Functional requirements for the Electrical Subsystem are to produce electric power at a suitable voltage and frequency, compatible with standard electric utility requirements and practices. The electrical generating equipment must produce this power using minimum cost equipment to achieve competitive energy costs.

The equipment must operate at a remote, unattended site. Therefore, automatic fault protection and synchronization of the WGS with the utility network must be provided. The equipment must be capable of operating over the required range of wind speed, temperature and other environmental effects at remote sites and, particularly, of maintaining a stable interface with the utility network under expected wind gusts. This represents an additional requirement beyond that usually imposed on conventional utility generation equipment, since the power source, the wind, is a random variable. The WGS must, therefore, include provisions to limit the adverse effects of wind gusts on the WGS and the utility network. This is accomplished primarily through control of blade pitch (as described in the Control Subsystem section of this report) and by control of generator field excitation.

8.2 Design Approach

In selecting the Electrical Subsystem concept and developing the preliminary design, primary emphasis was placed on minimizing the cost per kilowatt-hour of energy delivered to the utility. Since the WGS spends a large percentage of its time operating below rated power, good partial power efficiency was an important consideration in the selection of the generating and interface equipment. Emphasis was also placed on the selection of commercially available off-the-shelf equipment designs and conformance to standard utility practices. Other decision criteria included equipment cost, total weight of the electrical subsystem on the tower, which affects tower cost, electrical subsystem reliability and maintainability, and utility equipment preferences and control considerations.

Equipment and/or techniques which are not in use by utilities or have not been evaluated by utilities were generally avoided. The equipment was designed to be compatible with standard utility maintenance and safety practices and to conform to ANSI, NEMA and OSHA standards.

8.3 Available Components and Selected Technology Level

The four major components of the Electrical Subsystem are the generator, the generator protective relaying equipment and controls, the transformer and the main breaker, which is between the transformer and the utility company lines. Each of these is discussed below.

The standard synchronous generator, which is widely used by utilities, was considered both in brush type; and brushless version. Also considered were DC generators, motor generator sets and solid state inverters. Induction generators which are presently used by some utilities for small water power installations were also evaluated. These devices are essentially low slip induction motors operated above synchronous speed.

There have been new generator developments recently involving special types of machines which can deliver constant frequency output when driven by variable shaft input speeds. These devices may offer some promise for WGS applications in the future. However, none of these was considered in the study because none has reached the stage of development where any utility experience has been accumulated.

Similarly, for generator protective and relaying equipment, several new developments which hold promise for reduced costs are underway. These include microprocessor or minicomputer devices which could be used to replace the standard electro-mechanical or solid state relaying devices commonly used by utilities. Because of the very limited utility experience base in using microprocessor/computer devices as primary generator protective and relaying equipment, they were not considered in this role for the WGS application. This is especially important for the WGS, since the system operates at a remote, unattended site. A microprocessor was, however, selected for startup and shutdown sequencing control of the WGS.

Although solid state relaying devices have not seen as wide an application in utilities as standard electro-mechanical relays, there is enough experience on these devices to allow their use in the WGS. The electrical protective and relaying system has been designed to allow use of either solid state relaying or electro-mechanical relaying devices at the option of the using utility.

Options for the interface equipment which ties the WGS into the utility network are limited. Transformers fall into two broad categories, either air-cooled or oil-cooled. Both types were considered here. Similarly, breakers may be either standard types used by the utilities for generator protection or may be of the recloser type normally used in distribution systems. Both types were considered for the WGS application.

As mentioned in the Drive Subsystem discussion, paragraph 7.3.2, the use of slippage devices or clutches between the generator and the gearbox is considered undesirable, unless the need for such devices is dictated by system stability or torque limiting considerations.

8.4 Concept Selection

In the WGS study, several alternative concepts for the Electrical Subsystem were evaluated. This effort considered not only the efficiencies and costs of the electrical equipments themselves, but also the resulting effect on efficiencies and costs of other components of the WGS and the net effect on overall energy cost.

8.4.1 Selection Criteria

All of the subsystem concepts selected for investigation had to be compatible with utility requirements, as well as compatible with any special requirements of the Wind Generator System. These concepts utilized standard, off-the-shelf equipment and offered advantages of reduced cost or increased efficiency for the WGS as a whole. Some of the variable rotor speed WGS concepts which were considered increased the cost of the Electrical Subsystem, and decreased its efficiency. However, these had the potential of decreasing the cost of other portions of the WGS and increasing total system efficiency by trading off electrical efficiency for rotor or drive efficiency. These concepts were investigated to determine whether a net reduction in cost per kilowatt hour could be achieved.

Additional decision criteria included total weight of the Electrical Subsystem on the tower, which affects tower weight and cost, Electrical Subsystem reliability and maintainability, utility equipment preferences and control considerations.

8.4.2 Alternative Configurations

There are two fundamentally different approaches for the Electrical Subsystem of the WGS. One approach is to use an electrical subsystem which can accept variable shaft rpm, allowing the rotor to operate at variable rpm. The second approach operates the electrical equipment at a fixed rpm and requires a fixed rpm shaft drive from the transmission to the generator. Both approaches must provide constant frequency output. Although operation at variable rpm complicates the electrical equipment, it does permit the rotor to operate at its most efficient rpm over a range of wind speeds. Another means by which the rotor can be allowed to operate over a varying rpm range without requiring the electrical system to operate at variable rpm, is to provide a variable ratio gearbox between the electrical generator and the rotor. This option is described in Section 7, Drive Subsystem.

Two concepts were considered for fixed rpm electrical systems. One using an induction motor operated as a generator, and the second using a synchronous alternator of the type normally used by electric utilities. Both of the above systems would be interfaced to the utility network through a step-up transformer and breaker.

For variable speed systems, a means must be provided to convert the variable shaft rpm into the constant frequency required by the utility network. The three concepts considered in the study generate DC power from the variable rpm shaft and then convert to AC power at constant frequency for delivery to the utility network. The DC power may be generated directly by a DC generator, or by a variable rpm AC generator with a transformer-rectifier unit. Although the efficiency of these two approaches is approximately the same, the cost of the DC generator is considerably greater than the cost of the AC generator and transformer-rectifier unit. The DC generator, being very heavy, also puts more weight on the tower, whereas the AC generator weighs less, and the transformer-rectifier unit can be located on the ground. This evaluation concludes that variable shaft speed concepts should use an AC generator driving a transformer-rectifier unit to produce DC power.

The three variable speed alternatives were then evaluated on the basis of how to convert the DC power into the required constant frequency AC. The three concepts investigated for accomplishing this are as follows:

1. A DC motor driving an induction generator

2. A DC motor driving a synchronous alternator

3. A three-phase solid state inverter

The relative advantages and disadvantages of these concepts are described below and in Tables 8-1 and 8-2.

8.4.3 Component Characteristics and Limitations

As previously mentioned, two fixed rpm generator concepts were evaluated; the induction generator and the synchronous generator. Both types have rated and partial power efficiencies which are very nearly equal. The induction generator does have an advantage in that it is a very simple and highly reliable machine. It is somewhat lower in cost than the synchronous machine. It is more tolerant of wind gusts and shows better stability and damping characteristics when connected to a utility network because of its characteristic operation with slip. It also has a greater tolerance for overspeeds than the synchronous generator. The principal disadvantage of the induction generator is that it must draw its exciting power from the line and, therefore, needs power factor correction capacitors on the line to avoid excessive VAR flow, and does not provide any means for direct voltage control.

The synchronous machine has the advantage of its own internal source of excitation with considerable flexibility in management of VAR flow to the network. It is also the standard utility machine, well understood and accepted. It does have the disadvantage of having more complex control requirements and inherently less stable operation.

Several limitations are associated with electrical equipment needed to perform the DC to constant frequency AC conversion for the variable rpm concepts.

TABLE 8-1. ELECTRICAL SUBSYSTEM EFFICIENCY AND COST - 100 KW SYSTEMS

	TYPE	NET EFFICIENCY % LOAD: FULL	3/4	1/2	1/4	TOTAL COST
1.	Constant RPM/ Induction Generator	91.2	89.8	86.8	78.0	$ 27,445
2.	Constant RPM/ Synchronous Generator	92.0	90.7	88.2	81.6	30,605
3.	Variable RPM/MG Set (Induction Generator)	74.8	72.8	69.7	57.6	54,255
4.	Variable RPM/MG Set (Synchronous Generator)	75.5	73.6	70.8	60.3	57,235
5.	Variable RPM/ Inverter	75.9	74.5	72.9	42.5	69,540

TABLE 8-2. ELECTRICAL SUBSYSTEM EFFICIENCY AND COST - 1000 KW SYSTEMS

	TYPE	LOAD:	NET EFFICIENCY % FULL	3/4	1/2	1/4	TOTAL COST
1.	Constant RPM/ Induction Generator		94.3	94.0	92.2	87.0	$ 59,305
2.	Constant RPM/ Synchronous Generator		95.0	94.4	92.2	87.5	86,815
3.	Variable RPM/MG Set (Induction Generator)		82.6	81.3	79.4	70.7	208,350
4.	Variable RPM/MG Set (Synchronous Generator)		83.2	81.6	79.4	71.1	235,860
5.	Variable RPM/ Inverter		78.3	77.0	75.5	44.4	364,325

DC machinery tends to be about 2 to 3 times heavier and 2 to 2-1/2 times more costly than equivalent AC equipment of the same power rating. It also has higher maintenance requirements due to brush and commutator life limitations. Efficiencies of the DC machines are somewhat lower than the equivalent sized AC equipment, especially at part power operation. This is an important consideration for the WGS application since much of its time is spent operating below rated power.

The solid state inverter offers potentially lower maintenance and possible improved reliability over rotating DC machinery. However, it has the major disadvantage of still higher cost (approximately 2 to 2-1/2 times the cost of an equivalent rated DC motor-AC generator set. The inverter also suffers from very poor partial power efficiency, lower than than the motor-generator combinations. The high cost of the inverter is primarily due to the high cost of semiconductor equipment at the required WGS power levels, and the fact that the inverter must be self-commutating, rather than line commutating, to maintain stability in the event of faults. The inverter must also have low harmonic output which requires that it include special shaping circuits and filters to approximate a sinusoidal output. This also increases its cost.

Another factor must be considered when equipments are cascaded to perform the conversion processes necessary with variable speed systems. The net efficiency of the overall system becomes the product of the efficiencies of all the individual parts. The net efficiency is, therefore, considerably reduced, especially for partial power operation compared to a single generator. There is also the extra cost for additional controls and protective equipment required for each component in the chain. These costs have been included in the estimates on Tables 8-1 and 8-2.

8.4.4 Electrical Subsystem Concept Evaluations and Selections

8.4.4.1 Generator

In the conceptual design phase of the study, two WGS ratings were evaluated: a low power (nominal 100 KW) system, and a high power (nominal 1000 KW) system. A summary of the evaluations of the concepts described previously showing the relative efficiencies and costs of the five concepts is shown in Table 8-1 for the 100 KW system and in Table 8-2 for the 1000 KW system. Costs for each of the systems are based on estimates obtained from manufacturers for the major equipments, combined with allowances for switch gear and protective equipment from standard catalog prices. Full load and partial power efficiencies were either obtained from manufacturers or calculated from the fixed and variable losses of each machine.

It can be seen from the tables that the variable rpm systems would need considerably higher rotor efficiencies or reductions in rotor system costs in order to be economically attractive. This was not the case, as shown in Section 3. Therefore, the two fixed rpm concepts were selected for optimization and preliminary design. The advantages and disadvantages of these two concepts are summarized in Table 8-3. It was recommended that the synchronous and induction generators be offered as a user option, depending on utility preferences. To provide this option, protective equipment and switch gear specifications were developed to be compatible with either type of generator.

TABLE 8-3. INDUCTION VS SYNCHRONOUS GENERATORS FOR CONNECTION TO UTILITY NETWORK

SYNCHRONOUS FEATURES	INDUCTION FEATURES
More readily accepted and understood by utilities	Does not need precise synchronization before connecting to network
Directly compatible with off network applications (self-exciting)	Better transient operating characteristics
Greater flexibility in management of reactive power and system voltage	Lower cost (with power factor correction) for 1000 KW and above (see Note)
Slightly higher efficiency	Less weight on tower
Single bearing type readily available	Automatic loss of excitation if separated from network by fault (Safety feature)
	Better overspeed capability
	Higher reliability

NOTE: Cost advantage for induction machine disappears if synchronous machine is rated at 1.0 pf rather than 0.8 pf.

If a synchronous generator is selected, it is possible to use the generator as a source of power factor correction for the utility network when the wind velocity is too low to generate any power output. This requires the addition of a motor starter, so the machine can be started with no wind available, and the addition of a disengaging clutch or overrunning clutch between the generator and the gearbox so the generator will not have to drive the rotor, which would consume excessive power, in this mode of operation. The generator will now behave like a variable capacitor, delivering lagging reactive power to the network, as controlled by its field excitation. Taking a 1500 KW machine as an example, the cost of the starter and the clutch device are in the $26,000 range. The installed cost of an equivalent amount of power factor correction capacitors and associated control switches are in the range of $10,000. The capacitors have the further advantage that they may be located in the distribution network at the point where the power factor correction is needed. The generator, on the other hand, must be located at the WGS. This may be at the end of a long feeder, remotely located from the point in the system at which the VARS are really needed, which will cause additional line losses. Also, the reliability of the WGS will be reduced due to the additional components required (clutch and motor starter) and the extra equipment needed to transfer between the two modes of operation. These facts, combined with the previous cost considerations, lead us to recommend against the use of the synchronous generator as a replacement for power factor correction capacitors in the distribution system.

8.4.4.2 Relaying and Generator Protective Equipment

For reasons previously stated in 8.3, standard relaying equipment was chosen for use on the WGS, as opposed to microprocessor or computer based relaying equipment. Two main options were evaluated for circuit interrupting devices; the breaker and the recloser. The recloser is a relatively low cost device which is commonly used for protection in distribution systems. Since the power level at which the Wind Generator System operates is similar to typical distribution systems, the possibility of using a recloser to save cost was investigated.

Two primary factors prevented the recloser from being a practical choice for the WGS application. Since the generator must be synchronized to the utility network, the device that closes the circuit must have a repeatable closing time in order to assure accurate synchronization at the time of closure. Manufacturers of reclosing equipment did not feel that they could provide the kind of repeatability tolerance necessary for acceptable synchronization with a standard recloser. They also pointed out that if the necessary sophistication were added to the recloser to enable it to function as required, it would then cost as much as a breaker, and would not be a standard piece of equipment. The breaker has contacts designed to minimize maintenance and permit a larger number of operations than the recloser, which is important for the WGS. The breaker has the further advantage that utilities are familiar with the use of breakers for generator protection, but they have essentially no experience using reclosers for this purpose. The breaker, therefore, was selected as the appropriate protective device.

8.4.4.3 Emergency Power Supply

Two alternatives were considered for providing emergency power to the Wind Generator System in the event that the connection with the utility network is broken. In such cases, power is required to maintain essential functions until the station service power can be restored. Two types of systems were considered:

1. A gasoline or diesel driven generator
2. Batteries with a charger continuously floating on the line.

The relative advantages and disadvantages of these two approaches are summarized in Table 8-4. Based on the evaluation summarized in Table 8-4, the batteries and float charger were selected. This requires a system design minimizing power consumption during emergency operations, a requirement which significantly influenced the control system design.

8.4.4.4 Electrical Connections Through Pintle

Various methods for passing the electrical power and controls wiring through the rotating pintle joint at the tower head were evaluated during the conceptual design phase. Two primary approaches were considered; first, using slip rings, and second, allowing the cable to twist, then periodically using the yaw mechanism to turn the nacelle to remove the twist.

Allowing the cable to twist is a technique that has been used on other large systems with success. Usually, the net twist accumulated over a period of time does not amount to more than a turn or two. This approach has lower cost and higher reliability than slip rings and is more suitable for the large numbers of wires that might be required for the control and signal cables. It does require a long length of cable to absorb the twist and a sensor for the yaw servo during the shutdown sequence to remove the twist.

The major disadvantage of this approach with respect to the power cables is that manufacturers of these cables do not recommend any twist or flexure for reliability purposes. It was, therefore, decided to run the power wiring through slip rings and to use direct connections for all the control and signal wiring, with the yaw servo being used to untwist the control cable as necessary during each shutdown of the system.

8.5 Preliminary Design and Analysis

The detailed design of the Electrical Subsystem should be tailored to the needs of the particular utility company. The design can also be influenced by the particular location in the network system where the WGS is installed. Its location in the system can affect the type of generator (induction versus synchronous), the size of the breakers required, the type and settings of the required relaying and protective equipment, etc. The types and arrangement of protective and relaying equipment selected for the preliminary design represent a consolidation of recommendations of several utilities - in particular, Northeast Utilities and Colorado Springs Public Utilities. Detailed specifications were prepared for each of the WGS major Electrical Subsystem components selected

TABLE 8-4. COMPARISON OF EMERGENCY POWER SYSTEMS

TYPE OF POWER SUPPLY	ADVANTAGES	DISADVANTAGES
Engine Driven Generator	High power capacity	High initial cost and high maintenance requirement Low reliability if started remotely and only used occasionally Power may still be momentarily interrupted until unit gets "On Line" Northeast Utilities does not recommend Hazardous fuel at remote site
Batteries and Charger (Charger continuously "On Line")	Inherently well filtered DC output for critical circuits Recommended by Northeast Utilities and used extensively by them for remote emergency power Low cost Higher reliability Non-interruptable source of power for critical loads Utilizes station service power as source for charger	Small power capacity (critical functions must be designed for low power consumption)

from the conceptual design and optimization phase results. Descriptions of these major components and the rationale for their specific features are given in the following subparagraphs.

8.5.1 Generator

Since the generator for the WGS is mounted on top of the tower, its weight should be minimized. For a given power rating, weight is minimized by using high rpm generators. This also tends to reduce cost, providing the rpm is not greater than the design limit for the class of generator. For standard machines in the size of 500 KW to 1500 KW, the upper limit of rpm is 1800 rpm for synchronous machines and 3600 rpm for induction machines. The synchronous generator upper limit of 1800 rpm was selected so that the system gearbox could be standard for both synchronous and induction generator options. The effect of operating the generator at higher rpm has very little effect on the gearbox, since the size, weight and cost of the gearbox is primarily influenced by the input shaft torque.

Selection of generator voltage is important since, for a given power rating, it affects the size and weight of the cable and slip ring needed to carry current down the tower. Since the vertical run of cable down the tower has to be supported against gravity loads, there is considerable incentive to keep the weight of the cable as low as possible. Larger generators (1000 KW and above) are readily available at 4160 volts output voltage. This was selected as the standard voltage for the high power WGS to minimize cable weight and cost. In the low power WGS (500 KW and below), the highest standard voltage readily available is 2400 volts. This was, therefore, selected as the voltage for the low power WGS generator.

Generator cooling is another important consideration in the WGS application. Outside temperatures can reach 49°C (120°F) and since this is the normal maximum ambient temperature for standard off-the-shelf generating equipment, it is important that the cooling air exhaust from the generator be ducted to the outside of the enclosure on the turntable. If adequate cooling is not provided, special generator designs will be required to handle the higher ambient temperature inside the cell.

Some generator requirements are peculiar to the type of generator being used in the WGS application. For the synchronous generator, for example, damper bars have been provided in the pole faces to improve stability of the generator when connected into the utility network (because of wind gust torque fluctuations). Also, an integral brushless exciter has been incorporated into the design to eliminate brushes and slip rings which are potentially high maintenance elements. This also eliminates a potential source of electromagnetic interference.

There are also requirements peculiar to the induction generator. The power factor of the selected induction machine should be as high as possible, without sacrificing other design parameters, to reduce the cost of the power factor correction capacitors needed to connect the machine into the utility network. The induction machine is a low slip design so that it operates at a high efficiency, comparable to the efficiency of the synchronous machine.

8.5.2 Protective and Control Equipment

There are three main functions which must be performed by the generator protective equipment:

1. The equipment must prevent and/or limit damage to the generator and accessory equipment. Proper protection of the generator is especially important in order to reduce the probability of having to remove the generator from the top of the tower, which is an expensive and difficult operation.

2. The equipment must provide protection for utility company customers against failures which might occur in the Wind Generator System. Failures within the wind generator itself must not disrupt service on utility company lines.

3. The protective equipment must differentiate between internal failures of the Wind Generator System and external failures on the utility company line. In the case of internal failures, the WGS must be shut down and locked out until a repair can be made. In the case of external failures, the WGS must be removed from the line and brought back on line after the external fault has been cleared, and the line reenergized from the network.

A one line diagram of the protective equipment arrangement designed to meet these requirements is shown in Figures 8-1 and 8-2 and described below. The generator is shown at the left side and the connection to the network on the far right side in both figures. Note that relays in Figures 8-1 and 8-2 are identified by their NEMA device numbers.

8.5.2.1 Internal Fault Protection

The generator is normally Y-connected with the neutral point of the Y returned to ground through a ground fault detection relay and series current limiting impedance. Most likely failures inside the generator will begin as a fault to ground and can thus be detected and tripped off the line before significant damage occurs. The magnitude of the fault is limited by the ground return impedance.

The percent differential current relay, number 87 in Figure 8-1, will also detect phase to ground and phase to phase faults inside the differential protective loop of the WGS (main breaker, main transformer, slip ring, generator and associated wiring). This relay, however, cannot be set to operate at as sensitive a level as the ground fault relay, and although it can detect a broader class of failures, including phase to phase faults, it will generally not be as sensitive as the ground fault detector.

A reverse power relay is also provided. Its main function is to detect failures in the rotor pitch control system which could cause excessive power flow from the network to the generator. The generator would then behave as a motor to drive the rotor in a fan mode, thereby absorbing power from the network, and possibly causing voltage fluctuations.

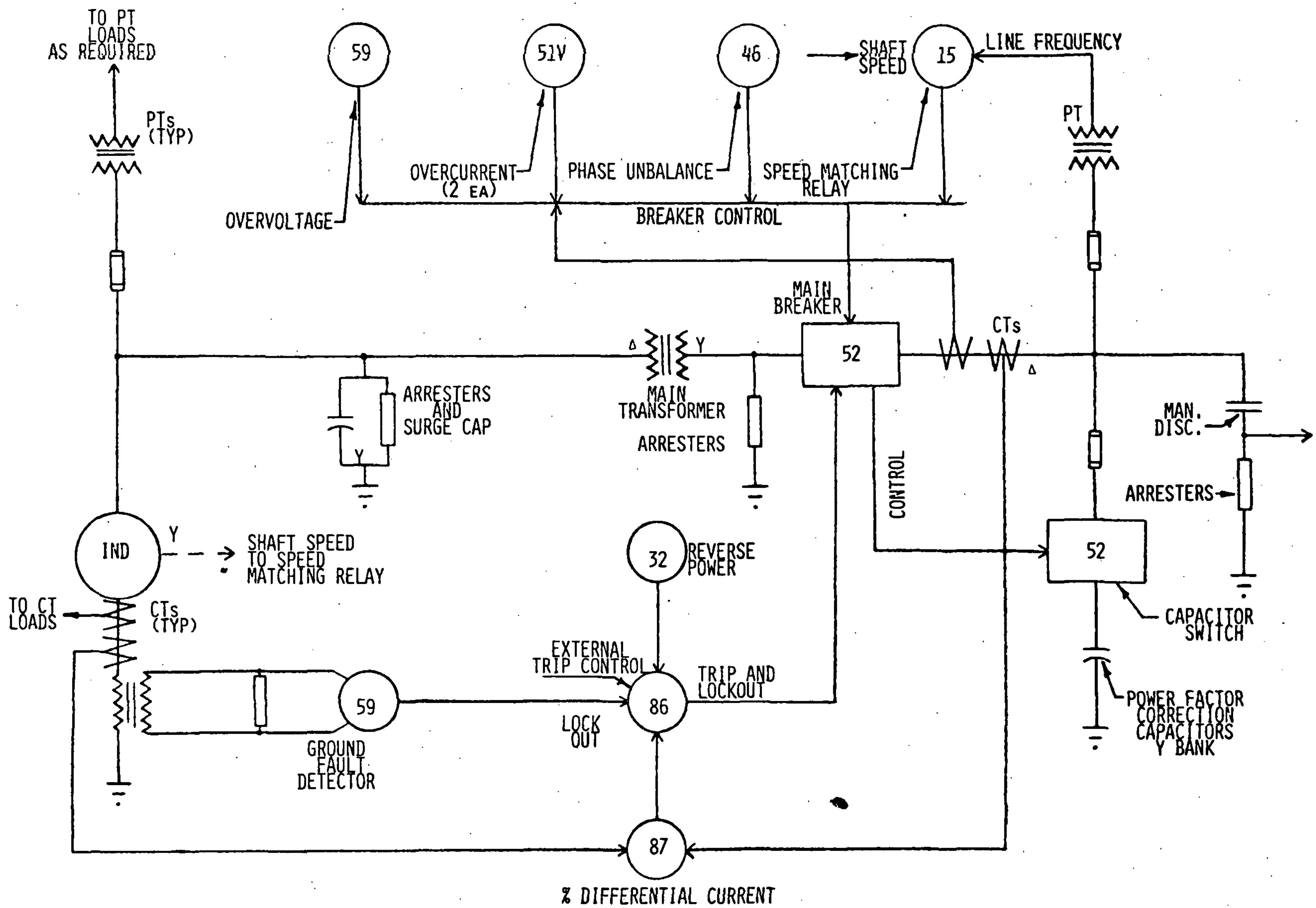


Figure 8-1. Induction Generator Protection and Controls

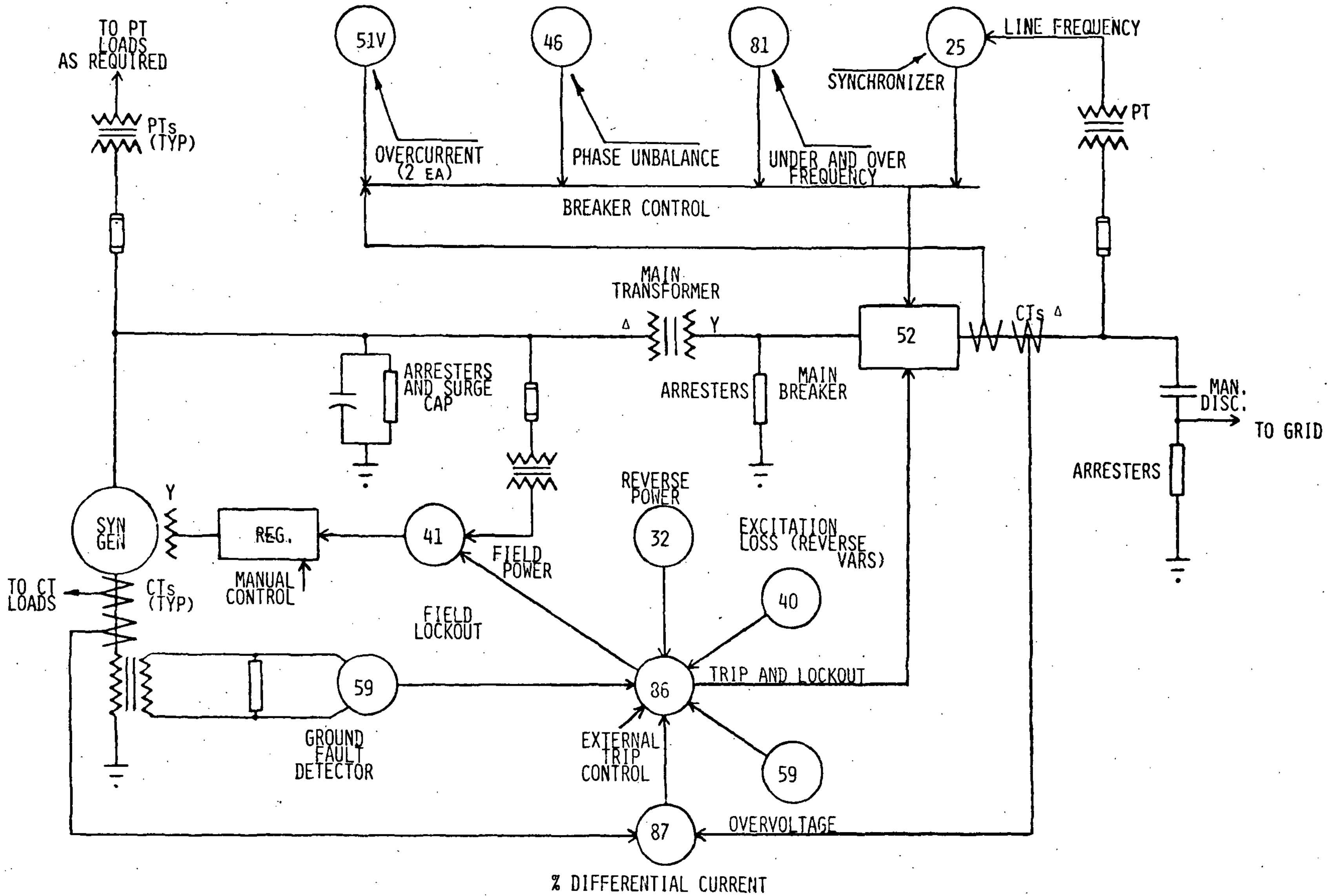


Figure 8-2. Synchronous Generator Protection and Controls

As shown in Figure 8-2, the synchronous generator requires additional relaying to provide protection for the field control of the machine and associated functions. This includes an over-voltage relay which indicates failures of the field regulator, and an excitation loss relay to indicate loss of field excitation and reverse VAR flow, which can allow operation of the synchronous generator as an induction generator causing excessive heating and vibration.

This combination of relaying equipment, recommended by Northeast Utilities, should detect all significant electrical faults which can occur within the WGS.

8.5.2.2 External Fault Protection

Failures outside the WGS are detected by relaying equipment shown at the top of Figures 8-1 and 8-2. The most likely type of external fault is a single phase-to-ground fault. This is detected by the phase unbalance relay (number 46) in both figures. This function is backed up by redundant overcurrent relays (number 51V) which detect three-phase faults, as well as phase-to-phase and phase-to-ground faults. Since the fault current may be very close in value to the normal maximum rated load current, the relay selected is a voltage restrained type. The current trip value is set very close to normal rated current and the relay only trips when it sees low voltage simultaneous with over-current. This permits more sensitive operation of the relay.

Certain protective relays for external faults are specific to the type of generator used. An over-voltage relay is required for the induction generator to protect against the possibility of the external connection to the utility network being opened and leaving the power factor correction capacitors connected across the generator. This can result in resonance between the capacitors and the generator and the production of high voltages. The over-voltage relay detects this occurrence and trips the main breaker until the external connection is restored. The capacitors are arranged in such a way that they are disconnected from the generator whenever the main breaker is tripped, as shown in Figure 8-1.

For the synchronous machine, an under- and over-frequency relay is required to protect against the possibility of the generator being separated from the network and the network subsequently attempting to reclose on the generator while in an out-of-sync condition. If the generator becomes separated from the network with a block of load, its frequency will drift off 60Hz network frequency and will be detected by the under- or over-frequency relay. This will open the WGS breaker to prevent network reclosure in an out-of-sync condition. Protective equipment, such as reclosers or breakers the utility may have located at the end of the feeder line for the WGS, should also be equipped to detect the presence of voltage on the line from the WGS and should not allow a reclosure unless the line is clear of excitation. This will provide back up protection for the under- and over-frequency relay in the WGS.

8.5.2.3 Generator Control Equipment

The only additional equipment required for the induction generator is a remote-controlled switch to pull the power factor correction capacitors on and off the line at the same time as the generator, and a speed matching device to match the shaft speed of the generator to the line frequency before closing the main breaker.

For the synchronous generator, the control problem is somewhat more complicated. The frequency, phase and voltage of the generator must be accurately matched to the line before closing the breaker. This is accomplished with a standard automatic synchronizer coupled to the rotor blade pitch controls, as described in 5.5.1.1.2. This is backed up by a sync verifier which prevents closure of the breaker in the event of failure of the main synchronizer. These devices are checked for proper operation on a monthly basis by the maintenance operator.

The synchronous generator also requires control of field excitation. The level of sophistication required will depend on requirements of the particular utility and the location in the utility network where the WGS is connected. In general, the following features would be required:

1. Voltage Regulator. The selected regulator includes the ability to sense generator output voltage and provide voltage regulation if required by the utility. The specific regulation requirements must be determined as a part of the detailed design as discussed in 8.7.3 and 8.7.4. For certain installations, simply holding constant field excitation or VAR flow may be sufficient. Other installations, near the end of distribution feeders, may require voltage regulation.

2. Minimum Excitation Limits. The regulator includes the ability to set minimum excitation limits so that excitation cannot fall below the level where the generator might be pulled out of synchronism by wind gusts.

3. Field Forcing. The regulator includes provisions for externally controlled field forcing so the field may be boosted during wind gusts to prevent the generator from pulling out of synchronism with the network.

4. Maximum Excitation Limits. The regulator includes provisions so that the maximum steady state excitation supplied to the generator may be set to the maximum steady state generator excitation limit. This limit does not interfere with momentary field forcing.

5. Underfrequency and Overvoltage. Underfrequency and overvoltage protection are provided which reduce generator output voltage proportional to frequency to maintain a maximum limit on volts per Hertz. This prevents damage to the generator if the field should remain excited at less than rated rpm. Provisions are included to remove power from the regulator and lock out the field excitation if the voltage exceeds 125% to 150% of nominal, indicating a possible regulator failure. Manual reset is then required to restore WGS operation.

8.5.2.4 Lightning and Transient Protection

The locations of the lightning and transient protectors for the WGS are also shown in Figures 8-1 and 8-2. A distribution type arrestor is provided on the line side of the manual disconnects. This arrestor also serves to protect the line side of the main breaker. Another distribution class arrestor is located on the high side of the main transformer to protect it from coupled transients back through the main breaker. A station class arrestor, with bypass capacitors, is located near the generator terminals to protect both the generator and slip ring assembly from transients coupled back through the power line. If the connection between the tower head slip ring and the generator is well shielded and enclosed in conduit, the arrestor might be located just below the slip ring assembly and would then protect both the slip ring and the generator.

The general philosophy for lightning protection of the WGS is to provide good current paths along the outside surface of the tower head and tower so that the equipment inside is effectively in a faraday shield. The blades are also provided with current carrying conductors to carry any lightning currents from the blades down to the hub where they are transferred to the skin of the tower head enclosure. Internally mounted equipment is enclosed in metal racks and wiring is run in metal conduits to provide protection against induced transients.

8.5.3 Electrical Utility Interface

As shown in Figures 8-1 and 8-2, the interface between the wind generator and the utility electrical distribution system consists of a stepup transformer, a breaker and a manual isolation switch. Unit connections to the distribution system have been recommended. That is, each WGS unit has its own independent transformer and breaker, even if several WGS units are located in the same area. This permits independent connection and disconnection of the units from the power grid.

An oil-filled transformer has been selected because it is lower in cost than the equivalent air-cooled transformer. Also, oil cooled units are more suitable for outdoor applications. It is also recommended that the WGS be connected to the utility grid at the distribution level, rather than at the transmission level in order to keep interface equipment costs down. This results in transformer high side voltage ratings in the 2400 volts to 34.5 KV range. The transformer can then be protected by the same protective devices which serve the generator with the addition of an overtemperature detector in the transformer oil. Northeast Utilities has also recommended that an allowance be made for broken delta ground fault detection on their feeder line. This is included in the cost estimates, although it is not shown in Figures 8-1 and 8-2.

The breaker is located on the high side of the distribution transformer so that when the breaker is open, the system is completely isolated from the utility network. This approach also eliminates the power loss that would result from having the main transformer draw magnetizing current from the utility lines when the WGS is shut down. If the WGS were supplying an isolated load and not connected to the utility network, the breaker could be sized to handle the generator fault current only. However, if the WGS is tied into a utility grid, large currents can be generated by the grid feeding back into faults within the WGS. Therefore, the breaker has been sized to interrupt these currents, leading to breaker interrupting ratings in the 250 to 500 MVA class for a typical distribution system connection. An investigation of typical fault currents and standard utility practices has led to the selection of the 500 MVA size as a standard. However, particular installations could use smaller ratings if analysis shows the expected fault current to be less. If changes are made, it is recommended that standard ratings such as 250 or 500 MVA be used. Back up protection of the breaker with fuses was not recommended by either Northeast Utilities or Colorado Springs Public Utilities and, therefore, was not included in the design.

Backup protection for the WGS breaker can be provided by the substation breaker at the other end of the distribution feeder. A communications link may be required to coordinate the tripping of the WGS and substation breakers. A detailed analysis of the particular installation would be necessary to determine need for backup protection and the required coordination. This is discussed further under the electrical stability analysis in 8.5.7.

Northeast Utilities recommended that the main stepup transformer and breaker be located at the base of the WGS tower. This reduces cost under a system where the transformer and breaker are located in a remote switch yard several hundred feet from the tower. The transformer and breaker are outside-mounted on a pad which is an extension of the foundation of the small building at the base of the tower used to house the protective equipment and switch gear. The breaker and transformer are further protected from the weather and flying ice by an extension of the roof of the building. This location eliminates the need to run long control lines from the protective relaying equipment out to the breaker trip coils and reduces the hazard of a fault in these lines preventing operation of the breaker. The power leaving the WGS site is, therefore, at distribution voltage levels which reduces the size and cost of the underground power cable. The recommended arrangement also saves costs by eliminating the need to have both power and control conduits running from the WGS site to a remote switchgear pad.

8.5.3.1 Starting Augmentation

Aerodynamic starting of the WGS rotor is described in Section 4. If necessary, the generator could be operated as a motor and used to provide assistance to reduce the starting time. The induction generator is better suited for this role than the synchronous machine. The synchronous machine can be operated as an induction motor by removing the field excitation. However, the synchronous machine is not well suited for this type of duty. The induction machine is capable of accelerating the large inertia of the rotor blades using a standard autotransformer type of starting device. The synchronous machine, on the other hand, would have to be overrated to handle this duty if it used a standard starter, or would have to be supplied with current from a variable frequency source to limit the starting surge. Either of these alternatives would add considerable complexity and cost to the WGS system. Even for the induction machine, the costs of starting augmentation are significant. A standard autotransformer starter for the 1500 KW size induction machine has a purchase price in the $20,000 range.

If the starter is operated on the lowest voltage tap (50% voltage) to maintain a minimum starting current surge on the line, the voltage pulldown of the line can still be excessive. Northeast Utilities normally allows a maximum fluctuation of 3.1% if the fluctuation occurs no more often than once per hour. Fluctuations produced by the starting surge are tabulated below for the WGS connected at various locations on the typical feeder shown in Figure 8-3. It can be seen that the fluctuations would exceed the established limits in all cases tabulated below:

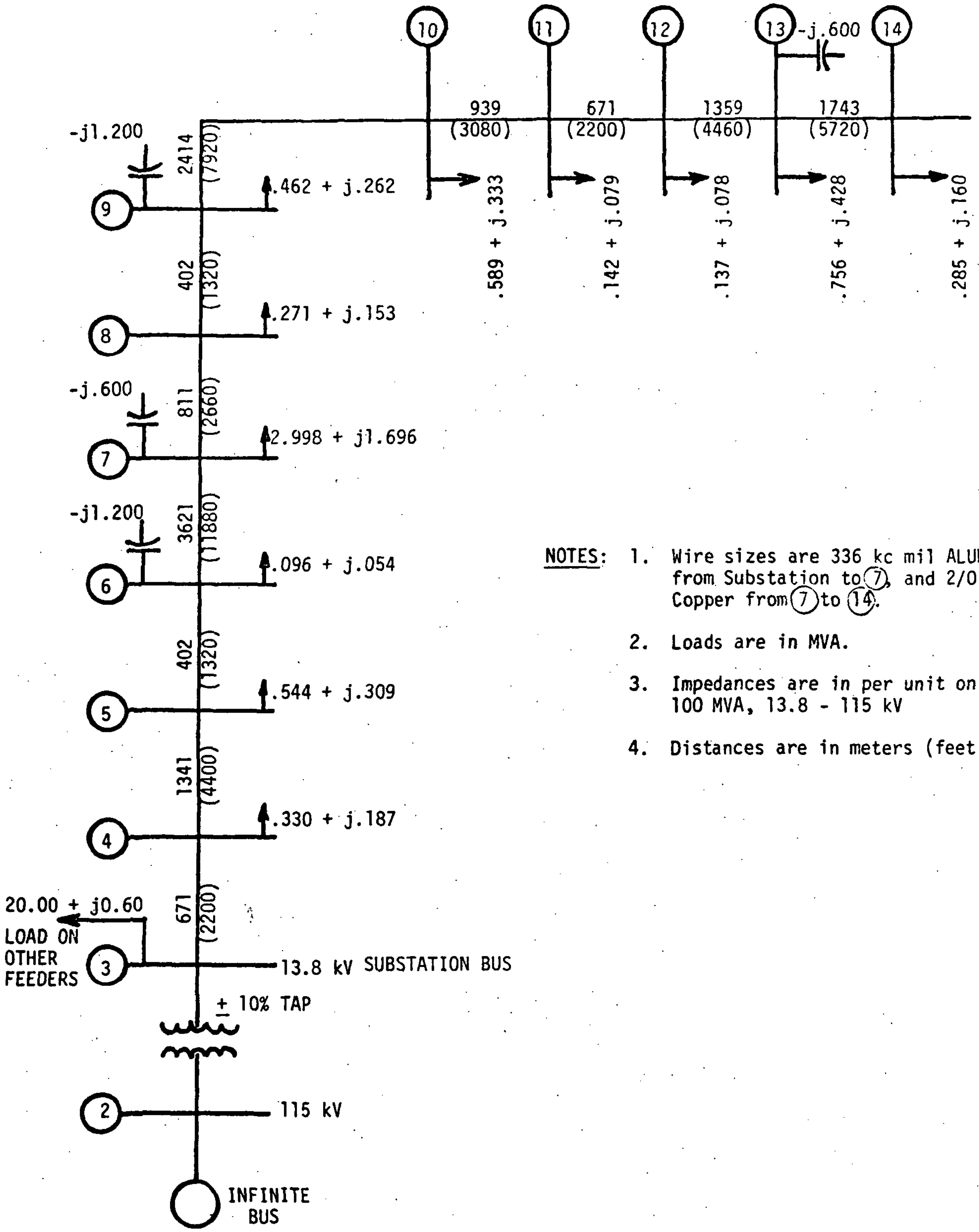
13.8 kV FEEDER	% VOLTAGE DROP	
LOCATION NUMBER	AT FEEDER LOCATION	AT GENERATOR
13	6.6	16.0
10	4.7	13.9
7	3.3	12.5

This would then force us to provide a more sophisticated starting scheme to limit the current surge, with still greater cost and complexity; or would limit the locations where the WGS can be connected to the feeder. Therefore, there is considerable incentive to eliminate electric starting augmentation if it is not essential. The analysis of Section 3 shows pure aerodynamic starting to be practical, therefore, no provisions for electric starting augmentation are included in the preliminary design.

8.5.3.2 Slippage Devices

Stability of the WGS when it is connected to the utility network and subjected to wind gusts is discussed under paragraph 8.5.7, herein. In addition to the recommended methods of controlling stability through control of the prime mover (blade pitch) and through control of the generator field excitation, it would also be possible to add a viscous coupling to the generator shaft. Table 7.4 of the Drive Subsystem shows some of the available devices which might accomplish this function. Of the devices listed, only the eddy current clutch, magnetic particle clutch and hydraulic coupling provide the necessary viscous coupling to augment stability. Of these, only the hydraulic coupling has sufficient power rating in off-the-shelf hardware for the WGS application. Mechanical slippage devices, such as plate or drum clutches, multiple V-belts, etc., have an extremely non-linear slip vs torque characteristic which makes them more suitable as overtorque limiting devices, rather than stability augmentation devices.

The hydraulic coupling, although capable of augmenting stability, would do so at the expense of increased power loss with a loss in overall system efficiency of 3 to 6%, and an increase in overall cost and complexity. Results of the preliminary stability analysis of 8.5.7 indicate that the WGS will remain on-line with a rigid coupling between the rotor and the generator, even under the most severe gusting conditions studied. This led us to conclude that reasonable system stability can be achieved without the need for slippage devices, and since these devices are subject to the disadvantages described above, they were not included in the preliminary design of the WGS. Detailed analysis of each



NOTES:

1. Wire sizes are 336 kc mil ALUM from Substation to 7, and 2/0 Copper from 7 to 14.
2. Loads are in MVA.
3. Impedances are in per unit on 100 MVA, 13.8 - 115 kV
4. Distances are in meters (feet)

Figure 8-3. Distribution Feeder

utility installation may show, in the case of some installations remotely located from the substation, that a slip coupling is necessary to achieve satisfactory stability with severe gusting. The slip coupling should only be used as a last resort if generator field control and blade pitch control cannot achieve satisfactory stability.

8.5.4 Maintenance Provisions

8.5.4.1 Manual Controls

The following manual controls have been provided for safety and/or maintenance purposes, for both the induction and synchronous generators:

1. Isolation Switch. A manual isolation switch on the high side of the breaker is provided so that the generating system can be isolated from the power lines for maintenance purposes. It is only required that this switch have isolation capability, not fault interrupt capability.

2. Manual Breaker Control. Manual trip and close control are provided for the main breaker. The trip control is capable of opening the breaker under any conditions at the WGS. The manual close control is only able to close the breaker if the synchronizer verifies proper conditions for closure.

3. Auto/Manual Switch. This device transfers all functions which are normally performed automatically over to manual operation for maintenance purposes.

4. Lockout Reset. In the event that a critical failure has caused a shutdown and lockout of the system, this control allows the lockout to be reset and the system to be operated after a repair has been completed.

For the induction generator, an additional control is provided to allow manual operation of the capacitor switch which connects and disconnects the power factor correction capacitors from the line. For the synchronous generator, an additional control is provided to allow manual operation of the field excitation for maintenance purposes.

8.5.4.2 Visual Indicators

The following visual indicators have been provided for the operator as an aid in maintaining and operating the system:

1. A switched AC voltmeter is provided to monitor the voltage on any one of the three phases selected, either at the bus or at the generator side of the breaker.

2. A switched AC ammeter is provided to monitor current flow on any one of the three phases selected.

3. An AC watt meter is provided to give continuous visual indication of the total real power being delivered by the system to the line.

4. A kilowatt-hour meter to provide continuous visual indication of cumulative kilowatt-hours delivered by the system is also included.

5. A frequency meter is provided covering a range from 55 to 65 Hz.

6. A power factor meter is provided to give a separate power factor indication for each of the three phases via switch selection.

7. A speed error meter is provided for the induction generator to indicate the relative speed difference between the generator shaft RPM and equivalent RPM of the network before the breaker is closed. Full scale range of indication is set at $\pm$ 5% of rated speed.

8. A synchroscope is provided for the synchronous generator to give visual indication of synchronization. This will assist in checkout and maintenance, as well as cross-check for proper operation of the synchronizer and sync verifier. In addition, the synchronous generator is equipped with DC field meters to indicate regulator output voltage and current.

8.5.5 Station Service Power

Single phase station service power is provided by a standard distribution transformer connected directly to the distribution line feeding the WGS unit. This station service connection is provided with its own kilowatt-hour meter for revenue metering purposes and has a standard 110/220 volt breaker panel. Loads provided by the station service power include lighting, equipment heaters, inching motor drive, auxiliary hydraulic pump motor drive and other utility circuits. The station service bus also provides power to the 24 volt DC power supply which is used as a float charger for the 24 volt emergency battery system discussed in the following paragraphs.

8.5.6 Emergency Power System

Connections for the emergency power systems are shown in Figure 8-4. The emergency DC loads, such as the rotor brake, servo electronics, and supervisory and sequencing control, are provided with power either directly from the 24 volt DC power supply or from the 24 volt battery with no switching required to transfer from one to the other. In the event of a battery failure, such as a battery short, a breaker is provided to isolate the battery from the supply and trip an alarm to initiate shutdown of the WGS. The main WGS breaker trip lines are redundant and independently connected to the power supply and the battery to insure a high reliability for tripping the main breaker.

Critical AC loads are normally supplied directly from the station service bus. In the event of failure of the station service power, the loads are automatically transferred to an inverter which is supplied, in turn, from the emergency

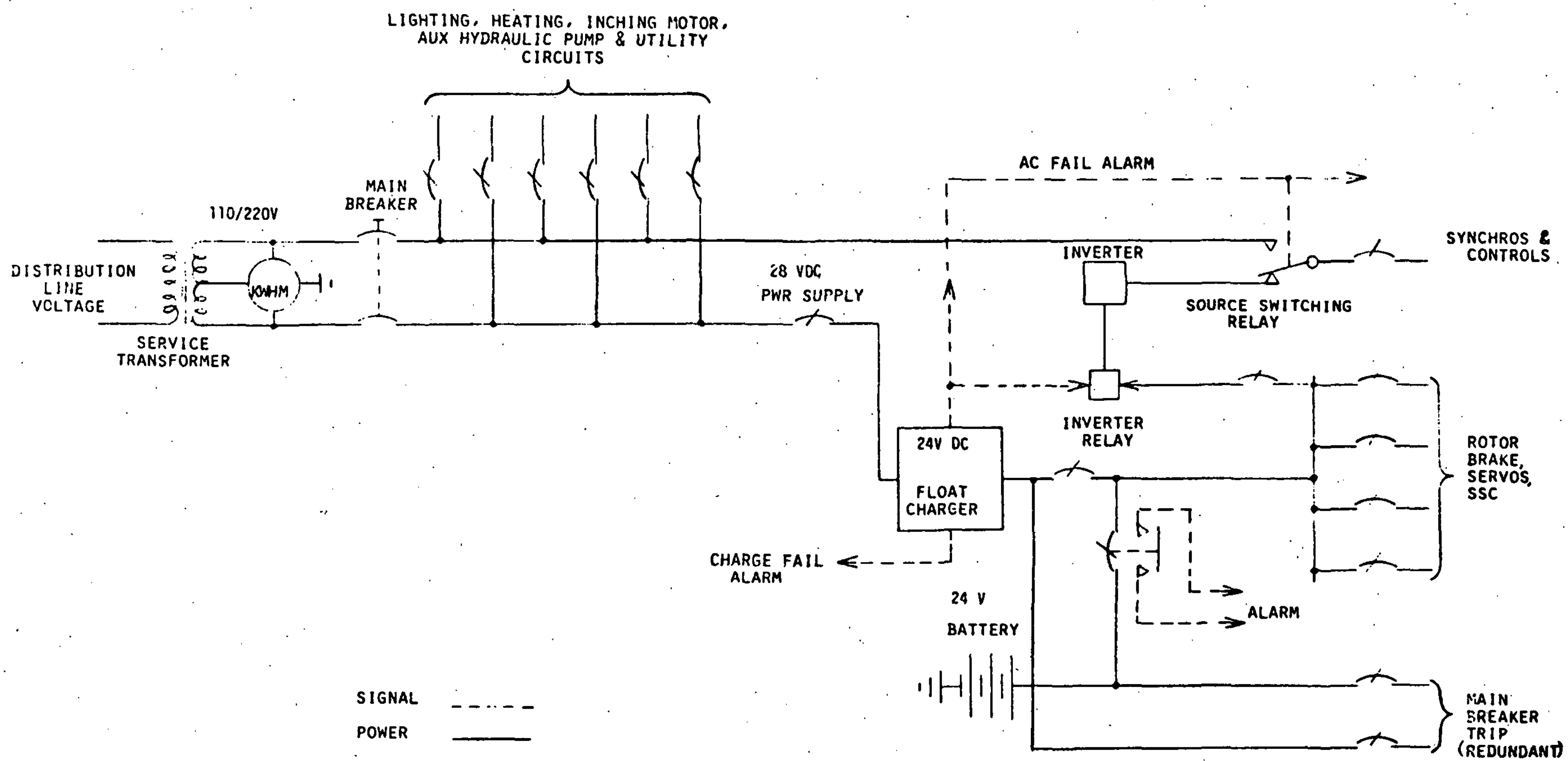


Figure 8-4. Station Service and Emergency Power Supply

power battery bus. Loss of AC station service power does not result in immediate shutdown of the system, but initiates a time delay which causes a shutdown if power is not restored within two minutes.

The 24 volt power supply is also equipped with a charge fail alarm which indicates failure of the charger independent of the AC power source. Failure of the charger would initiate immediate shutdown and lockout of the WGS to prevent excessive drain on the emergency battery.

The emergency battery is one component of the WGS Electrical Subsystem which is sensitive to ambient temperature. At high temperatures, battery life is seriously reduced; and at very low temperatures, battery capacity is reduced. It is, therefore, recommended that if the WGS is operated in climates where temperature extremes are expected, an underground enclosure should be constructed at the time the tower foundations are put in. The battery equipment should be located in the enclosure where its temperature will be stabilized by the surrounding earth. This eliminates the need for special environmental control equipment in the equipment shed. The same enclosure can be used for other sensitive equipments if the need arises.

8.5.7 Transient Analysis of Wind Generator

A stability analysis was performed by Northeast Utilities for the 1500 KW WGS. Only the 1500 KW system was studied for the preliminary design, since the stability of the low powered 500 KW system should be better than the 1500 KW when connected to the same network.

When the synchronous wind generator unit is connected to the electrical distribution system, the interaction must be analyzed to ascertain the effect this unit will have on the operation of the distribution system and to evaluate the requirements for protective and control equipment for the electrical system and wind generator. A study was conducted to determine the general response of the unit to faults and switching operations in the distribution system and to changes in unit input torque due to wind gusts.

To analyze system response, the electrical characteristics of a typical distribution feeder in the Northeast Utilities system and the mechanical and electrical constants of the wind generator were represented in a model. This model was then subjected to various disturbances which might be expected to occur on an electrical distribution system (i.e., faults caused by lightning or contact with tree limbs) and subsequent corrective action was simulated (open feeder circuit breakers and reclose). Three-phase faults were simulated, since special studies are required to represent unbalanced faults. However, these studies should be done for specific installations since the frequency of occurrence of these less severe faults is much higher than for three-phase faults. The model was also subjected to changes in generator input torque representing both increasing and decreasing wind gusts.

The response of the wind generator, in terms of its shaft speed, torque, voltage and current output, was determined as a function of time. Voltages and power flows on the distribution feeder were also monitored. The modeling was done by use of a transient stability program which is a standard tool used by electric power system engineers in the study of dynamic performance of power systems. While the results of this study are typical, giving general characteristics of system performance, each specific installation should be studied separately to establish operating conditions and requirements particular to that installation.

Figure 8-3 gives the electrical characteristics of the distribution feeder studied. The feeder is 13.8 kV primary voltage, approximately 14.5 kilometers (9 miles) in length and has a peak load of 7 MW distributed over its length. The wind generator is connected to the feeder by a three-phase tap with a length of 1.6 kilometers (1.0 mile). Various locations for connection to the feeder were analyzed.

The electrical synchronous generator was represented in detail with physical and transient saliency modeled and with generator saturation included. Inertia of the generator rotor was combined with the reflected inertia of the rotor blades and the dynamic response of this rotating system was monitored. Variations in prime-mover torque, due to wind gusting conditions, were applied to the rotating system after adjusting for the maximum corrective rate of the blade pitch control system, as described in Section 5 (5.5.2.1).

A static excitation system to control generator terminal voltage was also modeled. Generator and regulator parameters are listed in Table 8-5.

Table 8-6 summarizes the cases illustrated in Figures 8-5 through 8-10. These figures give typical results from study cases of the simulation of disturbances caused by feeder faults and wind gusts. In Figures 8-5 through 8-10, the generator rotor angle is the angle between the quadrature axis of the generator rotor and a synchronously rotating reference at angle 0°. Generator output power is electrical power, corrected for losses, as measured at the generator terminals. Shaft input torque represents the torque on the rotor blades after correction for control system response is made, as described in Section 5. Distribution feeder voltage is the voltage in per unit on a 13.8 kV base measured at bus 13 near the end of the feeder. Generator terminal voltage is also expressed on a 13.8 kV base.

Results of the study indicate that the wind generator should be disconnected from the distribution feeder for any disturbance which causes the normal supply to the feeder to open. This will prevent the wind generator from attempting to supply the load on the feeder in an isolated mode. Case 131-4R (Figure 8-6) shows the response of the WGS, located near the end of the feeder, to a distribution line fault on the feeder, assuming the WGS breaker is not tripped by the fault. After the feeder circuit breaker is opened (at 0.09 sec, normal operating time), the voltages along the feeder collapse to approximately 50% of normal as the 1500 KW wind generator attempts to supply 7 MW of customer load. If the WGS is not disconnected immediately, potential damage to customer equipment could result. In addition, proper fault clearing cannot be assured unless all sources, including the wind generator, are interrupted for a length of time sufficient for the fault arc to extinguish. Reclosure of the feeder circuit breaker can then take place, followed by re-synchronization of the wind generator. WGS protective relaying should, therefore, be designed to remove the WGS from the network for this type of a fault.

TABLE 8-5. WIND GENERATOR PARAMETERS
SYNCHRONOUS GENERATOR
1.875 MVA, 0.80 pf
4160 volt
1800 rpm
Direct-axis Synchronous Reactance = 1.847 per unit on 1.875 MVA
Quadrature-axis Synchronous Reactance = 1.792 per unit on 1.875 MVA
Direct-axis Transient Reactance = .195 per unit on 1.875 MVA
Stator Leakage Reactance = .076 per unit on 1.875 MVA
Open-circuit Time Constant = 2.72 seconds
STATIC EXCITATION SYSTEM
Regulator Gain = 200
Regulator Time Constant = .02 sec
Regulator Limits = 0.0 to 3.00 per unit
Exciter Time Constant = .128 sec
Feedback Time Constant = .480 sec

TABLE 8-6. DESCRIPTION OF TRANSIENT STABILITY CASES

<u>CASE 31-4R</u> (Figures 8-5a, b and c)

Wind generator is connected directly to substation bus through 1.6 kilometer (one mile) tap. Three-phase fault at bus 13 near end of distribution feeder is cleared in 0.09 seconds by opening feeder circuit breaker at substation. Feeder breaker is reclosed at 0.50 seconds re-establishing customer load on the feeder.

<u>CASE 131-4R</u> (Figures 8-6a, b and c)

Wind generator is connected to bus 13 near end of distribution feeder through 1.6 kilometer (one mile) tap. Three-phase fault at bus 13 near end of distribution feeder is assumed to clear in 0.09 seconds by opening feeder circuit breaker at substation. Wind generator remains connected to feeder attempting to supply customer load. Feeder breaker is reclosed at 0.50 seconds. Normally for this type of a fault, the WGS protective relays would remove the WGS from the feeder.

<u>CASE 10-1</u> (Figures 8-7a, b and c)

Wind generator is connected directly to substation bus through 1.6 kilometer (one mile) tap. Unit is subjected to positive wind gust of 2.0 second duration producing an input torque of 3.0 MW at peak with the gust peak at 1.5 seconds from the start of simulation.

<u>CASE 10-2</u> (Figures 8-8a, b and c)

Wind generator, connected as in Case 10-1, is subjected to decreasing wind gust of 2.0 seconds duration allowing input torque to decrease to zero at 1.5 seconds from start of simulation.

CASE 139-4 (Figures 8-9a, b and c)

Wind generator is connected to bus 13 near end of distribution feeder through 1.6 kilometer (one mile) tap. Unit is subjected to same wind gust as in Case 10-1.

<u>CASE 139-5</u> (Figures 8-10a, b and c)

Wind generator, connected as in Case 139-4, is subjected to the same decreasing gust as in Case 10-2.

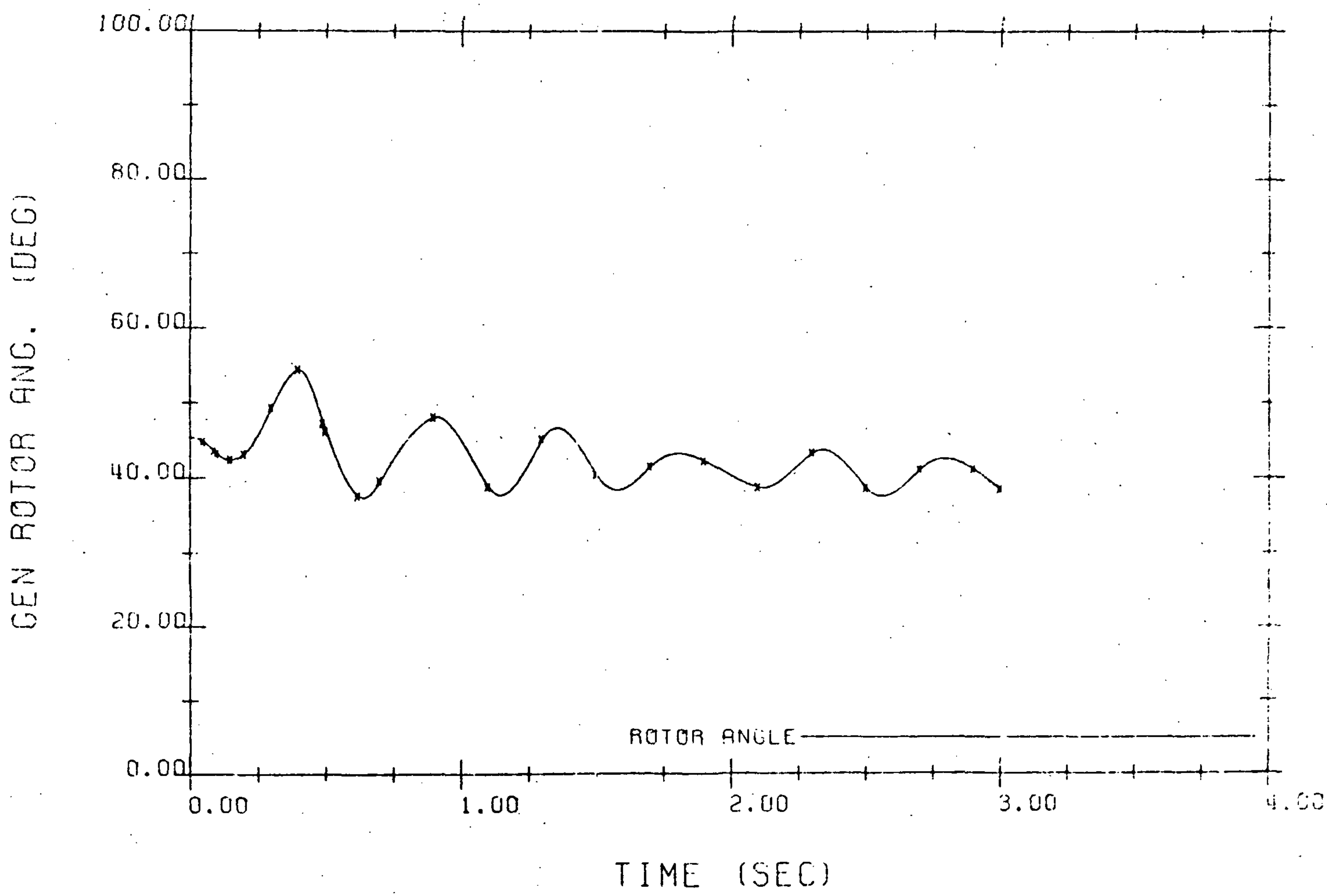


Figure 8-5a. Case 31-4R

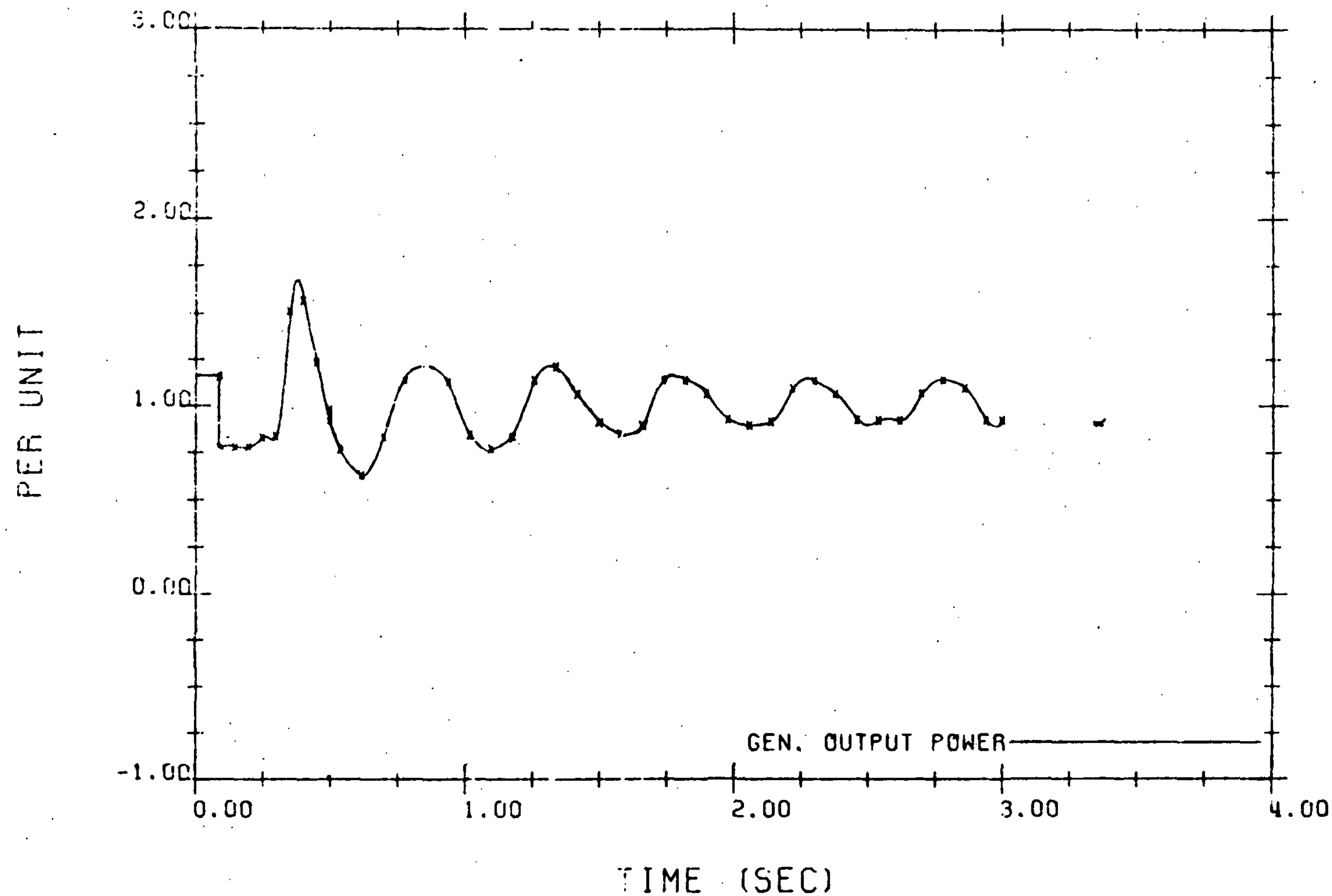


Figure 8-5b. Case 31-4R

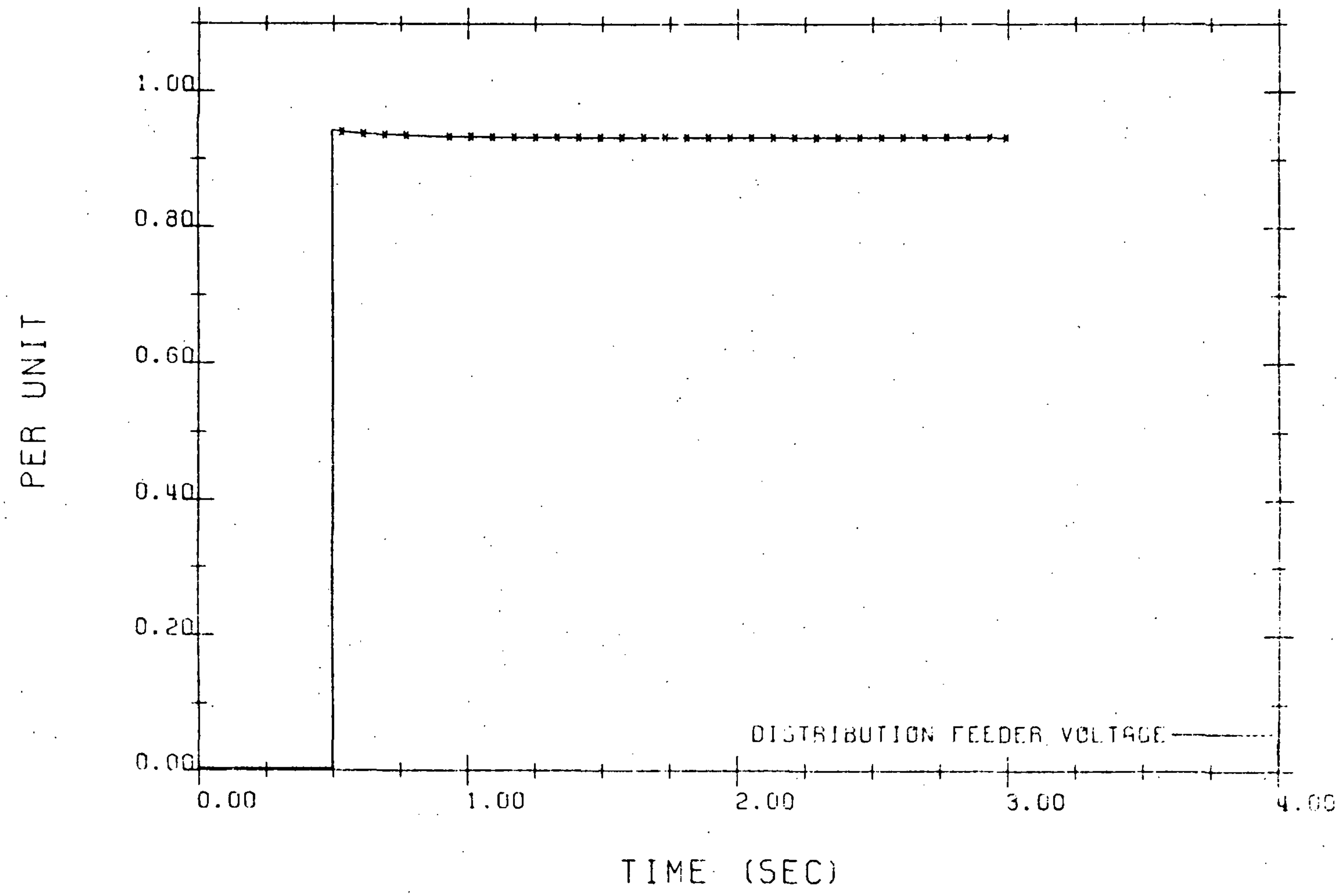


Figure 8-5c. Case 31-4R

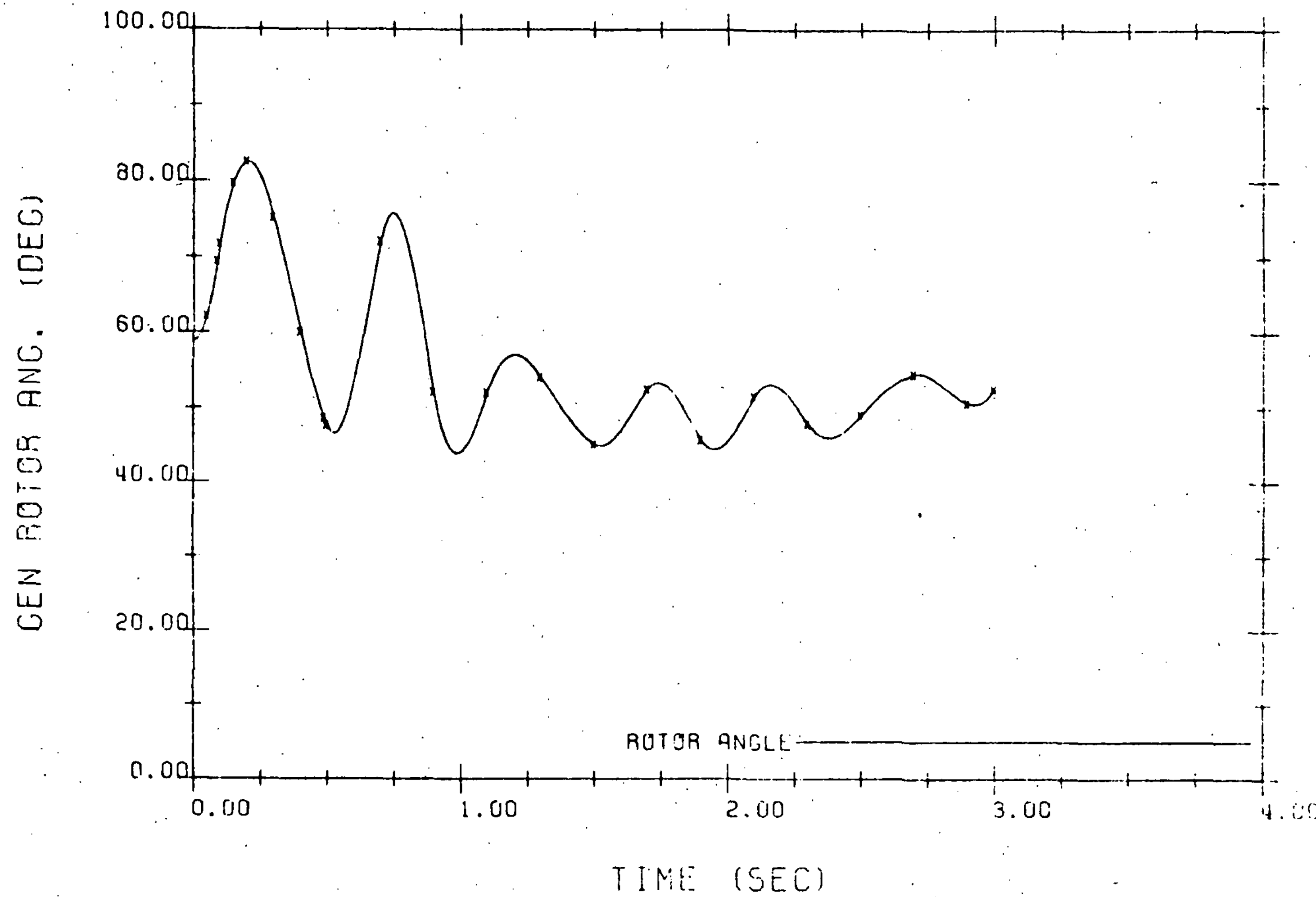


Figure 8-6a. Case 131-4R

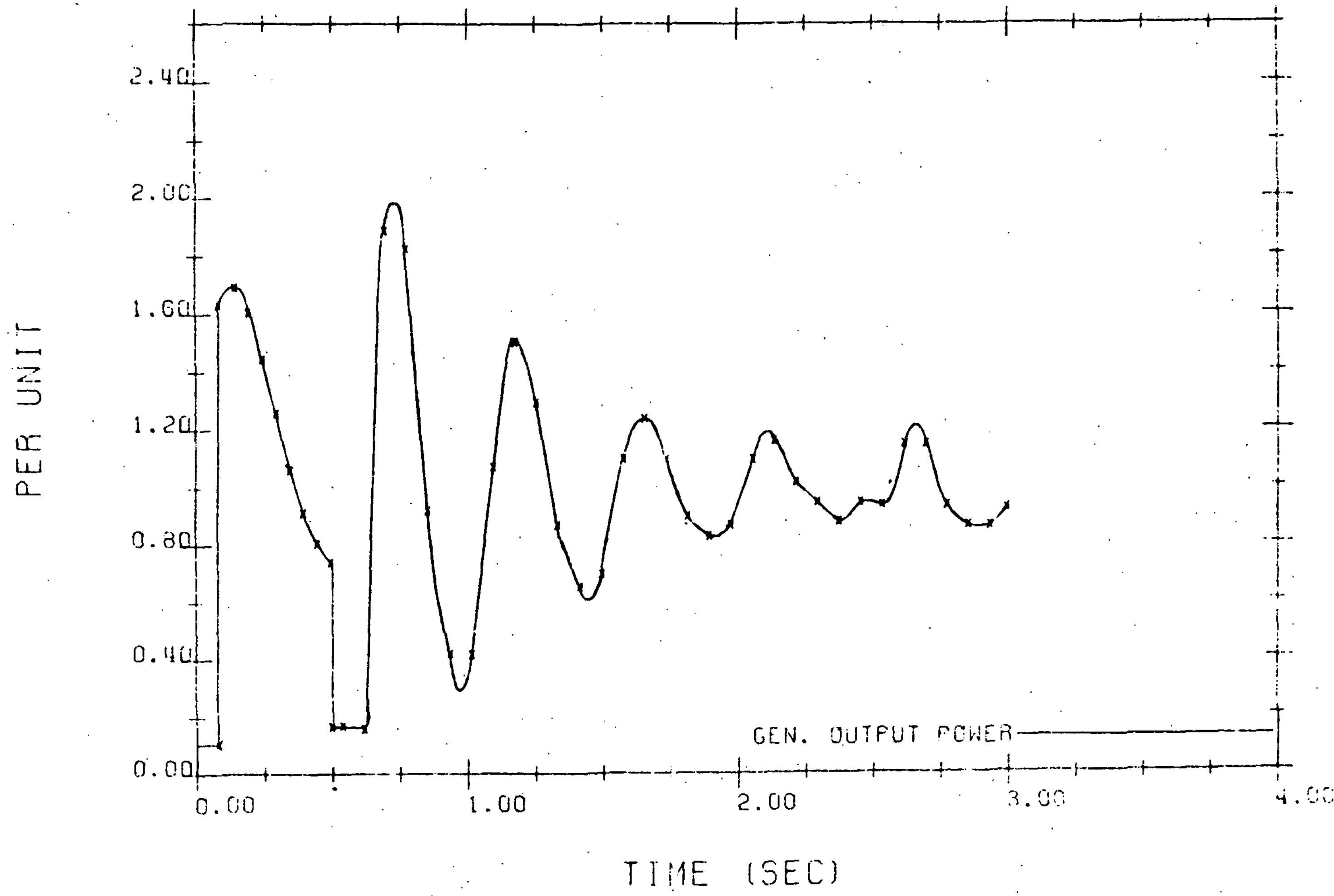


Figure 8-6b. Case 131-4R

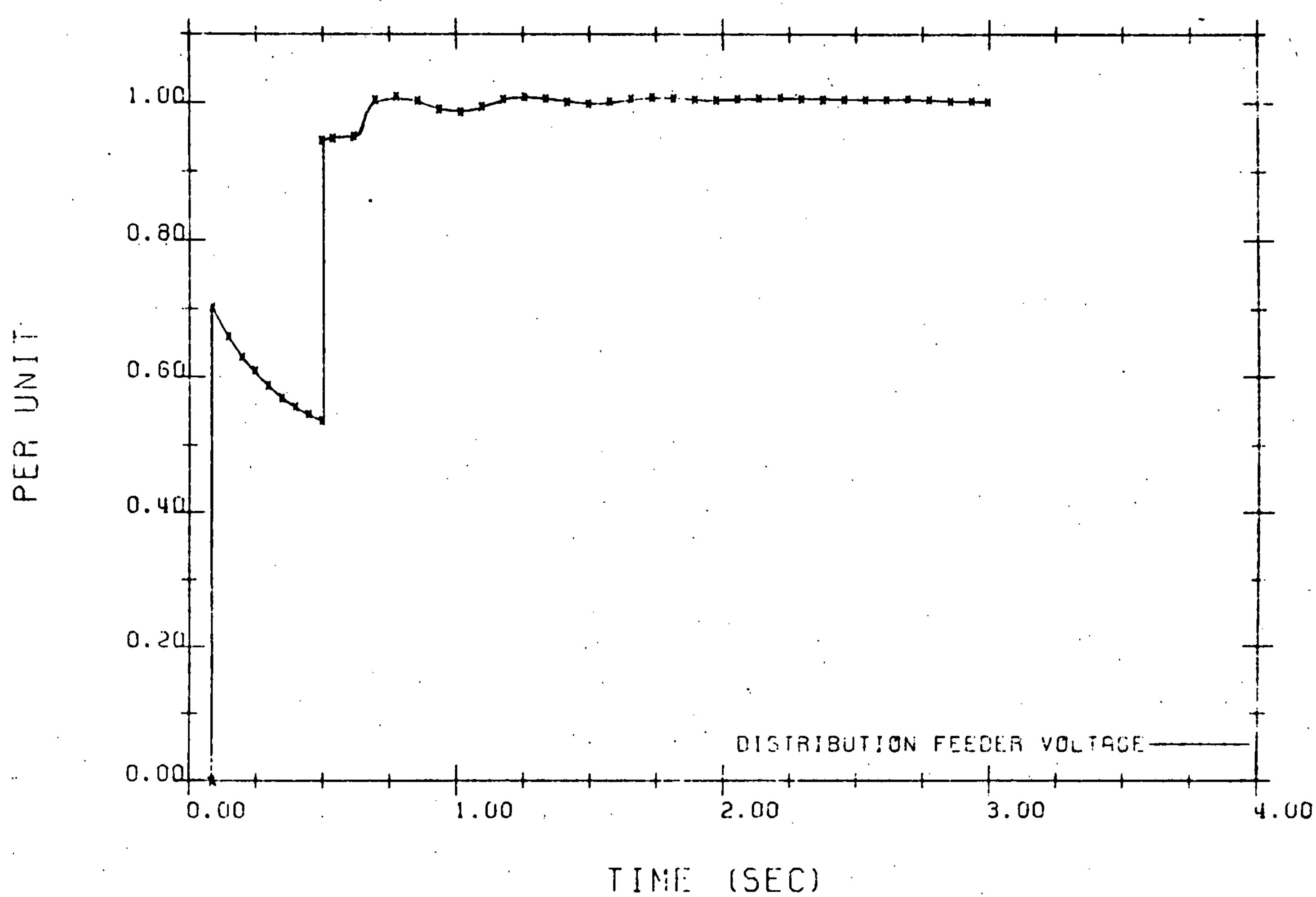


Figure 8-6c. Case 131-4R

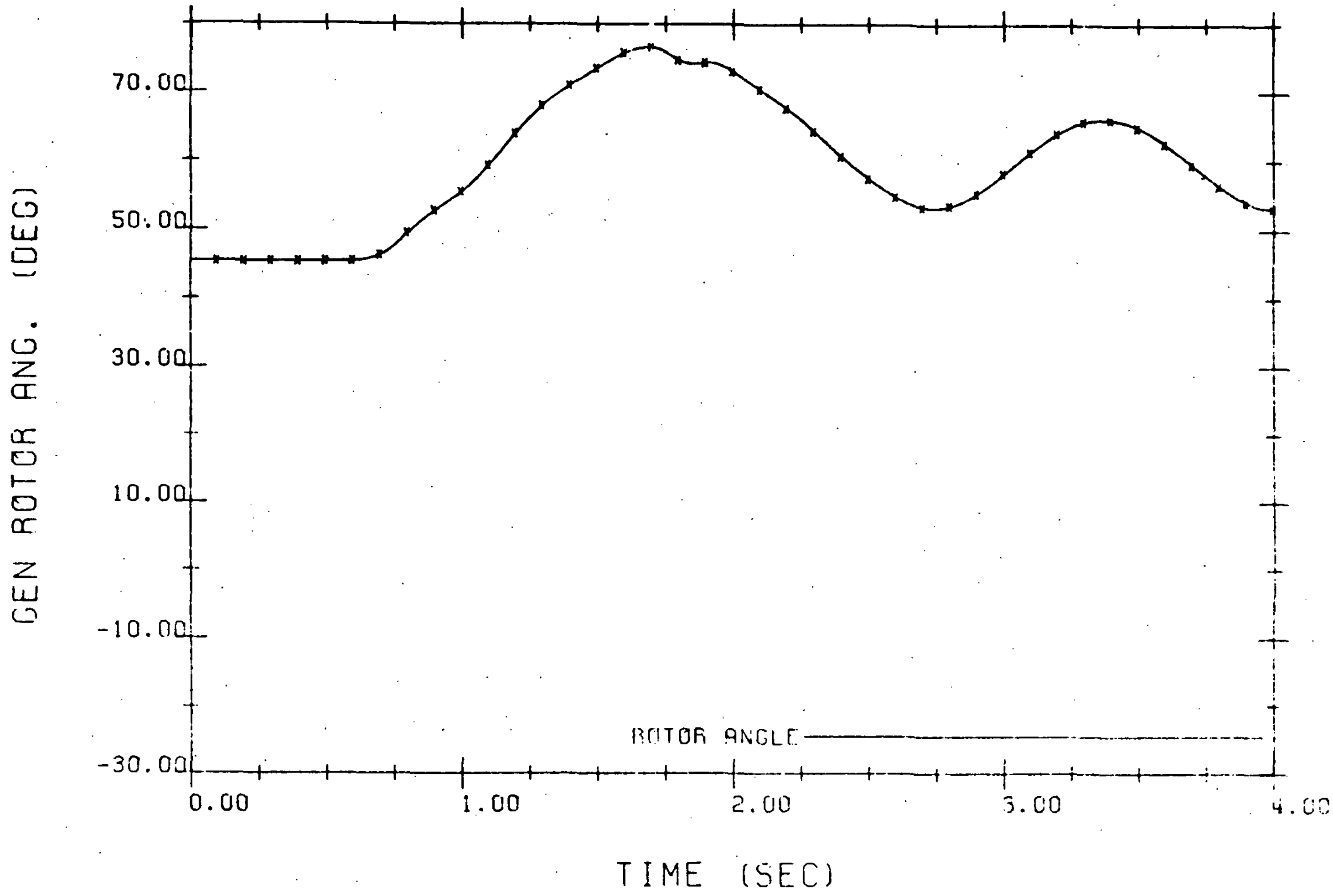


Figure 8-7a. Case 10-1

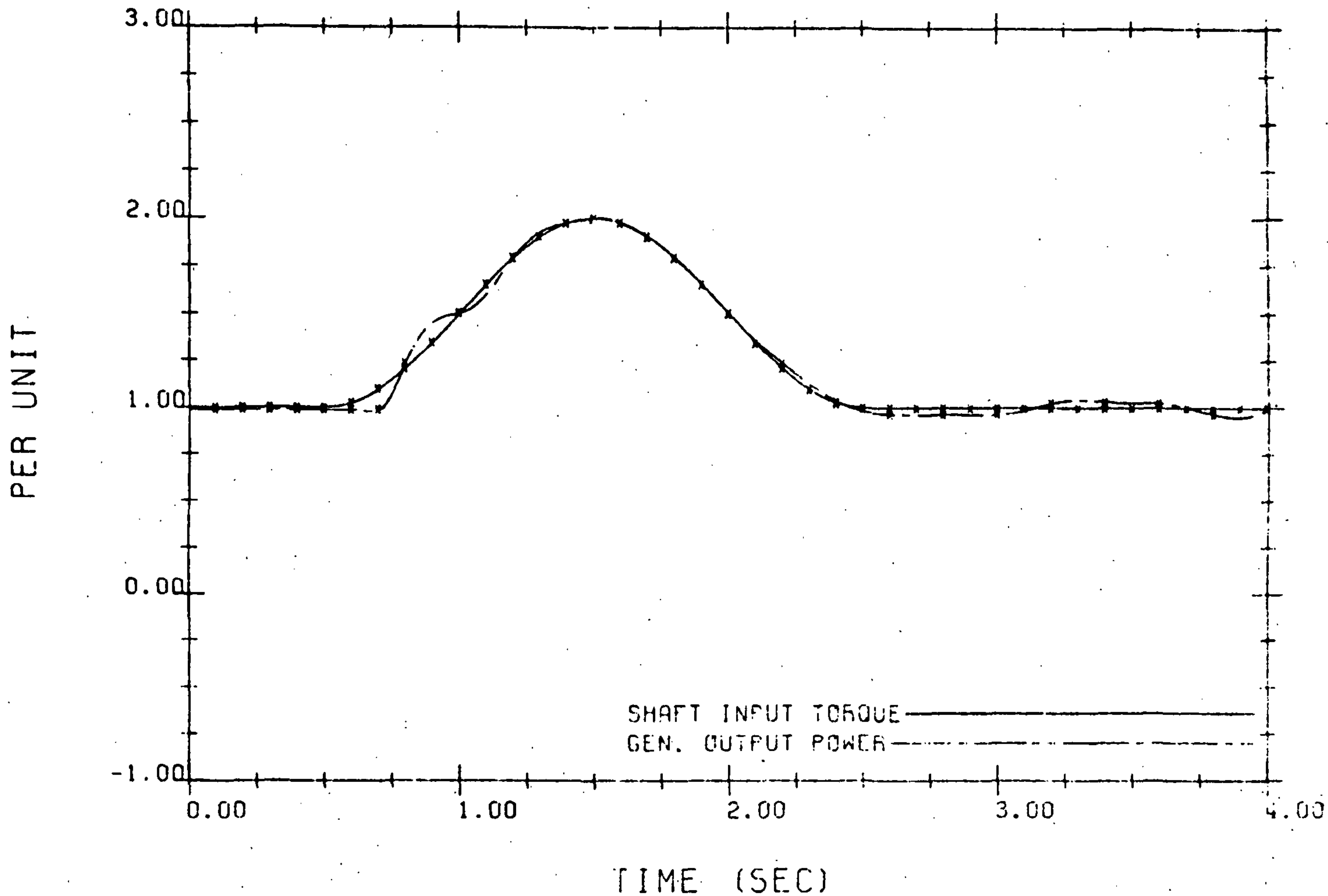


Figure 8-7b. Case 10-1

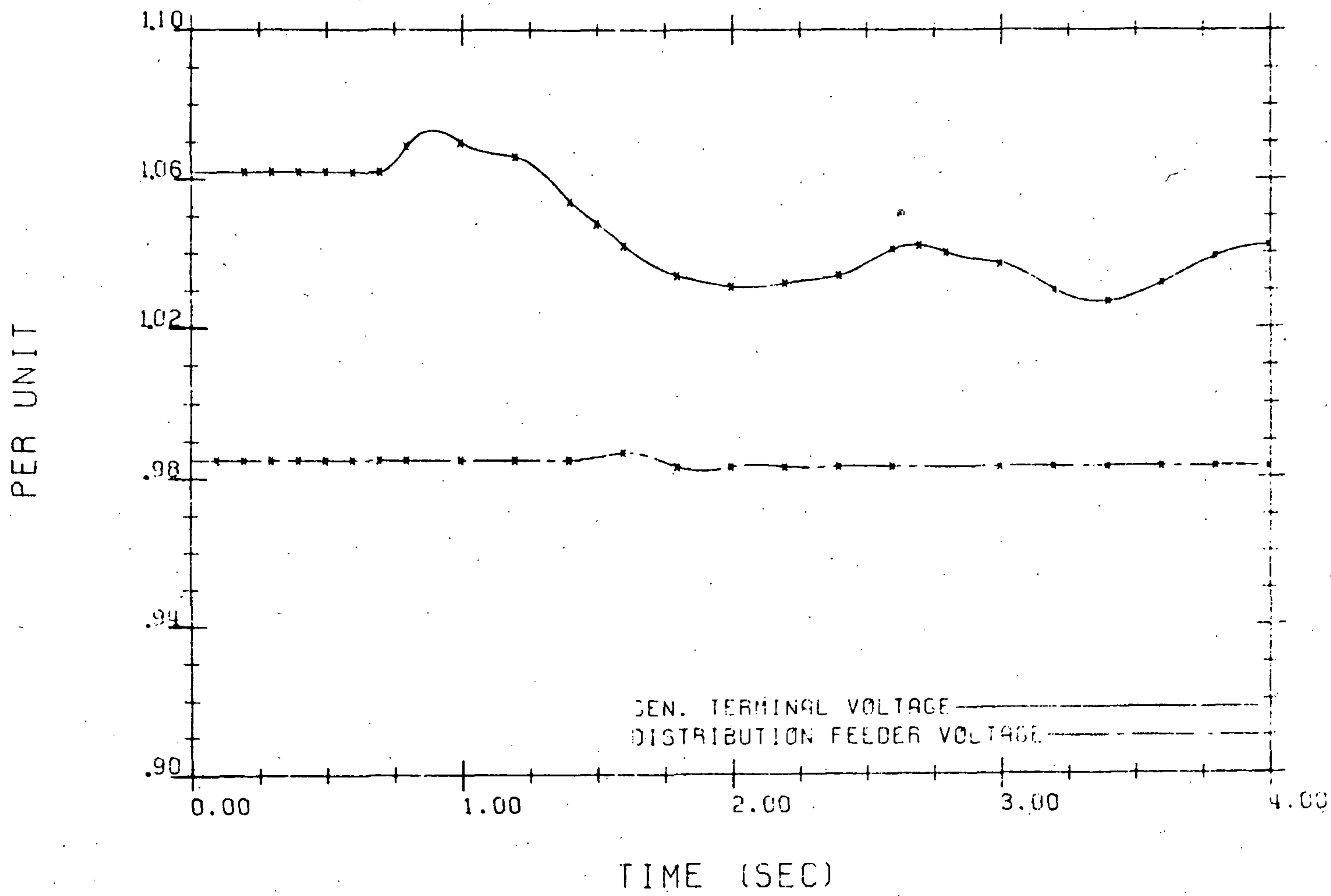


Figure 8-7c. Case 10-1

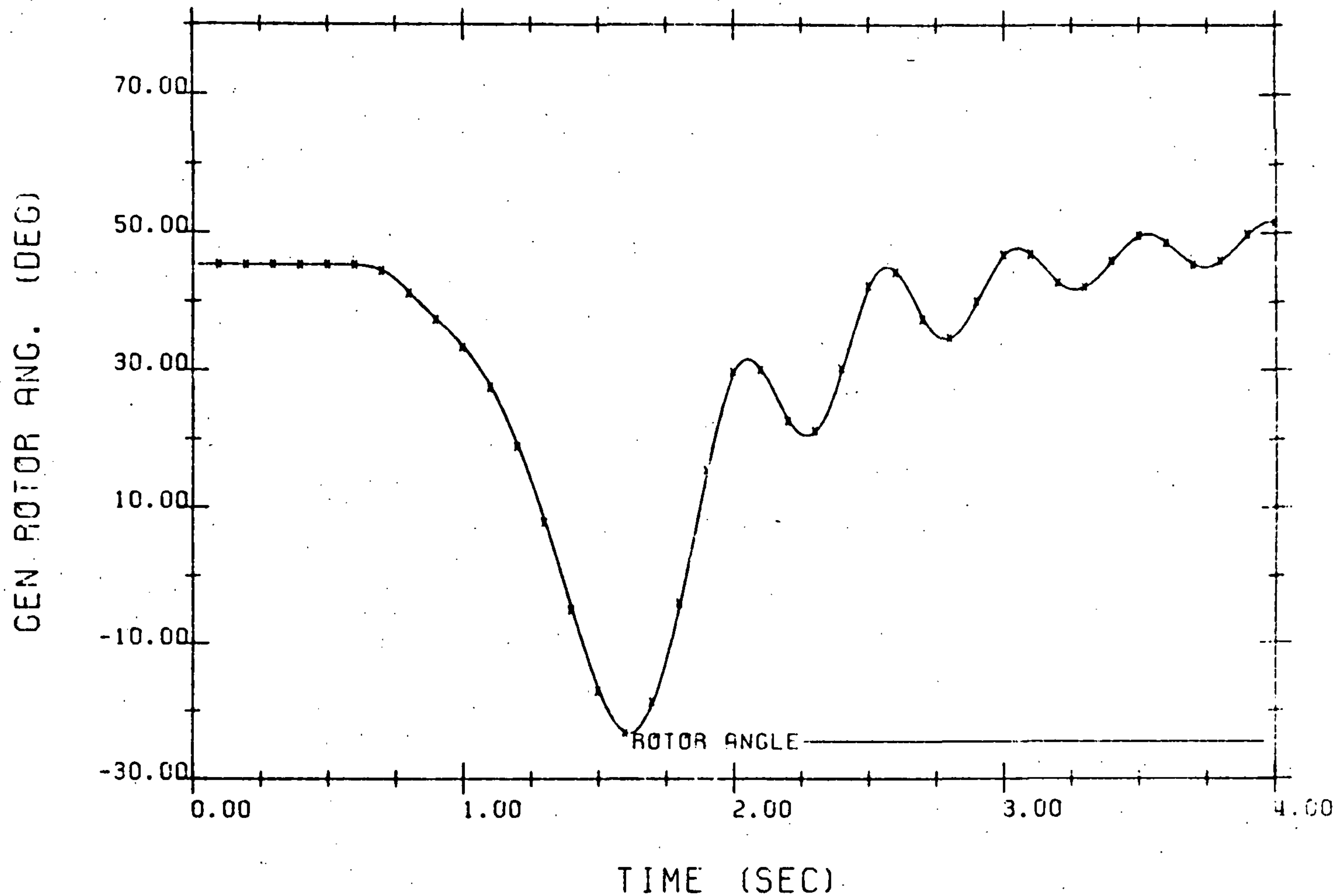


Figure 8-8a. Case 10-2

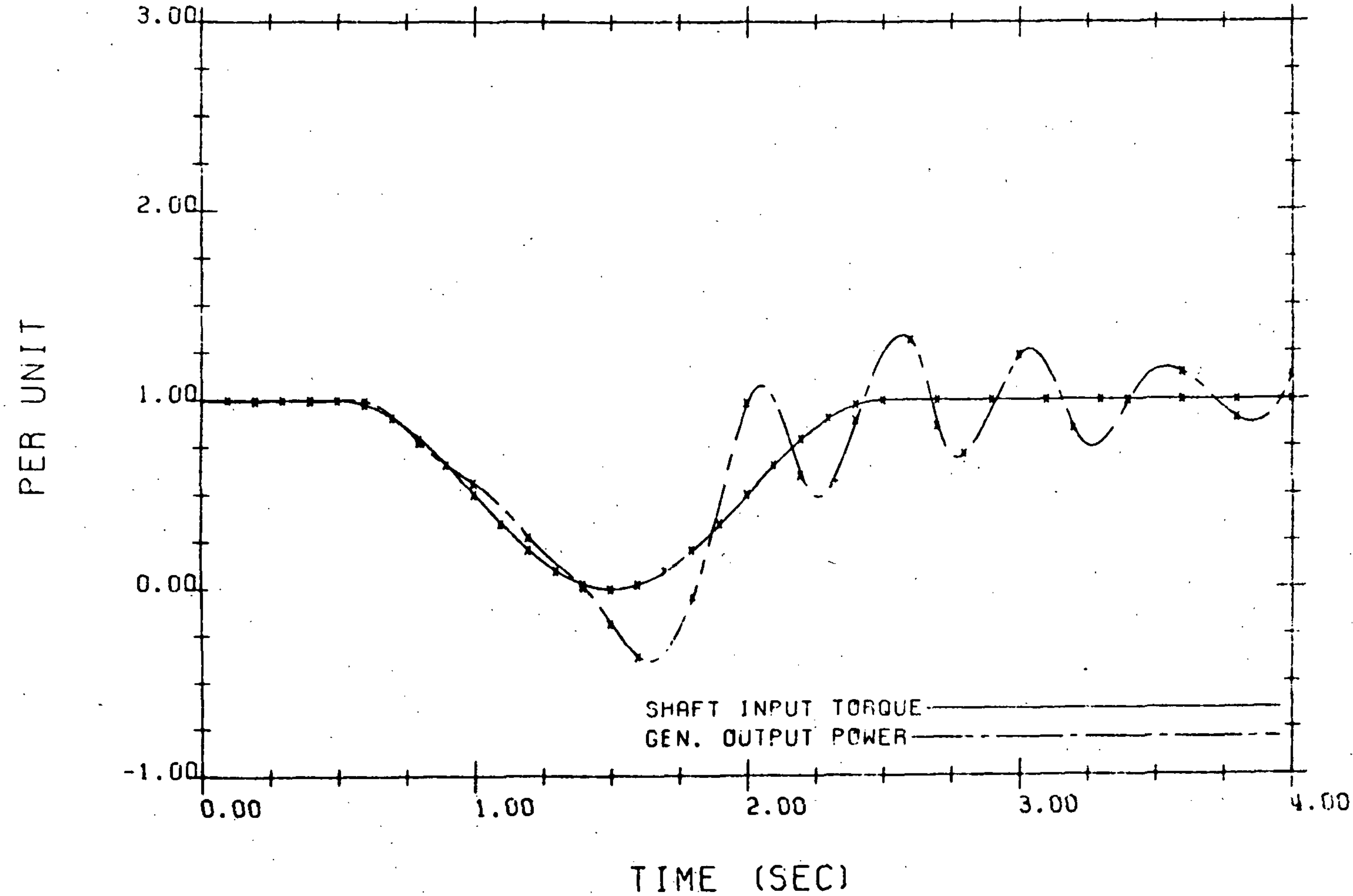


Figure 8-8b. Case 10-2

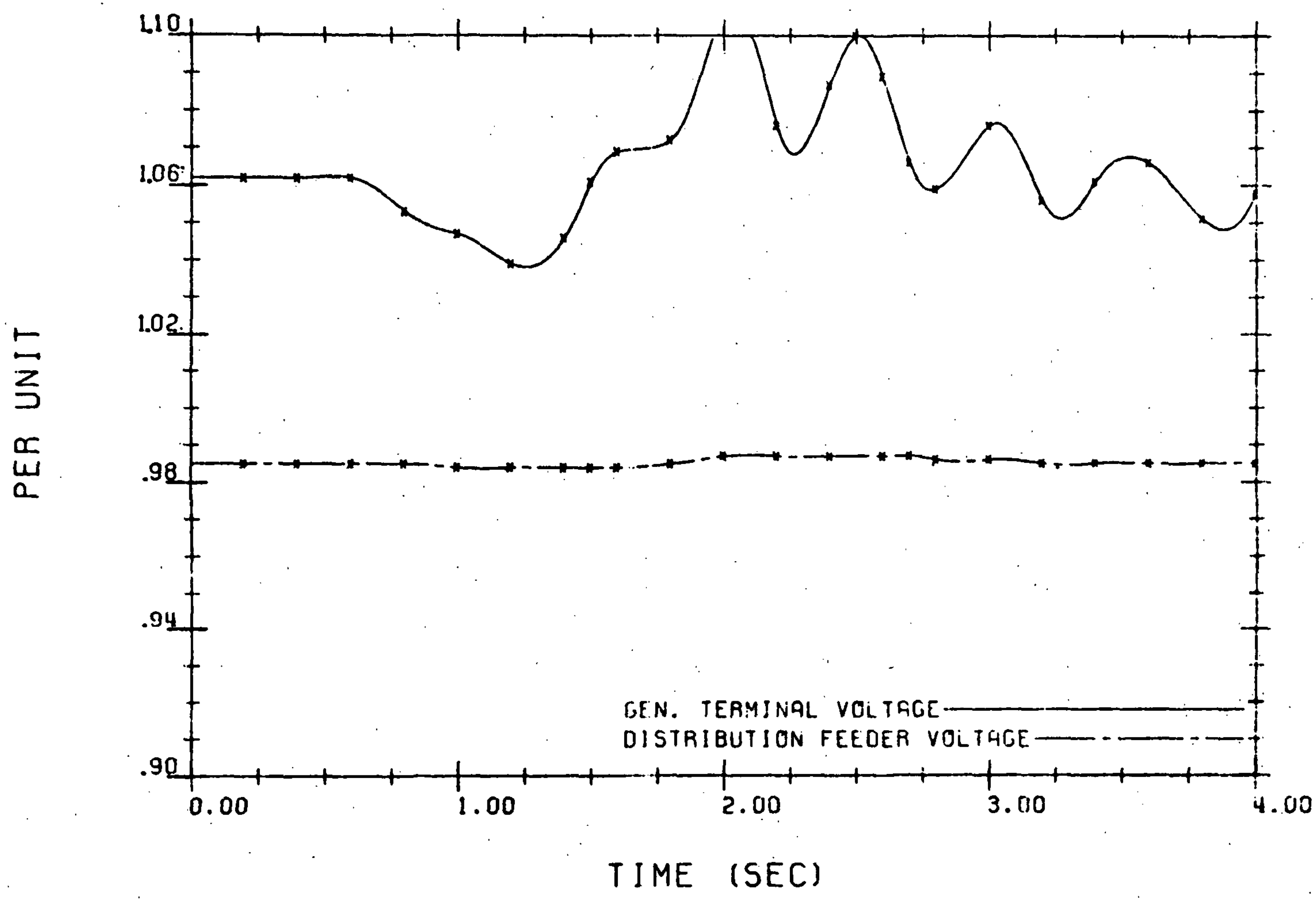


Figure 8-8c. Case 10-2

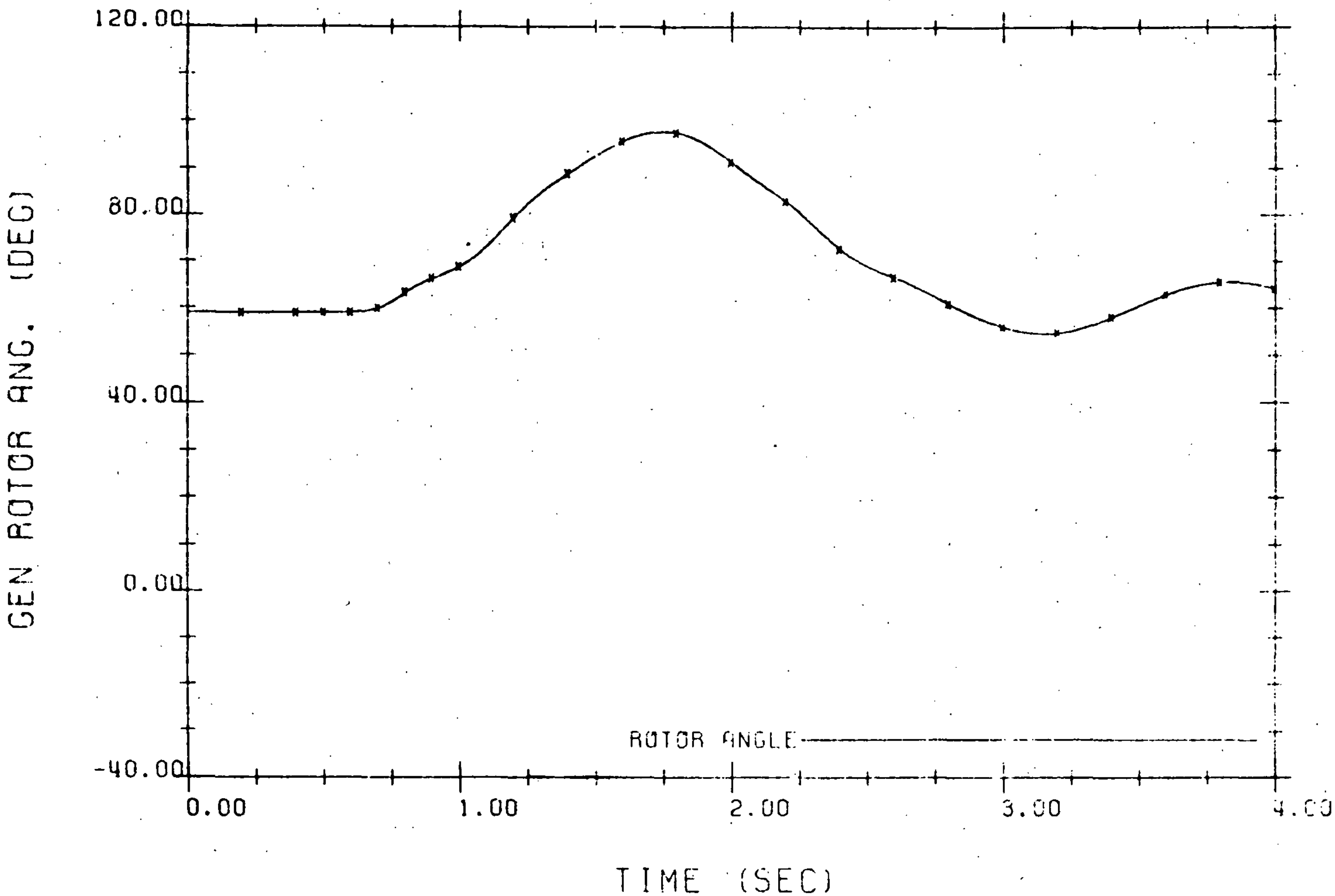


Figure 8-9a. Case 139-4

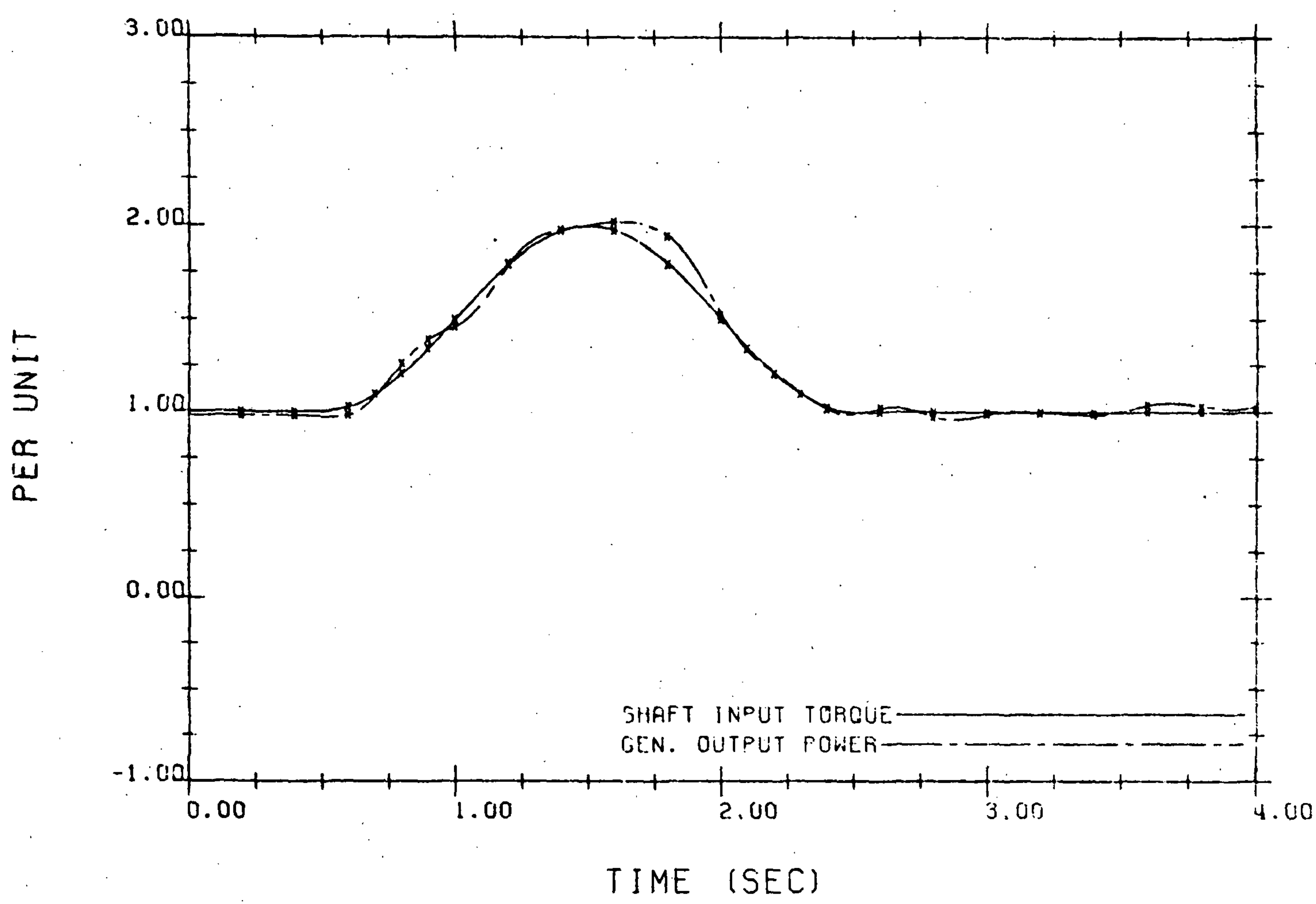


Figure 8-9b. Case 139-4

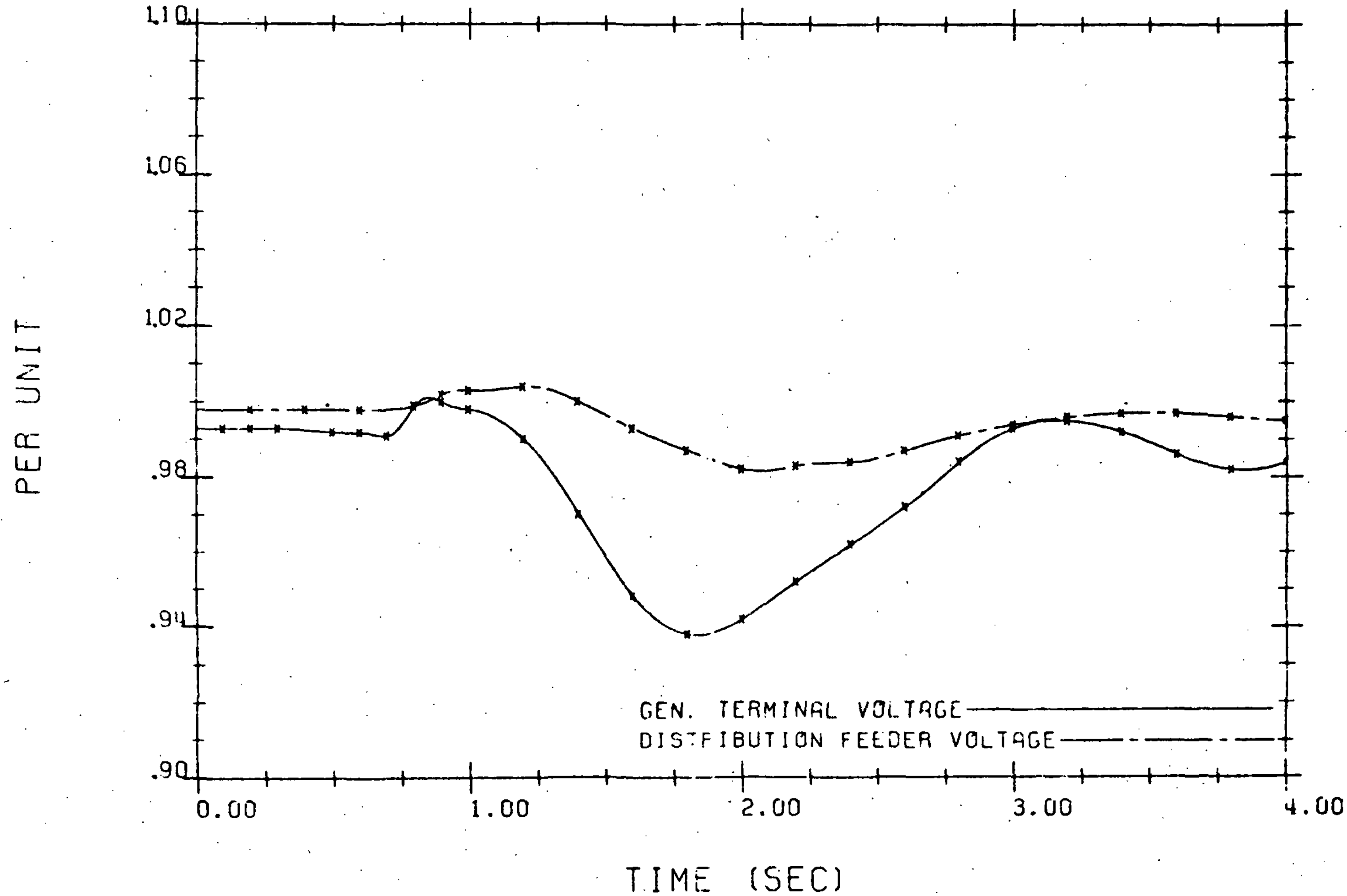


Figure 8-9c. Case 139-4

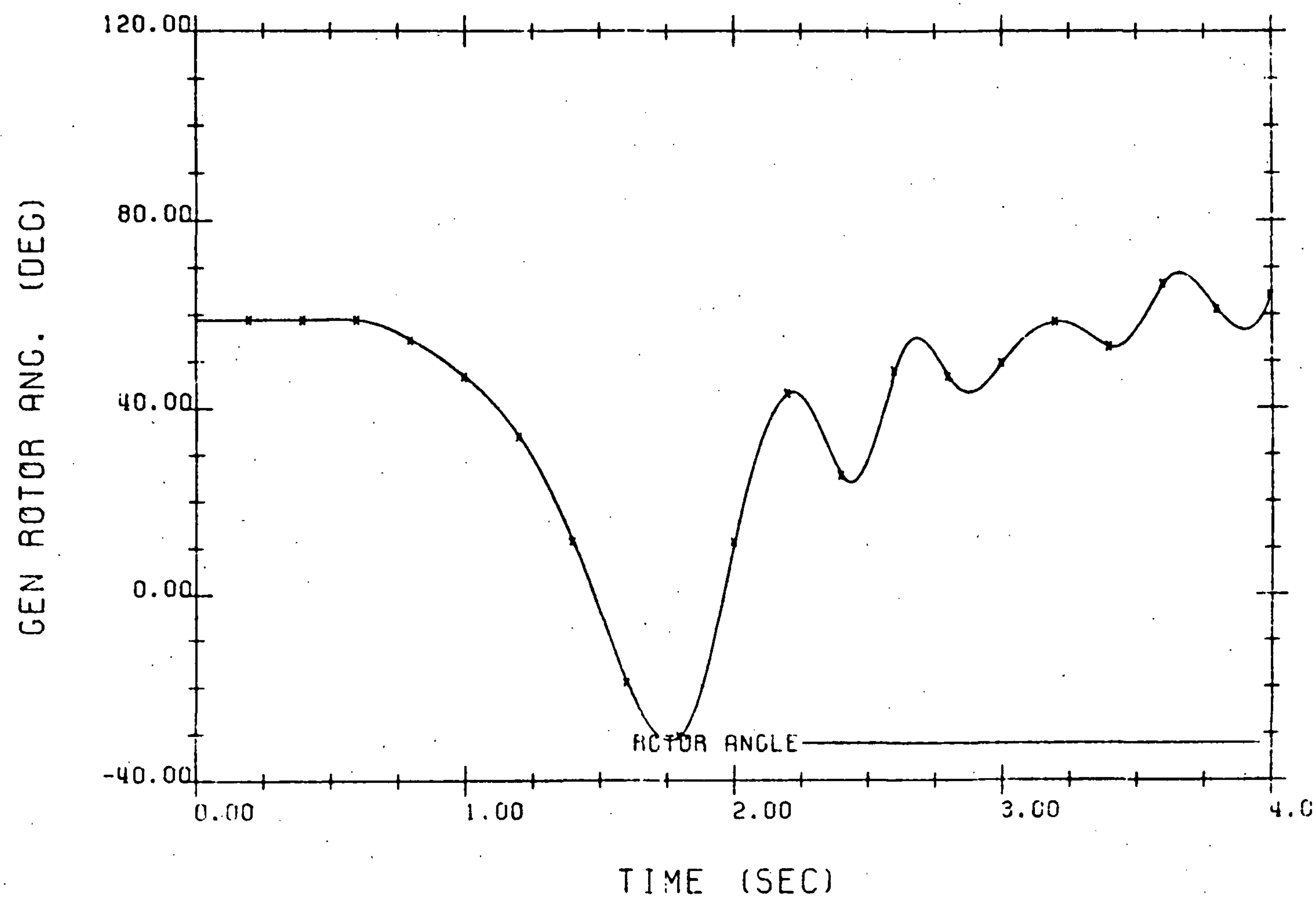


Figure 8-10a. Case 139-5

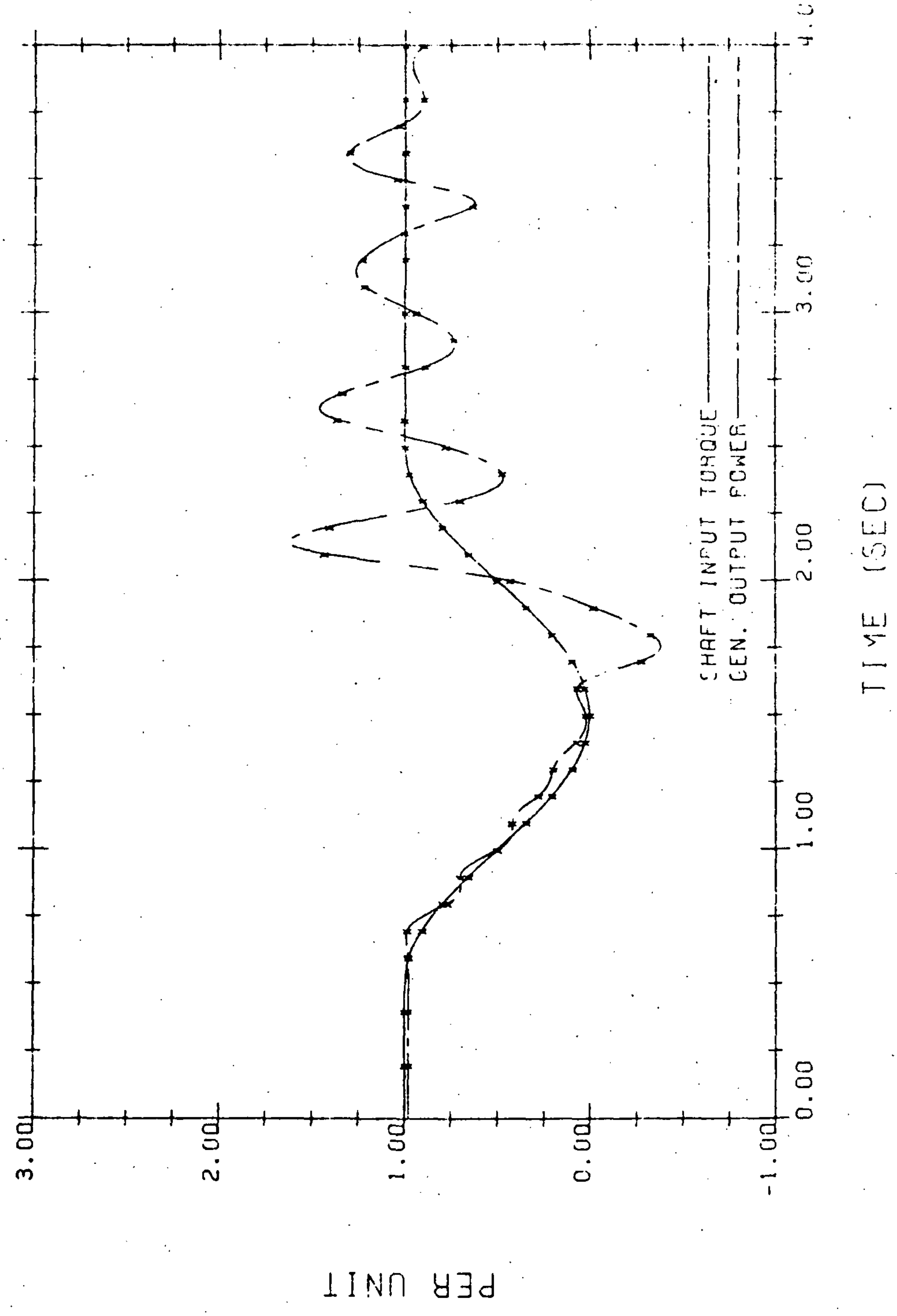


Figure 8-10b. Case 139-5

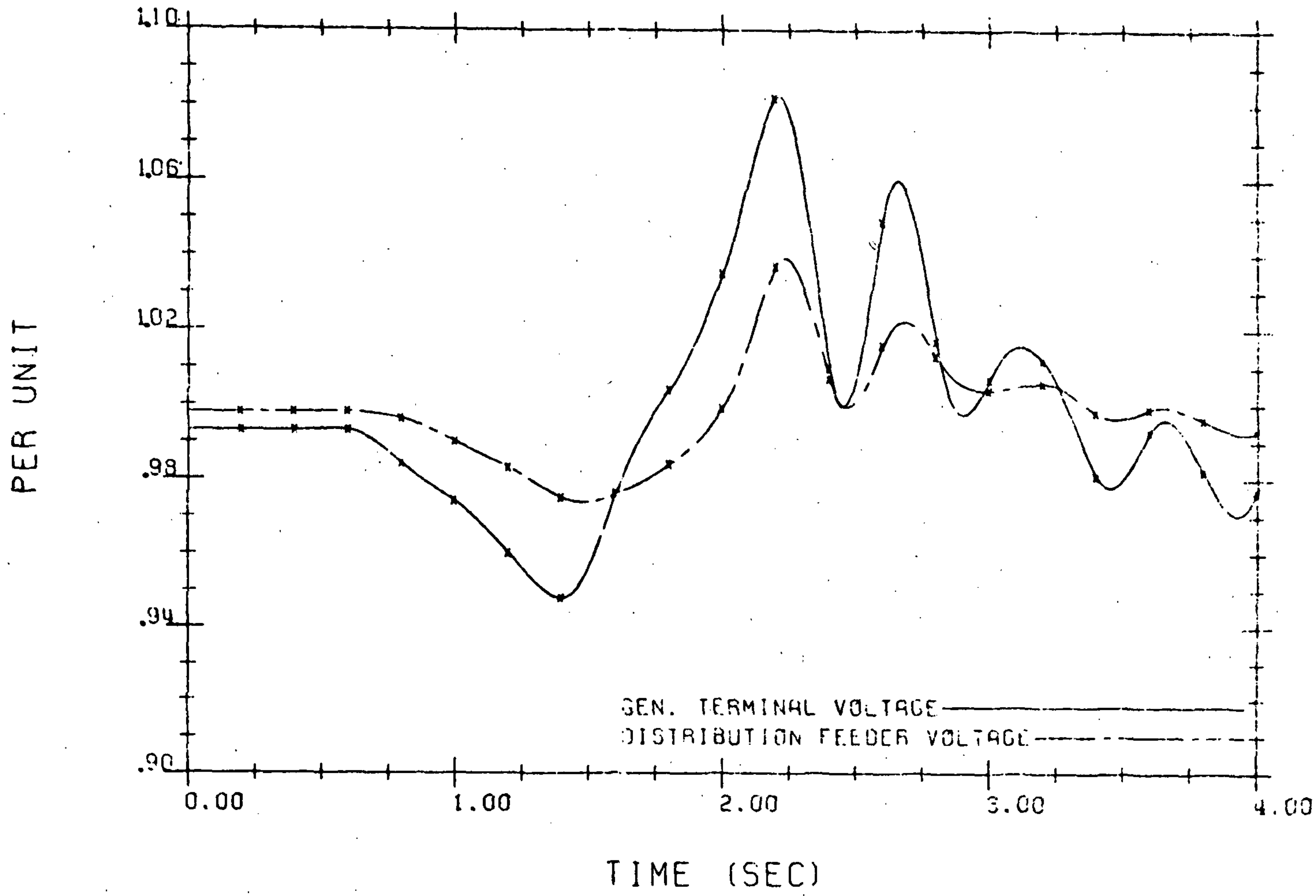


Figure 8-10c. Case 139-5

If the wind generator is located at or near the end of a load-carrying distribution feeder, detection of faults on that feeder, remote from the wind generator, may be difficult. Special relaying or communication channels may be required in order to coordinate the tripping of the WGS circuit breaker with the clearing at the substation end of the feeder. In some cases, the fault can actually increase the load on the generator, as illustrated in Figure 8-5b.

Wind gusts, up to the maximum design intensity, are not likely to cause the synchronous generator to pull out of synchronism with the system, regardless of the location of the wind generator on the distribution feeder (for the particular feeder studied herein, as shown in Figure 8-3). The limiting action of the blade pitch control and the ability of the generator output to "follow" the gust prevent major excursions in generator speed. Figures 8-7 through 8-10 show generator response to severe wind gusts starting from full-rated generator output. Note that in the case of severe, decreasing gusts (Figures 8-8 and 8-10), the generator output becomes negative, indicating power being drawn from the distribution system for a short period of time until the transient subsides.

Voltage variations on the distribution system, due to wind gusting conditions, are more severe if the synchronous generator is connected to the feeder remote from the source (substation). With the generator connected to the system at the substation, as shown in Figure 8-7c, voltage variations on any feeder supplied from that substation are less than 0.5% for even the most severe wind gust studied. With the generator connected near the end of the feeder (Figure 8-9c), the same wind gust causes distribution voltage to vary 2.2%. The acceptability of these more severe voltage variations depends upon their frequency of occurrence and upon the standard established by the particular utility company to which the generator is connected. Ultimately, optimization of the design of the pitch control system and the generator regulator could reduce these variations.

Large variations in voltage and power, which are especially evident when the WGS is connected near the end of the feeder and disturbed with decreasing wind gusts (Figures 8-10a, b and c), point out the need for a more detailed transient analysis for each specific installation. A more detailed analysis would answer such questions as: What is the optimum pitch control system/field regulator configuration to minimize these disturbances? Are pitch rates faster than the selected 5 deg/sec rates necessary? Is a hydraulic slip coupling required between the generator and the gearbox? This analysis should include, as a minimum, the following additional features over and above the analysis performed during the preliminary design:

1. The analysis should include the ability to separate the rotor blade inertia from the generator rotor inertia and should couple these two inertias through the equivalent spring constant and backlash of the shaft and gearbox. These inertias were lumped together in the present analysis; however, the existence of negative power flow during some of the larger wind gusts and fault transients means that backlash and shaft windup should not be ignored in a detailed analysis.

2. A dynamic model of the rotor blades and rotor pitch controls should be synthesized and coupled with the above gearbox/generator/network model. This must include the option of changing various control system gains, rotor parameters, etc., to evaluate the overall effects on system stability. RPM regulation should be investigated as a possible method of reducing or minimizing the effects of decreasing wind gusts on system voltage and power flow when the generator is connected to the network.

3. The model of the distribution feeder network must be tailored to a specific installation site, rather than a typical feeder, such as was used in the present analysis.

4. The transfer function of the generator voltage regulator, although included in the present analysis, should be studied in greater detail to evaluate its effectiveness in reducing or limiting voltage variations in each specific installation.

5. A transient analysis computer program should be obtained for the induction generator and it should also be analyzed with a utility network, using the same type of model described above for the synchronous machine (except without the voltage regulator).

Transient behavior of the 1500 KW generator connected to the utility network (Figures 8-5 through 8-10) indicates natural frequencies of approximately .77 Hz and 2 Hz, based on the present simplified model. These frequencies remain approximately constant, regardless of whether the WGS is located near the substation or out near the end of the distribution feeder. These frequencies must not coincide with any natural frequencies of the rotor blades or natural excitation frequencies of the rotor system in order to prevent system resonance problems. The significant natural frequencies of the rotor system are given below for the 1500 KW system, and it can be seen that the margins are adequate:

ROTOR NATURAL FREQUENCIES (1500 KW)

MODE	NATURAL FREQUENCY
First Flatwise	1.43 Hz
First Edgewise	2.67 Hz
Second Flatwise	3.90 Hz

ROTOR EXCITATION FREQUENCIES (1500 KW)

MODE	EXCITATION FREQUENCY
1/rev	.573 Hz
2/rev	1.147 Hz
3/rev	1.719 Hz

A similar check would be required for the 500 KW system, if it is interfaced with a utility network. The generator/network natural frequencies would also be rechecked as part of a detail WGS design, using a detailed transient model of the WGS and the actual electrical network installation.

8.5.8 Weight and Cost

A summary of the weight and cost analysis results for the Electrical Subsystem is given in Tables 8-7 and 8-8, respectively. The costs are based on a quantity of 1000 units. The switchgear costs include an allowance for the slip ring at the nacelle. They also include an allowance for a ground fault detector for the distribution line on the transformer high side which was recommended by Northeast Utilities. The instrument and control panel cost includes revenue metering equipment for the station service power and the cost of fire alarm equipment. It also includes the fault annunciator panel for all the WGS subsystems.

For the induction generator, the switchgear and transformer costs include the cost of power factor correction capacitors and associated capacitor switches.

8.6 Subsystem Adaptability

The electrical generation and protective relaying system described herein is applicable to any wind-driven electric power generation system, using an induction or a synchronous generator connected to a utility at the distribution network level. For isolated systems not connected to the utility network, some simplification would most likely be possible, especially in the protective relaying and switchgear. The configuration presented should remain essentially the same over a power range from 50 KW to 3000 KW, if connection to the utility network is provided.

Standard generators are only available at discrete values of KW ratings. If it is desired to remain with standard size generators, any changes in WGS system rating should take this into account.

8.7 Conclusions and Recommendations

8.7.1 Conclusions

1. Variable input shaft speed systems which require conversion equipment to produce constant frequency AC power do not offer any advantage from an overall cost or efficiency standpoint in large WGS applications. In addition, variable rotor speed is not practical for large WGS rotors because of possible resonance problems, as discussed in Section 4, Rotor Subsystem.

2. Standard, readily available electrical equipment is satisfactory, even for specialized WGS applications.

3. Either induction or synchronous generators are acceptable, depending upon the preference of the operating utility.

TABLE 8-7. ELECTRICAL SUBSYSTEMS WEIGHT SUMMARY, kg (lb)

	500 KW		1500 KW	
	INDUCTION	SYNCHRONOUS	INDUCTION	SYNCHRONOUS
Generator	1723 (3800)	2426 (5350)	3515 (7750)	6122 (13500)
Switchgear and Transformers	4190 (9238)	3813 (8408)	5704 (12578)	5337 (11768)
Protective Relays and Generator Controls	1349 (2975)	1406 (3100)	1349 (2975)	1406 (3100)
Service and Emergency Power	429 (946)	429 (946)	429 (946)	429 (946)
Instrument and Control Panel	517 (1141)	518 (1143)	517 (1141)	518 (1143)
TOTALS	8208 (18100)	8592 (18947)	11515 (25390)	13813 (30457)

TABLE 8-8. ELECTRICAL SUBSYSTEMS COST SUMMARY
($ 000)

	500 KW		1500 KW	
	INDUCTION	SYNCHRONOUS	INDUCTION	SYNCHRONOUS
Generator	8.0	11.0	18.1	28.6
Switchgear and Transformers	14.3	11.6	16.8	13.9
Protective Relays and Generator Controls	10.9	13.6	10.9	13.6
Service and Emergency Power	1.6	1.6	1.7	1.7
Instrument and Control Panel	3.8	3.9	3.8	3.9
Subsystem Integration	2.1	2.1	3.1	3.1
TOTALS	40.70	43.8	54.4	64.8

4. For the typical distribution feeder analyzed, wind gusts are not likely to cause the synchronous generator to pull out of synchronism with the system.

5. Electrical faults within the WGS equipment should initiate shutdown and lockout of the WGS until repaired.

6. For external faults which trip the main breaker, the WGS should be allowed to re-synchronize after a 1 minute delay if the line is re-energized.

7. The wind generator should be disconnected from the distribution feeder for any disturbance which causes the normal supply to the feeder to open.

8.7.2 Recommendations

1. Constant speed synchronous or induction generators should be used for large utility based WGS installations.

2. A detailed WGS/network transient analysis should be performed, as described in 8.5.7, for each specific WGS installation. Results of this analysis and the standard practices of the using utility should determine several of the final WGS design features, including the main breaker ratings, the selection and setting of the protective relays, the choice of generator and the regulator characteristics.

3. A lightning protection consultant should be involved early in the detail design effort to guide the mechanical and electrical design of the WGS as it evolves.

4. Use of gasoline or diesel generators to provide auxilary or emergency power at the WGS site is not recommended.

5. The main stepup transformer should be isolated from the feeder by the WGS breaker when the WGS is shut down.

6. It is recommended that the synchronous generator not be used to provide power factor correction in the absence of wind.

9.0 UTILITY APPLICATIONS AND OPERATIONAL REQUIREMENTS

For wind generators to assume a significant role in the production of electrical energy for the future, they must be economically competitive with other forms of energy production and must attain acceptance by the utility industry and the public. Factors which will be important to the industry's acceptance of the wind generator, beyond its economic feasibility, are the adaptability of these systems to existing utility networks and their potential for safe and reliable operation. Public acceptance will hinge mainly on safety and environmental considerations. One task of this program was to examine some of these issues.

9.1 Purpose and Scope of Analyses

Many of the economic and operational issues confronting the development and successful application of wind generators are complex, involving such diverse and interdependent considerations as the financial structure and operating policies of the utility companies, Government regulations, the world fuel situation and the environment. A number of independent studies, most being conducted under the auspices of ERDA's wind energy program, will study these problems in depth as research in wind energy systems continues.

In order to establish tentative cost goals for wind generator systems, however, and to guide the WGS preliminary design for maintenance and safety, some study of WGS economics and operations was necessary in the present program. The scope of the task did not allow extensive study in these areas, but did develop the basic information needed to carry out the parametric analyses and preliminary design. Much of the knowledge on these subjects came from one source, the Northeast Utilities Company, Kaman's principal utility consultant on the program, and as such is representative of that type of utility.

There were two basic areas of study: (1) operational and institutional issues, including maintenance and safety, licensing requirements and environmental impact; and, (2) wind generator applications, covering the economics of wind energy production. Since the issues in both areas involve the use of the WGS by the utility industry, it was from this source that information was primarily sought.

9.2 Sources of Information

Northeast Utilities supported Kaman throughout the program and was especially helpful in evaluations of utility economics and operations. A series of meetings were held with personnel from the various operating divisions of Northeast to discuss the wind generator program and to obtain their views on the requirements that would have to be met in the design and operation of these systems. In addition to providing guidance in such areas as maintenance and safety and environmental impact, Northeast was able to supply various cost and planning factors for use in the parametric model and cost analyses.

Another important source of data for the study of economic and operational requirements was Colorado Springs Public Utilities (CSPU). CSPU not only presented a much different geographical environment for consideration of wind energy systems but also offered a significant contrast with Northeast Utilities, in terms of size, market, plant and financial structure. These differences caused some issues to be evaluated quite differently by the two companies and added another perspective to the study. The two companies are described briefly below.

Colorado Springs Public Utilities - CSPU is a moderate sized municipal utility providing electric, gas, water and waste water services to 82,000 customers in the Colorado Springs, Colorado, area. Total assets are approximately $300 million. Electrical power sales for 1974 produced revenues of $23.5 million. Customers served by CSPU are primarily residential, small industrial users and military installations in the area. Approximately 70% of total sales are within the city limits of Colorado Springs.

CSPU has a current generating capacity of 320 MW, of which 314 MW is gas/oil or coal-fired steam and 6 MW is peaking hydro. All of the steam units are being converted to coal. CSPU projects a load growth of 7.5% per year and will add approximately 1000 MW of generating capacity to their system over the next 13 years. All of this will be coal-fired steam, except for 60 MW of gas turbine peaking power. The utility is a member of the Rocky Mountain Power Pool.

Northeast Utilities Company - Northeast Utilities is a large electric utility serving approximately 1,000,000 residential, industrial and commercial customers in southern New England. The corporation has assets of approximately $2.5 billion. Sales revenues for 1974 were approximately $650 million. Northeast has a total plant capacity of 5500 MW comprised mainly of nuclear (1100 MW) and oil-fired steam (2800 MW) units. Pumped hydro accounts for another 1000 MW capacity with the remaining 600 MW provided by gas turbines. Average load growth is projected at 5.5% per year and expansion will be mainly in the form of large nuclear plants.

Guidance on the regulatory aspects of wind generator systems was obtained from three Government agencies in Connecticut. The Connecticut Department of Environmental Protection was consulted in the areas of environmental impact and licensing requirements. Regulations pertaining to air traffic safety were obtained from the Connecticut Aeronautics Commission. The Hartford, Connecticut, office of OSHA was contacted for information on safety standards.

9.3 Utility Cost Analysis

An analysis was made of the costs associated with the acquisition and operation of wind generators in an electrical utility system. Consistent with industry practice in the analysis of capital investments, two costs were considered: (1) the direct capital cost, expressed in $/KW, for the wind generator system and its site and supporting facilities; and, (2) the average cost of the energy produced over the life of the system, expressed in ¢/KWh, accounting for recovery of the capital investment, interest, taxes and operations and maintenance expense.

The cost analysis was based on standard utility cost procedures and on the assumption of large scale production of wind generator systems. Because most of the cost and financial planning factors used in the analysis were provided by Northeast Utilities, the evaluation tends to be particularly representative of a large investor-owned utility situated in the Northeast. However, the results are representative of many utilities throughout the United States.

The description of the parametric computer model in Section 3 of this report contains the equations used in the calculation of direct capital costs and energy costs. The following paragraphs explain the basis for these equations.

9.3.1 Direct Capital Costs and Carrying Charges

The direct capital cost of the wind generator system is that of procuring, transporting, erecting and readying the system on a utility site, including the costs of acquiring and preparing the site and providing necessary facilities and security.

The parametric model description in Section 3 enumerates these costs and the basis on which they were estimated. Not included in the cost analysis are those direct costs which will tend to be highly variable, depending on the particular utility and the siting of the wind generator. These include the costs of access roads and transmission lines, and the costs borne by the utility during the construction period. These latter costs, generally termed "owner's costs", include allowances for interest on short term borrowing, taxes during construction and contingencies for delays, cost overruns, etc.

Northeast Utilities uses, for financial planning purposes, a "carrying charge" rate which expresses the average annual cost of a capital investment, excluding operations and maintenance, over its anticipated useful life. The carrying charges, which cover depreciation, interest charges, return on equity and taxes are calculated by Northeast Utilities using a financial planning model. Total expenses over the life of the investment are converted into an average annual premium or carrying charge, taking into account the time value of money.

The assumptions and financial planning factors supplied by NASA for the cost analysis are contained in Table 9-1. These factors were used in Northeast Utilities' financial planning model to calculate an annual carrying charge rate for the wind generator system. The 14.7% rate calculated by the model was rounded to 15% for use in the parametric modeling and cost analysis. For comparison, an independent analysis by Northeast Utilities is shown in Figure 9-1.

TABLE 9-1. ECONOMIC PLANNING FACTORS AND ASSUMPTIONS*

PARAMETER	FACTOR/ASSUMPTION
System Useful Life	30 Years
Financing	50% Debt, 50% Equity
Return on Investment	9% on Debt, 11.5% on Equity
Depreciation	Straight Line Over 30 Years
Corporate Tax Rate	48%

*Supplied by NASA

- DEPRECIATION
- COST OF MONEY
  - DEBT SERVICE
  - STOCK DIVIDENDS
- TAXES
  - INCOME (STATE & FEDERAL)
  - PROPERTY

CARRYING CHARGE RATE

[% OF INVESTED CAPITAL/YEAR]

15%

TYPICAL

- BOOK LIFE: 30 yrs.
- TAX LIFE: 22½ yrs.
- BOOK DEPRECIATION: S.L.
- TAX DEPRECIATION: S.Y.D.
- R.O.I. - 10%
- SALVAGE RATE: 0.0%
- CAPITAL STRUCTURE & COSTS

  52% Debt @ 7.5%
  13% Preferred Stock @ 7.5%
  35% Common Equity @ 14.5%
- FEDERAL TAX RATE: 48%
- STATE TAX RATE: 8%
- PROPERTY TAX RATE: 1½%

Figure 9-1. Northeast Utilities Carrying Charge Analysis.

9.3.2 Operations and Maintenance Costs

Yearly operations and maintenance costs were, with the exception of rotor maintenance, obtained from rates established by Northeast Utilities for comparable plants and equipment. Rotor maintenance costs were estimated from helicopter experience. Annual maintenance costs used in the model, expressed as a percentage of initial capital investment (FPC categories), are given below:

	PERCENT OF INITIAL COST
Rotor	10
Power (Drive, Electrical, Controls)	3.5
Tower	0.5
Site and Facilities	1.0

These rates are considered to cover the cost of labor and materials for routine inspection and servicing, repair and overhaul of system components. A conservative estimate of rotor maintenance cost was assumed for this initial analysis due to the lack of field experience in the WGS environment. The costs of extraordinary expenses, such as the replacement of major components, may be treated as a maintenance expense or may be capitalized, depending upon the accounting procedures of a particular utility. Maintenance policies for the wind generator are discussed later in this section of the report.

Based on the guidance received from Northeast Utilities, yearly operating cost was estimated at 1.5% of direct capital cost. This covers the cost of operating personnel and supervision, and related indirect costs and overheads.

9.4 Operational and Institutional Issues

The operational and institutional issues of safety, maintenance, licensing and environmental impact were examined under the contract. Information on each of these topics came primarily from discussions with personnel at Northeast Utilities. Important contributions were also made by the Colorado Springs Public Utilities and, in the areas of licensing and environmental impact, by the Connecticut Department of Environmental Protection. The Hartford, Connecticut, office of OSHA and the Connecticut Aeronautical Commission were consulted on questions of safety.

9.4.1 Factors Affecting Utility Views of Wind Generators

The two utilities consulted on the prospective use of wind generators on their systems have significant differences in terms of size, market, generating plant, etc., as mentioned before. From the opinions expressed in the course of these interviews, some of the important factors affecting a utility's view of wind generators were evaluated.

Fuel Costs - Fuel costs are the major consideration influencing the future of wind energy systems, for it is this problem that provided the impetus for current research in this and other forms of alternative energy. Despite the shortages and rising costs being experienced by the country as a whole, however, the problem is more acute for some than for others. Those utilities heavily dependent on oil for fuel, especially in areas which rely mainly on foreign imports, are suffering most. Others, whose generating capacity is primarily nuclear or low-sulphur coal, are much less affected.

The primary utility role for wind generators, until major improvements are made in energy storage systems, is that of a fuel saver. When wind is available, the systems can be used to displace other fuel-fired units, thereby reducing energy costs. The value of the wind generator in this role is obviously related to the costs of the fuel saved and comparative efficiency of the generating units displaced (a subject discussed more fully in Section 9.5). The attraction will be greatest for those utilities facing critical fuel shortages (gas and oil), and high costs of oil in the years ahead.

Environmental Problems - The environment is of increasing concern to the utility industry. Air and water pollution, the unsightliness of transmission lines and towers and the safety and waste disposal fears associated with nuclear power plants are problems of the environment to which utilities are required to respond, both by law and public scrutiny. Wind energy systems, although presenting minor environmental concerns of their own, have as one of their major attractions the prospect of offering a non-polluting source of energy.

As utility companies come under new pressures to reduce the environmental impact of their facilities, they will, at times, be unable to build the type of optimum generating plants they prefer. Because of outright opposition or the expense of environmental safeguards, wind generators may become more attractive options despite possible disadvantages in efficiency and economy.

Load Pattern and Growth - With the assumption that wind generators will be used, at least initially, as fuel saving devices, it is necessary that the using utility be able to employ them efficiently for the maximum amount of time they are generating energy. Because of the impracticality of cycling large, base-load units (such as nuclear plants), wind-generated energy will have minimal value during periods of low demand and will be of greatest value during periods of peak demand when the least efficient and most costly generating units are normally brought on line. Thus, daily and seasonal variations in the load pattern may be factors in the decision to employ wind generators. Actually, both wind patterns and load patterns, and their relative concurrence, will be the important determinants. The ideal situation would have the most windy periods, daily and seasonally, coincide with periods of peak demand.

Anticipated load growth will also be a factor. Utilities with surplus capacity will probably elect to bear heavy fuel costs rather than invest in unneeded capacity. And, as energy research and development expand, a utility with no near-term capacity problems may be cautious about accepting any new equipment in anticipation of the availability of future improved systems. The utility likely to be most receptive to adopting a new energy source, such as the wind generator, is one which cannot delay expansion.

9.4.2 Public and Operating Safety

Safety was discussed with the two consulting utility companies. Guidance on specific areas was also received from OSHA's Hartford office and from the Connecticut Aeronautical Commission. No company or agency contacted expressed any serious concern about public or operating safety for the wind generator system.

9.4.2.1 Public Safety

From the standpoint of public safety, few concerns were raised about the WGS. It was agreed that each site would have to provide proper security against unauthorized entry to the premises and equipment. Security provisions comparable to those found at a typical utility substation were judged to be adequate. These would include a locked perimeter fence around the site and locked cabinets or enclosures for any equipment housed at the base of the tower.

The possibility of damage or injury due to ice shedding or the accidental throwing of a rotor component beyond the confines of the perimeter fence raised the question of the need for a buffer zone around the site. In discussions with Northeast Utilities, it was learned that rights of way for high voltage transmission lines do not generally extend for sufficient distances on either side to prevent a tower from toppling onto neighboring property, although usually there is a restriction on the location and height of structures on property adjoining these routes. The probability of a transmission tower collapsing and causing damage to adjacent property is so remote that buffer zones are considered unnecessary. While the wind generator was generally considered an analogous situation, state and local zoning laws might view the WGS differently.

With respect to site security, only one other concern was expressed: the possibility of vandalism or sabotage to a remotely located and unattended wind generator. A large rotating rotor might present a tempting target for hunters, and occasional vandalism could be a problem. The concern was greatest at Colorado Springs Public Utilities where neighboring utility systems had been experiencing deliberate destruction of transmission towers, transformers, etc. It was concluded that little could be done to protect against occasional vandalism, however, and that the saboteur bent on disrupting public service would probably select a target whose destruction would cause a widespread power failure, rather than the localized disruption that might result from the loss of a single wind generator. Overall, the threat of vandalism and sabotage was regarded as a negligible concern.

Regulations pertaining to air traffic safety were discussed with the engineering division of the Connecticut Aeronautical Commission. FAA regulations contain restrictions on the heights of structures and requirements for obstacle lighting in the immediate vicinity of airports. It is highly unlikely that wind generators would be situated close enough to airports to be affected by these regulations. Structures under 500 feet in height and removed from the vicinity of an airport are generally not subject to regulations, except in rare cases where the structure and the terrain on which it is located combine to create an air traffic hazard. If this were to occur on an established air traffic route, the FAA might have to be consulted for a ruling.

9.4.2.2 Operating Safety

Since the wind generators will operate unattended, operating safety becomes a concern primarily during maintenance of the system. Requirements for personnel safety will be imposed on the design of the system through compliance with OSHA and industry standards. Established utility practices, incorporating applicable OSHA guidelines, prescribe standard safety procedures to be followed by personnel performing maintenance on various types of equipment. Operating safety requriements, as set forth in OSHA and industry standards, are comprehensive and very specific in many areas. Some of the more important of these requirements are discussed briefly below.

Tower Access - Stairs or ladders for personnel access to the WGS operating machinery are acceptable, although an inexpensive personnel hoist would be desirable. If a hoist is used, ladders or stairs would be required as backup. Landing platforms will be required every 20 feet for uncaged ladders and 30 feet for caged ladders. OSHA will probably require that climbing surfaces be enclosed or that heaters be provided to prevent icing.

Fire Protection - An automatic fire extinguishing system would not be needed for the WGS. Portable extinguishers at the top and base of the tower will be adequate. Fireproof hydraulic fluid should be used.

Emergency Egress - A secondary exit from the nacelle at the top of the tower will not be needed. Provisions will be required to remove an incapacitated person from the top of the tower. Utility crews normally carry with them emergency equipment which can be used to sling lower an injured worker from a tower.

Safety Devices - Utility personnel rely primarily on the use of proper procedures for personnel safety. Devices such as equipment interlocks and key switches are used infrequently. It can be assumed that most purchased equipment will be built to OSHA standards.

9.4.3 Maintenance

With the possible exception of the rotor components, which are unfamiliar to the utilities, maintenance of the WGS appears to present no unusual problems. The power train and generating components and their associated controls have counterparts in many of the facilities presently being maintained by the utility companies and are considered to be within their existing maintenance capability. However, because the WGS will operate remotely and unattended for long periods of time, reliability is viewed as a particularly important design parameter, and the capability to detect critical faults in the system and shut down automatically and safely was stressed. The areas of maintenance, in which most interest was expressed by the consulting utilities, are discussed briefly below.

9.4.3.1 Design Considerations

Scheduled maintenance and servicing, e.g., routine calibration, lubrication, oil replenishment, etc., should be minimal and at intervals no more frequent than every 30 days. The design should facilitate in-place repair to avoid having to remove components from the system, especially the large dynamic components on the top of the tower. This will require attention to external and internal access and to repair procedures which can be effected at the site with field-type equipment. The need for special skills and equipment, not normally available to the utility, should be strictly avoided. Major components of the system, such as rotor blades, transmission and generator, should be long-lived and designed for ease of replacement. It should be possible to remove each of the major dynamic components, with the exception of the rotor hub, without removing or displacing other major components of the system. In addition to the failure flags, which are actuated in the event of an unscheduled system shutdown, data which will assist maintenance technicians in localizing and diagnosing the failure should be recorded, if the cost of doing so is not excessive.

9.4.3.2 Maintenance Cycles

The utilities would expect to perform routine inspection and servicing of the WGS at 30 day intervals. A major inspection, possibly involving some component tear-down, would be performed annually. Upkeep of the tower and associated structures would be scheduled at 10 year intervals unless the WGS was situated in a particularly harsh environment. Overhaul and off-site repair of major components would be performed "on-condition", i.e., on the basis of observed wear or deterioration, and with the possible exception of the rotor blades, by the utility itself. The decision to acquire in-house capability for major repair of rotor blades would probably depend on the number of systems being operated and volume of blade repair work anticipated.

9.4.3.3 Lifts and Hoists

An equipment hoist, capable of lifting tools, parts, and supplies to the top of the tower, will be necessary. If it can be provided inexpensively, a hoist capable of lifting all of the major system components (rotor blades, transmission, generator, etc.) is desirable. (Preliminary analysis of component weights and sizes, variations in lifting geometry, and available space on the tower indicates that this will be difficult to achieve, however.) In lieu of a built-in hoist for the major components, the utilities would consider the use of a portable hoist to be brought to the site, temporarily assembled on the tower, and dismantled when the job was finished. Infrequent replacement of major components would make the use of an external crane also acceptable, although bringing heavy equipment into the site may require better access roads than would be needed for lighter vehicles.

The use of ladders or stairs, for personnel access to the top of the tower, is acceptable, although an inexpensive personnel hoist would be desirable. Several suggestions were made for providing access to the rotor blades for inspection and repair, including the use of a boatswain's chair. A chair of this type, currently used in various applications by the utilities, could be suspended at the rotor hub, tethered to the vertically positioned rotor blade, and lowered along its span for inspection and repair. There are a number of alternatives which would have to be studied during detail design, however.

9.4.3.4 Forced Outage Rates

Based on their experience with similar types of equipment, the utilities would expect WGS forced outage rates (unscheduled system down-time) not to exceed 2% to 3%.

9.4.4 Environmental Impact

Among the operational and institutional issues examined in this program, none evoked greater comment or diversity of opinion than did the subject of environmental impact. Very much a factor in utility development plans, views on environmental impact vary greatly with the locale, the relative influence of the parties involved and the weight of other priorities. Threats to the environment, which may accompany the construction and operation of various utility plants, tend to be measured against other problems of energy production, however. Energy shortages and high energy costs have a mitigating influence on environmental concerns.

One of the major attractions of wind generator systems is that they offer a non-polluting source of energy. Growing concern about the quality of the air and water has imposed increasingly restrictive and costly regulations on the utility companies and the operation of their generating plants. Wind generators, besides offering a potentially limitless supply of energy, avoid many of the environmental problems with which the utilities are now contending.

There are, however, some concerns about wind generator systems from an environmental standpoint. Some of these, such as those having to do with land use and the possible hazard to birds and wildlife, would only become problems for a utility in rare situations, and are viewed as minor hinderances. Only two possible problems with the operation of wind generators raise serious concern: noise and aesthetic acceptability.

9.4.4.1 Noise

The question of noise levels was raised by the Connecticut Department of Environmental Protection. It was suggested that the noise potential of the wind generator be examined, especially that of the rotor and tower-mounted machinery, and that the system be designed to levels which would not be bothersome to the surrounding population or passersby. It is believed, however, that noise from the wind generator will be below an objectionable level, and that the systems will generally be located in areas where populations will be unaffected.

9.4.4.2 Aesthetic Acceptability

Aesthetic acceptability is the one environmental issue over which most controversy has arisen. Similar to the problem now being experienced by many utilities in the construction of overhead transmission lines, the question is whether large wind generators can be designed and situated, perhaps in large numbers, so as not to create a highly displeasing visual effect. Opinions are varied and subjective.

Several factors will influence people's reactions to the presence of wind generators. One of the most important is siting. The proximity and visibility of wind generators to the surrounding population will be significant. It is generally felt that public acceptance will decline with increasing numbers in a given locale. (The solution is not as simple as removing wind generators from populated areas entirely, however, because it is these areas, close to the load centers, where they can be most efficiently employed.) It will be desirable, therefore, to seek locations in sparsely populated areas and to subdue the presence of the WGS, where possible, through planned spacing and the use of natural cover, such as hills and tree lines. (Unfortunately, locations most attractive from an aesthetic viewpoint, may often have undesirable wind characteristics.)

Tower design will be another important factor in the aesthetic appeal of wind generators. The shell type of tower is judged to be the more aesthetically pleasing design, although the truss style may find preference in some areas where the modern looking shell would appear out of place with the surrounding environment. There is some feeling that the large rotating rotor will dominate the visual appearance of the WGS, making the tower design of less importance. Unless the rotor is highly reflective or contrasts sharply with the natural background, however, it should be much less visible than the tower.

Background blending may help to subdue the presence of the WGS. At Colorado Springs Public Utilities, where the flat topography of the plains and the backdrop created by the mountains tend to accentuate the presence of any sizable structure, opposition to overhead transmission lines and towers has mounted steadily. CSPU has tried various paint schemes for towers and has experimented with decorative plantings, rustic fencing, and panels to disguise the base of towers, transformer yards, etc. These techniques have been successful in some applications, and similar approaches might be tried with the WGS.

9.4.5 Licensing

Licensing the construction of a new utility plant is a complex and costly procedure. For a large nuclear plant, the process can involve many months (sometimes years) and cost hundreds of thousands of dollars. The time and cost is related to the number of different agencies (governmental and private) involved, and to the number and types of different permits and approvals required. Both will vary with the type of project and the regulatory requirements of the state.

To provide some indication of the possible complexity of licensing a utility project, all of the agencies listed below were mentioned in discussions with the utilities as possible participants in the licensing procedure for the WGS:

Federal Level

Federal Power Commission

Environmental Protection Agency

Federal Aviation Agency

Army Corps of Engineers

Department of the Interior

State Level

Public Utility Commission

State Environmental Agency

Utility Siting Council

Department of Public Works

State Aeronautical Commission

Local Level

Planning Board

Zoning Board

Department of Public Works

Airport Authority

Town Council

Private Groups

Political

Environmental

Fraternal

It is unlikely that all, or even a majority of these agencies, would be involved in any single project but, depending on the state, the location and numerous other factors, each of these agencies could be involved. To further illustrate the possible complexity of the licensing process, some or all of the following types of permits and approvals were mentioned as possible requirements for a license:

Economic analyses and forecasts

Environmental impact statement

Construction permits

Transmission line permits

Eminent domain authority

Permits to cross public lands, waterways, and highways

Zoning variances/exceptions

Building permits

Again, every project would not require all of these permits and approvals, but each could be required at one time or another.

Neither of the consulting utilities was able to outline specifically the procedure that might be required to license a wind generator system or to estimate the costs involved. Both agreed that the WGS would be treated as a generating plant for purposes of licensing, but neither was able to draw a parallel between it and any of the other plants for which licenses had been obtained. The WGS will be unique both from the standpoint of individual unit size and the manner in which units may be combined into systems or networks, possibly over large geographical areas.

Apparently, the licensing procedure would differ between the two states in which the consulting utilities are located. Connecticut has a Utility Siting Council which conducts hearings on new utility projects, coordinates the participation of other state agencies and grants construction approval. While this does not eliminate entirely applications and approvals at the local level, approval by the Siting Council generally supersedes the authority of local agencies. Authority for the licensing of new utility projects in Colorado is more diffused, involving several environmental commissions and a land use commission at the state level, all of which may be required to pass on a project.

The requirements for licensing a wind generator, as mentioned, are not specifically known at this point. From discussions with the two consulting utilities and the Connecticut Department of Environmental Protection, some overall opinions were reached, however.

The principal interaction will be at the state level, involving such agencies as the Utility Siting Council, environmental agencies and the Rate Setting Agency. Federal agencies will rarely enter the licensing process. Local planning and zoning boards will be involved to the extent of issuing building permits and approvals.

Applications and permits will be most costly if they are required for each site. It may be possible to apply for permits on a regional system basis to reduce these costs.

Environmental surveys and impact statements will be the most costly of the licensing requirements and will probably be required for each site. The cost could be minimized, however, by developing "boiler plate" data applicable to WGS sites in general, leaving only site-specific data to be developed in each case.

As part of the licensing process, utilities are often required to carry out a certain amount of public relations activity. Its purpose is to acquaint local groups, possibly in opposition to the project, with its value to the community.

For a nuclear plant, the cost of these activities can be very substantial. Not anticipated to be a major cost in licensing a WGS, public relations costs may be significant in some cases.

9.5 Applications

The application of wind generators to the power generation requirements of utility companies is, ultimately, a question of whether such systems are economically competitive with other alternatives. With this in mind, a preliminary analysis was performed to evaluate the relative cost of wind generator systems in typical application scenarios. A straight-forward approach was followed which computes the direct capital cost, in dollars per kilowatt, of a WGS which will provide a power generation capability comparable to systems now in use. A utility company's typical power generation requirements were represented in the analysis by three specific applications: a fuel saving system only, a fuel saving system with base load capacity and a fuel saving system with stored peak load capacity.

In evaluating the economic aspects of WGS energy production for each of the three applications considered, the basic approach followed was to calculate the break-even cost of the Wind Generator System for replacing an existing fuel-fired system. Simplifying assumptions were made in each case and only the more significant system characteristics and operational factors were included in the analysis. Since a WGS would only be an attractive alternative system for an operating utility if it is economically competitive with fuel-fired systems, the main purpose of the analysis was to predict wind generator system costs which meet this requirement, rather than to conduct a detailed analysis of the operational capability of the system.

9.5.1 WGS as a Fuel Saver

The use of the wind generator as a fuel saving system is, perhaps, the most obvious immediate application in an era of rapidly increasing fuel costs, particularly for oil and gas. A WGS connected directly to existing utility nets can displace other oil or gas-fired generating units during periods of available wind. This could involve any number of individual wind generators, since electrical energy would be provided to displace other systems on an "as available" basis.

For the fuel saving application, it is assumed that the WGS supplies energy directly to the utility company's distribution system as it is produced, without storage and with no base-load capacity credited to the WGS. This energy displaces an equivalent amount generated by a fuel-fired peaking or cycling unit. The analysis accounts for the capacity and energy cost of the WGS, as well as the efficiency of the cycled unit and the displaced fuel cost. Break-even costs of the WGS are computed as a function of the energy cost of displaced fuel.

The results of the analysis for the fuel saving application are presented graphically in Figure 9-2 in terms of the maximum WGS cost (in dollars per kilowatt) as a function of fuel cost (in dollars per million Btu) for various types of alternative fuel-fired systems based on present price levels. The maximum

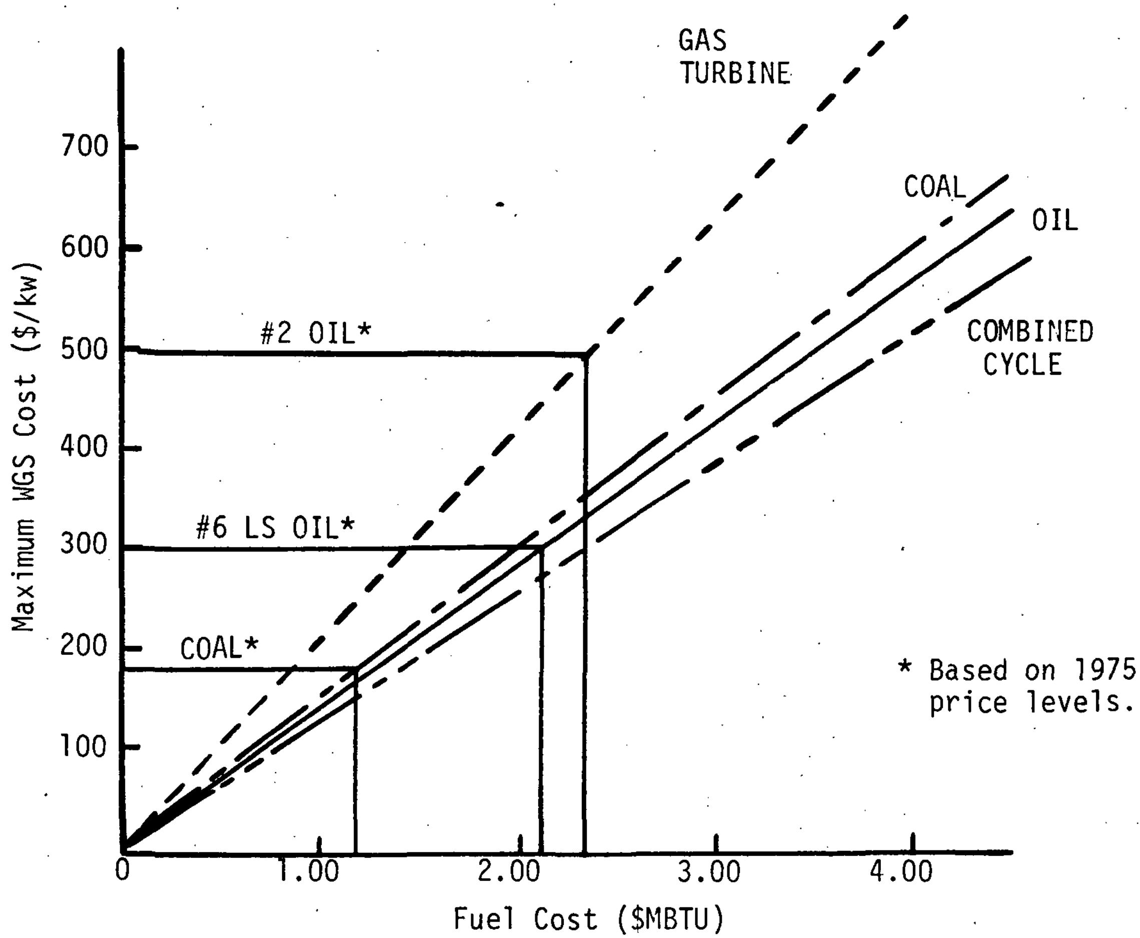


Figure 9-2. Break-Even Costs - WGS vs Fuel Costs.
(No Allowance for Reduced Efficiency)

WGS cost is the computed break-even cost above which the WGS energy cost is greater than the cost of fuel saved. Therefore, the actual WGS cost must be less than the break-even cost if the WGS is to have an economic advantage over the fuel-fired systems. The results show that a WGS is competitive with a gas turbine system using relatively high-cost #2 oil at a break-even cost of approximately $500 per kilowatt. However, compared to systems using lower-cost fuels, such as coal-fired systems and systems burning #6 oil, the WGS break-even cost is considerably lower: $300 per kilowatt compared to #6 oil and $180 per kilowatt compared to the cost of coal.

9.5.2 WGS With Base-Load Capacity

The WGS can be credited with base load capacity when some or all of its energy output is available to meet the daily base load or energy demand. (Capacity credit is not to be confused with the term capacity factor which expresses the ratio of energy actually produced over some period to the maximum energy producible in that period). To obtain capacity credit, the WGS energy contributing to base load must be available on demand with a high degree of assurance. Its value is equal to the total energy cost of the base load generating plant it replaces.

The application of the WGS as a fuel saving system with base-load capacity credit assumes that a large number of dispersed wind generators are used to obtain benefit from the effect of wind variability over the system. This allows some base-load capacity to be credited to the WGS. Energy produced in excess of the rated base-load capacity can be used, when available, in a fuel saving application. The capacity credit of the WGS and the displaced base-load energy cost are considered in the analysis in addition to the factors considered in the fuel saving application. Break-even cost for the WGS is computed based on the cost of the base-load energy displaced and the cost of fuel saved.

The analysis of WGS break-even cost for a fuel saving application with base-load capacity was performed for a 1000 MW installed capacity wind generator system with a 100 MW base-load capacity at 0.7 capacity factor. This means that 100 MW will be available to meet the base load demand with 70% reliability. For a total system capacity factor of 0.35, the WGS would have an effective capacity factor as a fuel saver of 0.31. The analysis used a displaced base-load energy cost of $0.035 per kilowatt hour and a displaced fuel cost of $0.0235 per kilowatt hour ($2.35 per MBtu) at a heat rate of 10,000 Btu/KWh. The results of the analysis predicted a WGS break-even cost of approximately $510 per kilowatt for fuel saving with base load capacity, a value only slightly higher than the value of fuel savings alone.

9.5.3 WGS With Storage

The final application considered combines the WGS with some form of energy storage. This would enable the utility to dispatch the energy in a controlled manner, even though it is generated with considerable variability, and gives capacity credit to the WGS for peak load periods. Peak load capacity credit is equivalent to base load capacity credit, except that the WGS energy is used to displace more costly peaking units such as gas turbines.

In this application, it is assumed that the stored energy produced by the WGS is used to displace high-cost peaking units, while surplus energy not needed for storage is used for fuel saving as before. Sufficient energy production to charge storage on at least 90 percent of the available days is also assumed. In addition to the factors bearing on fuel saving applications, the analysis of peak load applications also considers the WGS yearly and daily capacity factors, storage efficiency and the displaced peak unit energy cost. The WGS break-even cost in this case includes the cost of the storage system and depends on the cost of peak energy displaced, as well as the cost of fuel saved.

Figure 9-3 presents the results of the analysis for the WGS with storage for peak load applications. The maximum (break-even) WGS cost is shown as a function of the WGS daily capacity factor for two values of storage efficiency. Note that this analysis compares the WGS cost to the cost of a gas turbine peaking unit. The results indicate that both the daily capacity factor and the storage efficiency have a significant effect on the WGS break-even cost. Assuming nominal values of 0.10 for the daily capacity factor and 0.75 for storage efficiency, the resultant WGS break-even cost is approximately $480 per kilowatt, including storage.

9.5.4 Interpretation of Results and Other Factors

The results of the economic analysis of possible utility company applications of wind generating systems are more meaningful when interpreted in light of the system cost analysis performed for the 500 KW and 1500 KW WGS designs during the study program. This detailed cost analysis predicts direct capital costs of approximately $900 per kilowatt for the 500 KW wind generator and $480 per kilowatt for the 1500 KW system. Obviously, the larger WGS is more attractive economically.

The $480 per kilowatt capital cost of the 1500 KW WGS compares favorably with the estimated $500 per kilowatt break-even cost required in fuel saving applications for displacing gas turbine generating systems. However, compared to systems using coal and #6 oil as fuels, the 1500 KW WGS is not competitive at present. Whether this system will be more competitive with lower-cost fuels in the future depends on how high fuel costs will rise and how much WGS costs might be reduced through continued development.

Fuel saving is judged to be the most promising application of wind generators at the present time. Systems for this application could range in size from a single wind generator to large multi-unit installations and, therefore, they might find widespread use by both large and small utility companies.

For the fuel saving plus base-load capacity application, the predicted capital cost of the 1500 KW WGS is slightly under the estimated break-even cost of $510 per kilowatt. The assumptions considered in the analysis are very optimistic, however. Even if the necessary wind conditions could be found, a very large "system" of WGSs would be required and transmission costs associated with such a system, which were not accounted for in the predicted WGS capital cost, would increase this cost further. Despite the large number of wind generators required to satisfy the application, only about 10% of the total installed power, based on the example chosen, would be added to the user's base-load

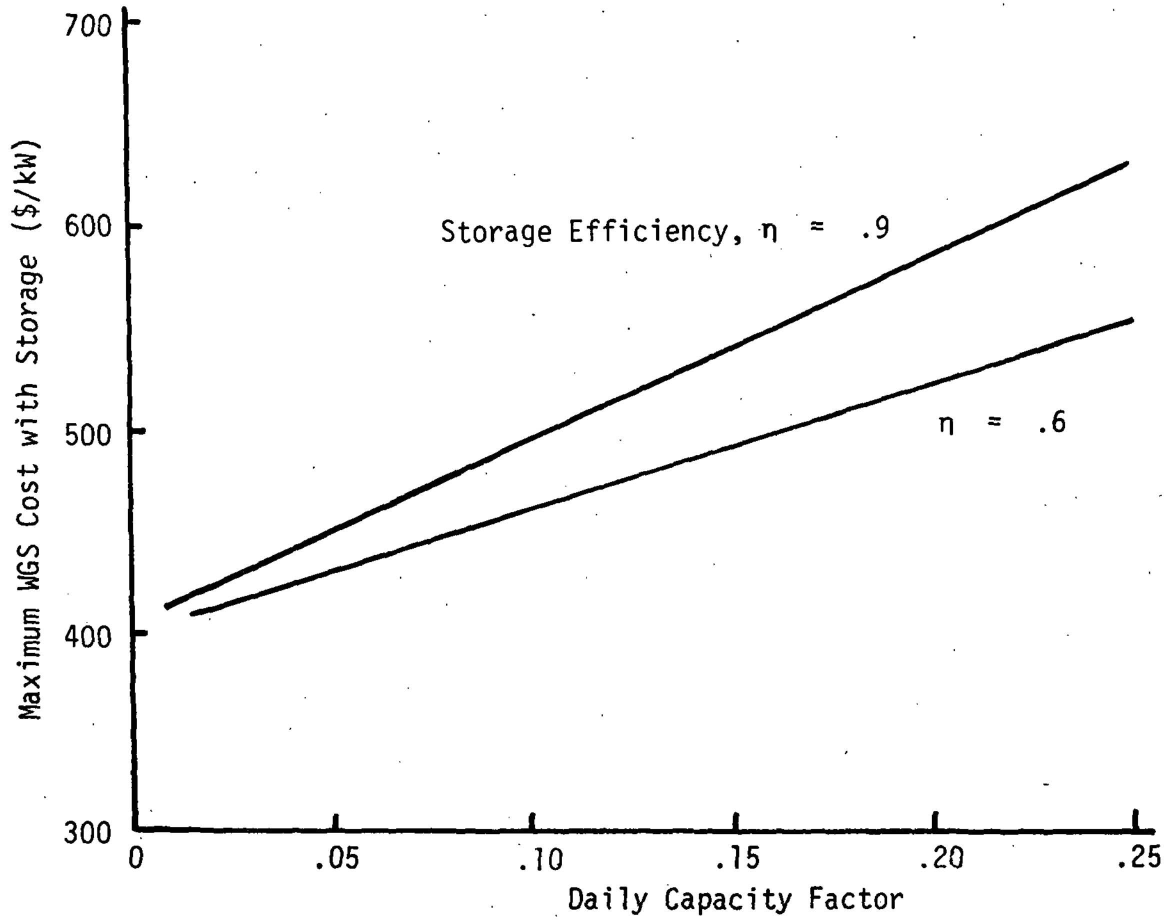


Figure 9-3. Value of WGS with Storage.

<u>Factors:</u>

Daily Peak Load = 500 MWH
Alternate Peak Energy Cost = $.0485/kWh
Cost of Off-Peak Fuel Saved = $.025/kWh

capacity. In effect, such a system is still primarily a variable output fuel saving system. Because of the large number of units required and the rather small base-load capacity afforded by the system, it appears that this application offers limited promise of being economically feasible for a typical utility in the foreseeable future.

The 1500 KW WGS predicted capital cost is roughly equal to the break-even cost computed for the fuel saving plus peak load application. It should be noted, however, that the 1500 KW WGS capital cost is optimistic for this application since storage costs and allowances for system downtime are not included. The analysis of break-even cost assumed also that the stored wind energy would displace high-cost gas turbine generated energy. If a less costly generating source is displaced, the WGS break-even cost would be correspondingly lower. The viability of the WGS in this application appears to be contingent upon the development of inexpensive and efficient storage systems.

The applications analysis has attempted to evaluate, in gross relative terms, the economic worth of a wind energy system to a typical electric utility. Because it has been conducted in the context of a utility operation, it did not explore applications requiring departures from present day industry practice and regulation. Some of the regulations affecting utility operations, such as those governing rate structures and reserve capacity, do not easily accommodate variable or intermittent generating capacity, even though this type of power might be entirely feasible in specific applications. Using the wind generator as a variable energy source for selected segments of the market that could operate with anticipated power interruptions, and structuring the rates accordingly, might open up new areas of application.

Wind energy costs in the applications analysis have been based on capital, operations and maintenance, and fuel costs currently anticipated by a relatively large northeastern utility and on present-day projections of future fuel availability. These factors vary considerably among other utilities in different parts of the country and might change drastically in the future due to unforeseen circumstances. Should the supply of oil and natural gas approach depletion more rapidly than predicted and the costs escalate accordingly, wind energy would become increasingly more competitive in the United States. In other areas of the world, primarily the non-industrialized nations and even in some remote areas of the United States, conditions today may be such that wind energy is technically and economically a competitive alternative.

9.6 Conclusions and Recommendations

This program has explored, briefly, some of the operational and institutional issues that will be involved in the development and application of wind generator systems. It has also examined, in simplified terms, the economic value of wind generators in some potential applications with the utilities. The purpose of these studies was to acquire a basic understanding of operations and economics from the viewpoint of the utility industry and to apply this knowledge in the analysis and preliminary design of the WGS. This objective was met.

The study of operations and economics in this program, although limited, has allowed general conclusions to be drawn in some areas. These conclusions and some recommendations for future work conclude this section of the report.

9.6.1 Conclusions

1. Preliminary estimates of direct capital costs and energy costs indicate that the WGS could be competitive with other energy sources under present assumptions of fuel costs and availability over the next decade.

2. Operational and institutional issues, although requiring greater study, appear to present no serious obstacles to WGS development.

3. Maintenance and safety requirements appear to be within the existing capability of the utilities.

4. Environmental impact will focus mainly on aesthetic considerations, primarily the visual element. Public reaction to the initial installations is expected to be favorable, but opposition to the WGS will probably develop with increasing numbers in a given locale. Some measures might be taken to improve the aesthetic appeal of the WGS.

5. Licensing could become a time-consuming and costly requirement of WGS acquisition for a utility. Environmental surveys and statements are expected to be the most costly element of the licensing process. These costs might be reduced through licensing WGSs on a regional system basis and developing "boiler plate" environmental data.

9.6.2 Recommendations

1. When, through the results of ERDA's mission studies and similar work, it is possible to better define application scenarios for the WGS (numbers, spacing, proximity to load centers, etc.) the costs of access roads, transmission lines and other necessary site interfaces should be evaluated.

2. Maintenance and safety, although not expected to present unusual problems for the utilities, should be carefully considered factors in future design efforts.

3. One of the factors in the choice of sites for the upcoming experimental test programs should be the amount of public exposure the site will provide. The test programs should include provisions for sampling public opinion on the WGS and, to the extent that funding might allow, experiments should be conducted to test reactions to various measures aimed at improving the aesthetic appearance of the WGS.

4. The subject of WGS licensing requirements and costs should be examined more thoroughly in future programs.

5. Economic analyses of potential WGS applications should be conducted in greater detail using data on actual load and wind conditions as a basis. (Northeast Utilities is presently constructing a computer model to test various application scenarios using sampled wind data on a quasi "real time" basis. This type of analysis might be adopted for future wind energy programs.)

10.0 CONCLUSIONS AND RECOMMENDATIONS

The major conclusions and recommendations derived from the results of this study are summarized below. Detailed results, conclusions and recommendations for the WGS and its subsystems and components are presented in the appropriate sections of this report.

1. The WGS concept selected for preliminary design consists of a two-bladed, variable pitch rotor, driving an A/C synchronous alternator through a standard commercial gearbox, mounted on either a steel truss or concrete shell tower. This system concept offers the lowest capital and energy costs, highest operating efficiency and reliability, and lowest maintenance and technical development risk of all the concepts examined in the study. Therefore, this system configuration is recommended for development, test and demonstration in electric utility applications.

2. Two machines are recommended for development: a 500 KW rated WGS and a 1500 KW rated WGS. These machines fit the contract requirement for a low power machine, installed at a site with a 5.4 m/s (12 mph) median wind speed, and a high power machine for a site with an 8 m/s (18 mph) median wind speed. The low power machine will yield energy at a cost comparable to systems optimized for sites with median wind speeds between 3.5 m/s (8 mph) and 6.3 m/s (14 mph). The high power system will yield energy at a cost comparable to machines optimized for sites with median wind speeds between 6.3 m/s (14 mph) and 9 m/s (20 mph). Therefore, these two designs can be economically used at most feasible power plant sites in the United States.

3. For WGS rotors of the size required, 40 to 60 m (130 - 200 ft) in diameter, technical requirements indicate that composite material construction should be employed for the rotor blades. Composite fabrication technology offers the most feasible method of tailoring the blade to meet the demanding structural and dynamic properties required in the WGS application. It is recommended that automated filament wound fabrication processes be used to construct the composite blades to meet the demanding cost requirements for the WGS rotor.

4. A rigid, non-articulated hub design should be employed to minimize the hub cost and potential dynamic problems in the WGS rotor. Although the loads transmitted through the conversion machinery to the tower will be higher than with rotors employing other hub design approaches, these loads can be handled by the conversion machinery and tower with little additional cost. Therefore, the additional complexity, cost and potential technical problems of other hub concepts do not appear warranted.

5. Rotor torque control through blade pitch variation minimizes blade operating loads and simplifies rotor operation in potentially unstable aerodynamic flow regimes. Therefore, this type of rotor torque control is recommended. Since the blade pitch control rate is determined by gust requirements, it is further recommended that a design gust spectrum be defined prior to detail design of the control system.

6. Standard, commercially available brake concepts for preventing rotor overspeed under extreme gust or failed control system conditions are not available. Therefore, it is recommended that an independent mechanical emergency blade feathering device be incorporated into the rotor design to prevent excessive rotor overspeed and subsequent fatigue damage and/or potential structural failure.

7. Conventional electromechanical controls are recommended for the blade pitch and orientation primary controls. Use of conventional electromechanical controls is the most acceptable approach to operating electric utilities, and permits the economical integration of the emergency mechanical blade feathering device into the rotor pitch controls. Utility proven microprocessor-based sequencing and supervisory controls are recommended for the startup, shutdown, operational monitoring, failure detection and reporting, data transmission and recording, and other similar functions.

8. A standard fixed ratio gearbox is the most economically and technically feasible speed increaser available for the WGS application. Since commercially available units can meet all of the system requirements at low cost, standard, commercial gearboxes are recommended.

9. Standard, utility-compatible, off-the-shelf electrical equipment, including generators, transformers, switchgear, etc., can meet all the technical requirements of the WGS. Therefore, all standard electrical equipment is recommended for the WGS. Either synchronous or induction generators are suitable for, and compatible with, the WGS with either type offering minor advantages and disadvantages in cost and technical characteristics. Therefore, the choice of generator should be offered as a user option and the control system, protective devices and electrical interface equipment should be compatible with either type of generator.

10. Stability analyses of the WGS on a typical utility distribution line indicate that, under most operational fault and wind gust conditions, the system remains stabilized and synchronized with the utility network. However, it is recommended that reference utility distribution line physical and operational characteristics be defined to permit selection of breaker and relay ratings, and generator and regulator characteristics for the detail WGS designs. These network characteristics should be defined so that a minimum of adjustment will be required for each specific installation.

11. Either steel truss or pre-cast concrete shell tower designs offer low cost towers for the WGS in large production quantities. The steel truss tower is significantly lower in cost than any other type of tower in small quantities. *It is recommended that the concrete shell tower be used in production WGS units because of its better appearance. However, the truss tower should be used for test and demonstration programs due to its lower cost and ease of modification.

   <u>*NOTE</u>: The effects of tower shadow on blade loads and blade life, which have come to light subsequent to the original writing of this report, have made this statement

incorrect. It seems certain that towers will be constructed to minimize the disturbance of the airflow for a downwind rotor, eliminating the solid concrete tower.

12. Preliminary analyses of standard electric utility requirements indicate that a WGS can be competitive with other utility energy sources. However, it is recommended that more detailed analyses be conducted during the detail design of the WGS units to determine specific applications and interface requirements.

13. Operational and institutional issues, including licensing, environmental impact, public acceptance and safety appear to present no serious barriers to the widespread introduction of WGS units in standard electric utility networks. However, these issues, particularly the visual acceptability of large numbers of units, should be explored to help guide the selection of the tower design and installation provisions.

APPENDIX A

WGS STUDY REQUIREMENTS

Contract NAS3-19404

WGS STUDY REQUIREMENTS

I. DESCRIPTION AND SPECIFICATIONS

A. General Design Requirements

The wind-driven, electric-power generating system usually consists of the following major components;

1. A propeller-type rotor to interact with the wind

2. A tower to support and locate the rotor in a more favorable wind regime away from the ground effects

3. A generator or alternator to convert the mechanical energy into electrical energy.

4. A transmission to transmit the torque developed by the rotor to the generator at a proper rotational speed. A coupling or clutch which permits slip between the rotor and generating equipment and limits the maximum torque that is transmitted. A brake which will lock the rotor on shutdown.

5. A control system which will orient the rotor into the wind, regulate the power output of the system and ensure safe operation for most wind conditions.

Wind generator systems may or may not include an energy storage system. It is not clear that a practical and attractive energy storage system can be designed and built using existing technology. Therefore, it is currently planned that the 100 kW Mod-1 and MW Mod-1 wind generators will have no provision for energy storage and it is not required that energy storage be studied in this SOW.

Existing wind turbine technology indicates that a propeller-type rotor driving a synchronous or induction generator through a step-up gear type transmission with suitable controls may be the most efficient system available. However, NASA is interested in designing wind generator systems that will produce electrical power at the lowest cost per kilowatt-hour. Wind generator systems studies on this contract shall be restricted to propeller-type rotors, with the axis of rotation near horizontal and supported on a tower. All other components of the system are optional and shall be studied in the conceptual design and parametric analysis phases. The system representing minimum cost of generating electrical power shall be recommended for preliminary design.

The wind-driven, electric-power generating systems shall also be designed to meet the following requirements (design requirements are goals which are to be substantiated by acceptable analyses rather than actual demonstration):

1. Power Rating - Conceptual designs shall be prepared for two power levels, 100 kW and 1 MW. Preliminary designs shall be prepared for a low power (50 - 250 kW) and a high power (500 - 3000 kW) system which represents minimum cost per kW-hour for the wind regimes provided by NASA.

2. 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.

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

4. 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.

5. Generation Costs - The goal of the 100 kW and MW designs shall be minimum cost per kW-hr. These costs shall include capital, amortization of capital over lifetime of equipment, operation and maintenance costs as a minimum.

6. 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.

7. Wind Speeds - The 100 kW and 1 MW conceptual designs shall be based on mean wind speeds of 12 and 18 mph, respectively, and for rated wind speeds which minimize the cost of electric power produces (mils/kW-hr). The resulting concepts must represent minimum cost over the full range of wind speeds to be studied parametrically. If not applicable over the full range of wind speeds, the Contractor shall

determine what changes in the basic design are appropriate and shall direct attention to such changes. Such changes might include multiple rotors per tower, alteration or elimination, or rotor speed control (feathering, flaps), etc.

The low power and high power preliminary designs shall be based on a rated wind speed which will provide minimum cost per kW-hr for the mean wind speed and wind speed duration provided by NASA.

8. 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.

9. 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.

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

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

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

13. 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.

14. Applicability - The designs shall meet the interface requirements of public utilities in the various areas of the United States having favorable winds. It would be desirable to meet all requirements with one design or a design adaptable with minimum change.

B. Specific Design Requirements

The major components of the wind-driven, electric-power generator shall meet the following design requirements:

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 are optional and shall be decided by this 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. This may require utilization of a method such as blade feathering or folding to reduce wind loads.

b. Gusts with 200 mph/sec rates of change.

c. Blade unloading caused by tower effect.

The design shall provide for locking of the rotor when the WGS is in the shutdown condition.

2. Tower - The tower shall be designed for a service life of 50 years, the same wind loading conditions specified for the rotor in IV-B-1, the forces imposed by the rotor, the weight of all equipment located atop the tower and all varying loads which may lead to fatigue.

   The type of tower (lattice, stressed skin, shell, cantilever or guyed) and tower height are optional and shall be determined by this study.

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 kW-hr.

4. Power Transmission - The method of transmitting the rotor torque, coupling to the electric generator at the proper angular velocity and locking of the rotor is optional. The Contractor shall study various methods and recommend an approach consistent with the general design requirements for the system.

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 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 recommended by the Contractor. 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 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 fonverts all available mechanical energy into electrical and then tailors electrical output 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.

6. Application Requirements - 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. Connection to such systems will require suitable switchgear, transformers and transmission lines. Design of the MW Mod-1 (high power) system shall accommodate the application requirements.

C. Specifications

Applicable Government and industry specifications are to be determined by the Contractor.

APPENDIX B

ZODIAC II WGS COMPUTER PROGRAM

ZODIAC II WGS COMPUTER PROGRAM

The ZODIAC II WGS computer program is presented in this appendix. The basic description of the WGS model and computer program is given in Section 3.0, System Analyses.

Nomenclature for the program is given in Table B-1. The nomenclature used conforms to the ZODIAC requirement that variable names contain not more than four characters. A code was developed to identify the variables, in which the first letter of four letter words identifies the bype of variable. Three letter words are used where the variable is recognized in that form.

A complete printout of the most frequently used version of the model is shown in Figure B-1. This version uses rotor diameter as an input variable and was found to be more useful for parametric studies than the other version, which uses rated wind speed as an input variable. However, there are very few differences between the two versions, and all of the pertinent relationships are shown.

Figure B-2 shows the two modules (TOP WEIGHT and COST) of the program of Figure B-1 which were changed to reflect the rotor configuration resulting from the preliminary design phase. This configuration followed considerable refinement in the rotor's design, with better definition of weight and cost.

TABLE B-1. NOMENCLATURE

FIRST LETTER CODE FOR FOUR LETTER WORDS

A = AREA (FT^2) or ANGLE (DEG)
C = COST ($)
D = DIMENSION (FT) (except height dimension)
E = EFFICIENCY, FRACTION (never %, e.g., .95 not 95%)
F = FORCE OR THRUST (KIPS, i.e., 1000 lb units)
H = HEIGHT (FT)
K = CONSTANT
N = NUMBER OR SPEED (RPM)
P = POWER
Q = TORQUE (KIPS - FT)
R = RATIO
V = VELOCITY (MPH or FPS)
W = WEIGHT (KIPS)

NOMENCLATURE

ABLD	Projected flatwise area of rotor blades
ACBL	Cable cross section, in units of 350 mcm
ACLR	Cleared portion of site
AENC	Maximum projected area of enclosure (on top of tower)
AKW	Average power output kW/hr
ALND	Total site area
AROT	Rotor disc area
ATOW	Projected area of tower normal to wind
AT75	Pitch setting at 0.75 rotor radius
CBLD	One blade
CBLS	Total blade assembly
CBPT	Bedplate (part of pintle)
CBRK	Brake
CBRL	Barrel assembly, including pitch and hub bearings (part of hub)
CCAB	Cable
CCAR	Carrying charges

TABLE B-1. NOMENCLATURE (continued)

CCLH	Clutch
CCLR	Land clearing
CCP1	Low speed coupling
CCP2	High speed coupling
CCTR	Controls for pitch control and pintle yaw control, system shutdown
CDIR	Direct capital (total system)
CDRS	Drive system (sum of components)
CEL3	Electrical equipment related to power generation
CEL4	Electrical utilities, including tower lighting
CELC	Electrical peripheral equipment
CENC	Enclosure (nacelle)
CFNC	Fence
CFND	Foundation
CELP	One blade flap
CGEN	Generator
CGRP	Blade grip (part of hub)
CHSG	Hub housing, part of hub
CHUB	Hub and controls
CINC	Controls installation
CIND	Drive installation
CINE	Electrical installation
CINP	Pintle installation
CINR	Rotor installation
CINS	Total installation, excluding tower
CKW	Direct capital cost per kW of rated power, \$/kW
CKWH	Cost of energy, \$/kW-hr
CLAD	Ladders, gratings, platforms on tower
CLND	Land acquisition
CMAP	Maintenance of power system
CMAR	Maintenance of rotor
CMAS	Maintenance of site
CMAT	Maintenance of tower
CMNT	Total maintenance

TABLE B-1. NOMENCLATURE (continued)

COAM	Operations and maintenance
COPS	Operations
COWN	Cost of ownership (interest on construction loans, etc.)
CPAD	Transformer pad
CPCH	Rotor pitch mechanism, bearing part of hub
CPTL	Pintle or turntable
CRCL	Rotor mechanical control linkages, part of hub
CRGR	Ring gear for pintle
CROT	Total rotor
CSFT	Rotor shaft
CSHD	Shed or base hourse
CSLP	Slip device
CSIT	Total site
CSTL	Structural steel components of tower
CCTOW	Tower, including foundation and installation
CTRF	Transformer
CTRN	Transmission
CYAW	Pintel (turntable) drive, yaw control
CYR	Yearly total cost
DBLD	Blade length from grip to tip (assumed airfoil starts at grip)
DCRD	Blade chord at root
DFNC	Fence length
DHUB	Horizontal distance from tower centerline to rotor plane, (ft)
DIA	Rotor diameter
DRAD	Rotor radius
DSID	Tower base width (ft)
EAER	Aerodynamic efficiency of rotor (rotor power/wind power)
EART	Aerodynamic efficiency at rated wind
ECBL	Cable efficiency
EGEN	Generator efficiency
EPLT	Plant efficiency, average power output to rated power fraction
ETRF	Transformer efficiency
ETRN	Transmission efficiency

FENC, FEN#	Wind force on enclosure
FHRZ, FHZ#	Total horizontal force acting on tower (effective)
FROT, FRO#	Horizontal thrust of rotor or drag force on rotor if stopped
FTOP, FTO#	Wind force on structure on top of tower
FTOW	Wind force on tower projected area
HCLR	Rotor clearance from ground
HTOW	Tower height, ground to rotor shaft
IP	Moment of inertia of system on top of tower
K1	Allowable stress factor K1 = 1.0 for operating loads K1 = 1.333 for earthquakes coupled with high wind load
K2	Flexibility correction factor K2 = 2.0 per empirical data
K3	Earthquake load factor K3 = .25 for USA K3 = 0 if earthquake is not to be considered
K4	Constant used for tower foundation cost calculations
K5	Foundation cost factor applied to cost of concrete to allow for soil testing, site excavation and backfilling
KCTS	C_T/σ, thrust coefficient of rotor
KTIP	Design to rated wind velocity ratio (for rotor parameter optimization)
MY, MY#	Turnover moment on tower due to wind shear effect on rotor
MZ, MZ#	Yawing moment on tower
NBLD	Number of blades
NFQB	Natural frequency of tower in lateral bending (cpm)
NFQT	Natural frequency of tower in torsion (cpm)
NGEN	Generator speed (RPM)
NROT	Rotor speed (RPM)
PGEN	Generator power output
PLON	Generator loss at rated power
PLO9	Power to drive mechanical accessories (oil pump, hydraulic pump)
POUT	System output power at any condition

TABLE B-1. NOMENCLATURE (continued)

PRAR	Rotor power at rated condition
PRAT	Rated power of system
PRAN	Rated power
PROT	Rotor power at any condition
PTRF	Transformer (and system) power output
PTRN	Transmission power output
QGEN	Torque at generator input shaft (at high speed)
QROT	Torque at rotor shaft (at lower speed)
RADV	Velocity ratio, VTIP/VEFF
RAMX	Velocity ratio at rated condition
RASP	Aspect ratio of blade (from grip to tip)
RCLR	Rotor clearance ratio, HCLR/DRAD
RCUT	Rotor root cut-out ratio, R_{cutout}/radius
RDEN	Atmospheric density ratio
REWS	Ratio of effective wind to reference wind, VEFF/VWND
RGLF	Generator load factor, fraction of rated output
RTAS	Tower aspect ratio, HTOW/DSID
RTIM	Fraction of year during which power is generated
RTRN	Gear ratio of transmission
RVV	Ratio of rated to median wind speed, VRAT/VBAR
SDYN	Dynamic pressure, lb/ft^2
SOL	Solidity, effective
STS	Allowable stress
TIME	Time (hours)
TTOT	Total WGS operating time per year (POUT > 0)
VBAR	Median wind velocity (at 30 feet, mph)
VEFF	Effective, i.e., integrated wind velocity of the rotor disc (fps or mph)
VMIN	Minimum wind velocity at which rotor rotation is sus- i.e., POUT = 0.
VRAT	Rated wind speed (minimum wind speed at which rated power is generated, mph)
VTIP	Rotor tip speed, ΩR (fps)
VWND	Free stream velocity (reference) at 30 ft height (mph)

TABLE B-1. NOMENCLATURE (continued)

WBAS	Tower base support structure
WBLD	Blade weight (one blade)
WBLS	Total blade assembly
WBPT	Bedplate (part of pintle)
WBRC, W#B, WB	Tower horizontal bracing
WBRL	Barrel assembly, including pitch and hub bearings (part of hub)
WBRK	Brake
WCAB	Main cable - portion attached to the tower
WCHD, W#C, WC	Tower chords (legs)
WCLH	Clutch
WCP1	Low speed coupling
WCP2	High speed coupling
WCTR	Controls for pitch control and pintle yaw control, system shutdown
WDIA, W#D, WD	Tower diagonal bracing
WDRS	Drive system (sum of components)
WEL1	Electrical equipment on top of tower
WEL2	Electrical equipment distributed along the tower
WELE	Electrical systems on top of tower (total)
WENC	Enclosure (nacelle)
WFLP	One blade flap
WGEN	Generator
WGRP	Blade grip (part of hub)
WGUS	Tower gussets and connections
WHSG	Hub housing, part of hub
WHUB	Hub and controls
WLAD	Ladders, gratings, platforms on tower
WMSC	Weight of miscellaneous tower items
WPCH	Rotor pitch mechanism, bearing part of hub
WPTL	Pintle or turntable
WPTS	Pintle (turntable) support
WRCL	Rotor mechanical control linkages, part of hub
WRGR	Ring gear for pintle

TABLE B-1. NOMENCLATURE (continued)

WROT	Total rotor
WRTG	Total of rotating assembly on top of tower
WSFT	Rotor shaft
WSLP	Slip device
WSTL	Structural steel components of tower
WTOP	Total weight of items on top of the tower for tower weight calculations
WTOW	Tower distributed weight, for tower weight calculations
WTRN	Transmission
WTTW	Total power
WWGS	Total wind generating system above foundation
WYAW	Pintle (turntable) drive, yaw control
YKWH	Yearly energy output, kW-hrs

NOTES: # denotes one of five design criteria for which the given item was sized:

1. Blowover
2. Normal operation + earthquake
3. Gust
4. Stiffness for natural frequency in bending
5. Stiffness for natural frequency in torsion

n refers to rated power or power loss at rated condition of one of the four power system components:

1. Transformer
2. Cable
3. Generator
4. Transmission

```
CONTROL MODULE
COMMON RCUT,RDEN,NGEN,NBLD,VWND,END,DIA,VTIP,PT,KWHR,PRAT,HCLR,TO,VBAR
COMMON TIME,TTOT,RTIM,AKW,EPLT,YKWH,KWHK,DHUB,ACBL,RTAS,DCRD,RASP,CBLD
COMMON NROT,HTOW,DSID,QGEN,QROT,STS,K2,NFQB,NFQT,IP,ATOW,RAMX,REWS,ABLD,AENC
COMMON ATOW,DRAD,KTIP,RVV,EGEN,ETRF,PRA4,PRA2,PLC9,PLO3,PLO2,PLO1
COMMON WBLD,WFLP,WBLS,WPCH,WRCL,WHSG,WHUB,WROT,WSFT,WTRN,WCP1,WCP2,WBRK
COMMON WCLH,WSLP,WDRS,WGEN,WEL1,WELE, WCTR,WENC,WBPT,WYAW,WPTL
COMMON WCAB,WEL2,WRGR,WPTS,WCHD,WDIA,WBRC,WGUS,WMSC,WBAS,WSTL,WTTW,WWGS
COMMON PLOS,VRAT,PRAR,PRA3,PTRF,PTRN,PROT,SOL,PGEN,PCBL,ECBL,ETRN,POUT
COMMON W1C,W1D,W1B,W2C,W2D,W2B,W3C,W3D,W3B,WC,WD,WB,W9,W4C,W5D,W5B
COMMON WTOP,WTOW,WRTG,WTTW,WLAD,W1
COMMON CBLD,CFLP,CBLS,CPCH,CRCL,CHSG,CHUB,CROT,CSFT,CTRN,CCP1,CCP2,CBRK
COMMON CCLH,CSLP,CDRS,CGEN,CCAB,CTRF,CEL3,CEL4,CELC,CENC,CBPT,CYAW,CPTL
COMMONCCTR,CRGR,CSTL,CFND,CTOW,CINS,CLND,CCLR,CFNC,CSHD,CPAD,CSIT,CDIR
COMMON CMAR,CMAP,CMAT,CMAS,CMNT,COPS,COAM,CCAR,CYR,CKW,CKWH
COMMON AT75,KCTS,FRO2,FEN2,FHZ2,MY2,MZ2,FRO1,FEN1,FTO1,FHZ1,MZ1,FHZ3,FTO2
COMMON CINR,CINP,CIND,CINE,CINC,CLAD
COMMON RTRN,D1,MY,EAER
END=0
POINT BGIN
RUN MOD CONSTANTS
POINT STRT
RUN MOD EFFECTIVE VELOCITY
VWND=VRAT
RUN MOD ROTOR POWER
EART=EAER
RUN MOD RATED WIND
ITERATE ON RVV,PTOL=0.1 ,TIMES=20,FROM STRT
TO=0
TTOT=0
KWHR=0
VWND=0
POINT WIND
VWND=VWND+1
IF VWND IS GT 45,GO TO FIN
RUN MOD ROTOR POWER
IF PROT IS GT PRAR,GO TO X1
RUN MOD TRANSMISSION POWER
IF PTRN IS LT 0,GO TO WIND
RUN MOD GEN POWER
RUN MOD CABLE POWER
RUN MOD TRANSFORMER POWER
IF PTRF IS LT 0,GO TO WIND
IF TO IS GT 0,GO TO SKIP
RUN MOD INIT TO
POINT SKIP
POUT=PTRF
GO TO X2
POINT X1
PGEN=PRA3
PTRF=PRAT
POUT=PRAT
POINT X2
RUN MOD INTEGRATE
TTOT=TTOT+TIME
KWHR=KWHR+PT
PRINT VWND,PROT,PGEN,PTRF,KWHR
IF TIME IS EQ 0, GO TO FIN
GO TO WIND
POINT FIN
```

Figure B-1. ZODIAC II WGS Computer Program.

```
RUN MOD TOP WEIGHT
RUN MOD TOWER WEIGHT
RUN MOD COST
RUN MOD PLANT EFFICIENCY
PRINT  PRAT, WBLD, CBLD, CFNC, W1C , VBAR, WFLP, CFLP, CSHD, W2C
PRINT  DIA , WBLS, CBLS, CPAD, W3C , DRAD, WPCH, CPCH, CSIT, WC
PRINT  SOL , WRCL, CRCL, CDIR, W4C , NBLD, WHSG, CHSG,     , WCHD
PRINT  DCRD, WHUB, CHUB, CMAR, W1D ,RCUT, WROT, CROT, CMAP, W2D
PRINT  DBLD, WSFT, CSFT, CMAT, W3D , RASP, WTRN, CTRN, CMAS, WD
PRINT  VTIP, WCP1, CCP1, CMNT, W5D , NROT, WCP2, CCP2, COPS, WDIA
PRINT  QROT, WBRK, CBRK, COAM, W1B , NGEN, WCLH, CCLH, CCAR, W2B
PRINT  QGEN, WSLP, CSLP, CYR , W3B , RTRN, WDRS, CDRS, CKWH, WB
PRINT  DHUB, WGEN, CGEN, CKW , W5B , VRAT, WEL1, CCAB, EPLT, WBRC
PRINT  EART, WELE, CTRF,     , STS , RAMX, WCTR, CEL3,     , IP
PRINT  RDEN, WENC, CEL4,     , D1  , REWS, WBPT, CELC,     , WTOP
PRINT  TTOT, WYAW, CENC, EART, WTOW, RTIM, WRGR, CBPT, ETRN, W1
PRINT  YKWH, WPTL, CYAW, EGEN, W9  , AKW , WRTG, CRGR, ECBL, FR01
PRINT  EPLT, WCAB, CPTL, ETRF, FEN1, ABLD, WEL2, CCTR, PRAR, FT01
PRINT  AENC, WLAD, CLAD, PRA4, FHZ1, ATOW, WPTS, CSTL, PRA3, MZ1
PRINT  ACBL, WCHD, CFND, PRA2, FR02, HTOW, WDIA, CTOW, PRAT, FEN2
PRINT  RTAS, WBRC, CINR, PL09, FT02, DSID, WGUS, CINP, PL03, FHZ2
PRINT  HCLR, WMSC, CIND, PL02, MZ2 , RVV , WBAS, CINE, PL01, MY2
PRINT  KTIP, WSTL, CINC,     , FHZ3, AT75, WTTW, CINS,     , NFQB
PRINT  KCTS, WWGS, CLND,     , NFQT,     ,     , CCLR,     ,
READ
RUN MOD INPUT
IF END IS EQ 1, GO TO STOP
GO TO BGIN
POINT STOP
```

DATA

```
 RVV =  1.500000E 00
VBAR =  1.800000E 01
 SOL =  3.000003E-02
PRAT =  1.500000E 03
NGEN =  1.800000E 03
NBLD =  2.000000E 00
 DIA =  1.800000E 02
ETRN =  9.700003E-01
DHUB =  7.000000E 00
ACBL =  1.000000E 00
RDEN =  1.000000E 00
KTIP =  8.000002E-01
RTAS =  4.000000E 00
RCUT =  1.000000E-01
HCLR =  5.000000E 01
```

Figure B-1. ZODIAC II WGS Computer Program (continued)

TABLE EAER SIZE = 5, 34

Y	X: 2.0000E-02	3.0000E-02	5.0000E-02	1.0000E-01	2.0000E-01
0.0	0.0	0.0	0.0	0.0	0.0
1.0000E 00	1.0000E-02	2.2000E-02	3.5000E-02	5.6000E-02	1.0500E-01
2.0000E 00	3.5000E-02	5.5000E-02	8.7000E-02	1.4200E-01	2.3300E-01
3.0000E 00	7.0000E-02	1.0500E-01	1.6000E-01	2.7000E-01	3.5000E-01
3.5000E 00	9.4000E-02	1.3600E-01	2.0600E-01	3.2000E-01	3.7000E-01
4.0000E 00	1.2000E-01	1.7200E-01	2.5500E-01	3.5000E-01	3.8000E-01
4.5000E 00	1.5000E-01	2.0800E-01	3.0000E-01	3.6800E-01	3.7500E-01
5.0000E 00	1.7800E-01	2.4500E-01	3.3200E-01	3.7800E-01	3.6500E-01
5.5000E 00	2.1000E-01	2.8000E-01	3.5500E-01	3.8000E-01	3.5000E-01
6.0000E 00	2.3800E-01	3.1300E-01	3.7000E-01	3.7500E-01	3.3000E-01
6.5000E 00	2.6600E-01	3.4000E-01	3.8000E-01	3.6800E-01	3.0000E-01
7.0000E 00	2.9100E-01	3.5500E-01	3.8000E-01	3.5300E-01	2.5800E-01
7.5000E 00	3.1400E-01	3.6800E-01	3.7700E-01	3.3500E-01	2.0300E-01
8.0000E 00	3.3600E-01	3.7600E-01	3.7000E-01	3.1000E-01	1.4000E-01
8.5000E 00	3.5500E-01	3.8000E-01	3.5600E-01	2.7300E-01	6.0000E-02
9.0000E 00	3.6800E-01	3.8000E-01	3.3500E-01	2.2700E-01	-2.5000E-02
9.5000E 00	3.7600E-01	3.7300E-01	3.0700E-01	1.7700E-01	-1.0000E-01
1.0000E 01	3.8000E-01	3.6300E-01	2.7000E-01	1.2300E-01	0.0
1.0500E 01	3.7500E-01	3.4800E-01	2.2800E-01	6.3000E-02	0.0
1.1000E 01	3.6600E-01	3.2700E-01	1.8500E-01	0.0	0.0
1.1500E 01	3.5400E-01	3.0000E-01	1.4000E-01	-6.0000E-02	0.0
1.2000E 01	3.3500E-01	2.6300E-01	9.8000E-02	-1.1500E-01	0.0
1.2500E 01	3.1300E-01	2.2400E-01	5.2000E-02	0.0	0.0
1.3000E 01	2.8600E-01	1.8500E-01	1.0000E-02	0.0	0.0
1.3500E 01	2.5000E-01	1.4300E-01	-3.3000E-02	0.0	0.0
1.4000E 01	2.1000E-01	1.0300E-01	-7.6000E-02	0.0	0.0
1.4500E 01	1.7000E-01	6.3000E-02	0.0	0.0	0.0
1.5000E 01	1.3000E-01	2.3000E-02	0.0	0.0	0.0
1.5500E 01	9.0000E-02	-1.7000E-02	0.0	0.0	0.0
1.6000E 01	5.0000E-02	-5.6000E-02	0.0	0.0	0.0
1.6500E 01	1.0000E-02	-9.5000E-02	0.0	0.0	0.0
1.7000E 01	-2.7000E-02	0.0	0.0	0.0	0.0
1.7500E 01	-6.7000E-02	0.0	0.0	0.0	0.0
3.0000E 02	0.0	0.0	0.0	0.0	0.0

Figure B-1. ZODIAC II WGS Computer Program (continued)

```
TABLE TWND  SIZE =  47,  8
      Y                X
                0.0           1.0000E 00   2.0000E 00   3.0000E 00   4.0000E 00   5.0000E 00   6.0000E 00   7.0000E 00
                8.0000E 00    9.0000E 00   1.0000E 01   1.1000E 01   1.2000E 01   1.3000E 01   1.4000E 01   1.5000E 01
                1.6000E 01    1.7000E 01   1.8000E 01   1.9000E 01   2.0000E 01   2.1000E 01   2.2000E 01   2.3000E 01
                2.4000E 01    2.5000E 01   2.6000E 01   2.7000E 01   2.8000E 01   2.9000E 01   3.0000E 01   3.1000E 01
                3.2000E 01    3.3000E 01   3.4000E 01   3.5000E 01   3.6000E 01   3.7000E 01   3.8000E 01   3.9000E 01
                4.0000E 01    4.1000E 01   4.2000E 01   4.3000E 01   4.4000E 01   4.5000E 01   4.6000E 01

   7.5000E 00   8.7600E 03    8.4000E 03   7.9500E 03   7.3000E 03   6.6500E 03   6.0000E 03   5.3000E 03   4.5500E 03
                3.9500E 03    3.2000E 03   2.5500E 03   2.0000E 03   1.5000E 03   1.1500E 03   8.5000E 02   6.0000E 02
                4.0000E 02    3.0000E 02   2.0000E 02   1.5000E 02   1.0000E 02   5.0000E 01   1.0000E 01   0.0
                0.0           0.0          0.0          0.0          0.0          0.0          0.0          0.0
                0.0           0.0          0.0          0.0          0.0          0.0          0.0          0.0
                0.0           0.0          0.0          0.0          0.0          0.0          0.0
   1.0000E 01   8.7600E 03    8.5500E 03   8.3000E 03   8.0400E 03   7.6750E 03   7.1500E 03   6.5500E 03   6.0000E 03
                5.4500E 03    4.9000E 03   4.3250E 03   3.7750E 03   3.2500E 03   2.6500E 03   2.1500E 03   1.6250E 03
                1.2600E 03    9.8000E 02   7.4000E 02   5.2500E 02   3.8000E 02   2.7500E 02   1.9000E 02   1.2500E 02
                7.5000E 01    4.0000E 01   1.0000E 01   0.0          0.0          0.0          0.0          0.0
                0.0           0.0          0.0          0.0          0.0          0.0          0.0          0.0
                0.0           0.0          0.0          0.0          0.0          0.0          0.0
   1.2000E 01   8.7600E 03    8.6000E 03   8.4000E 03   8.1750E 03   7.9150E 03   7.6000E 03   7.2250E 03   6.7800E 03
                6.3050E 03    5.8200E 03   5.3250E 03   4.8250E 03   4.3200E 03   3.8150E 03   3.3150E 03   2.8250E 03
                2.3500E 03    1.9000E 03   1.5400E 03   1.2400E 03   9.8000E 02   7.6000E 02   5.8000E 02   4.3000E 02
                3.0500E 02    2.0500E 02   1.3000E 02   7.5000E 01   3.5000E 01   1.0000E 01   0.0          0.0
                0.0           0.0          0.0          0.0          0.0          0.0          0.0          0.0
                0.0           0.0          0.0          0.0          0.0          0.0          0.0
   1.5000E 01   8.7600E 03    8.6500E 03   8.5300E 03   8.3500E 03   8.2000E 03   7.9500E 03   7.7500E 03   7.5000E 03
                7.2000E 03    6.9000E 03   6.5000E 03   6.0500E 03   5.6000E 03   5.2000E 03   4.7500E 03   4.3500E 03
                3.9000E 03    3.4500E 03   3.0500E 03   2.6000E 03   2.2000E 03   1.9000E 03   1.6000E 03   1.3500E 03
                1.1000E 03    9.0000E 02   7.5000E 02   6.0000E 02   5.0000E 02   4.0000E 02   3.0000E 02   2.0000E 02
                1.5000E 02    1.0000E 02   7.0000E 01   4.0000E 01   1.0000E 01   0.0          0.0          0.0
                0.0           0.0          0.0          0.0          0.0          0.0          0.0
   1.8000E 01   8.7600E 03    8.6750E 03   8.5600E 03   8.4500E 03   8.3250E 03   8.2000E 03   8.0500E 03   7.8500E 03
                7.6500E 03    7.4250E 03   7.1500E 03   6.8400E 03   6.4600E 03   6.1250E 03   5.7750E 03   5.4000E 03
                5.0500E 03    4.7250E 03   4.3750E 03   4.0250E 03   3.6750E 03   3.3250E 03   2.9750E 03   2.6250E 03
                2.3000E 03    2.0250E 03   1.7600E 03   1.5200E 03   1.3000E 03   1.1100E 03   9.5000E 02   8.0000E 02
                6.8000E 02    5.7500E 02   4.7500E 02   3.9000E 02   3.2000E 02   2.6000E 02   2.0000E 02   1.6000E 02
                1.2000E 02    8.0000E 01   5.5000E 01   3.0000E 01   1.0000E 01   0.0          0.0
   2.0000E 01   8.7600E 03    8.7000E 03   8.6250E 03   8.5300E 03   8.4250E 03   8.3000E 03   8.1750E 03   8.0000E 03
                7.8250E 03    7.6400E 03   7.4250E 03   7.1750E 03   6.8750E 03   6.5500E 03   6.2500E 03   5.9300E 03
                5.6250E 03    5.3250E 03   5.0000E 03   4.7000E 03   4.3800E 03   4.1750E 03   3.7700E 03   3.4500E 03
                3.1500E 03    2.8250E 03   2.5400E 03   2.2600E 03   2.0000E 03   1.7600E 03   1.5500E 03   1.3600E 03
                1.1900E 03    1.0300E 03   8.9000E 02   7.7000E 02   6.6000E 02   5.7000E 02   4.9000E 02   4.2500E 02
                3.6000E 02    3.1000E 02   2.6000E 02   2.2000E 02   1.8000E 02   1.5000E 02   0.0
   2.2000E 01   8.7600E 03    8.7250E 03   8.6600E 03   8.6000E 03   8.5100E 03   8.4100E 03   8.3000E 03   8.1800E 03
                8.0250E 03    7.8600E 03   7.6750E 03   7.4600E 03   7.2250E 03   6.9750E 03   6.6800E 03   6.3800E 03
                6.1200E 03    5.8300E 03   5.5500E 03   5.2800E 03   5.0000E 03   4.7300E 03   4.4500E 03   4.1750E 03
                3.9000E 03    3.6250E 03   3.3500E 03   3.0750E 03   2.8000E 03   2.5500E 03   2.3000E 03   2.0800E 03
                1.8750E 03    1.6800E 03   1.5000E 03   1.3400E 03   1.1800E 03   1.0300E 03   9.0000E 02   7.7500E 02
                6.7500E 02    6.0000E 02   5.2500E 02   4.7000E 02   4.1000E 02   3.6000E 02   0.0
   2.5000E 01   8.7600E 03    8.7500E 03   8.7000E 03   8.6500E 03   8.6000E 03   8.5000E 03   8.4000E 03   8.3000E 03
                8.2000E 03    8.1000E 03   7.9500E 03   7.7500E 03   7.5500E 03   7.3500E 03   7.1500E 03   6.9000E 03
                6.6500E 03    6.4000E 03   6.1500E 03   5.9000E 03   5.6500E 03   5.4500E 03   5.1500E 03   4.9500E 03
                4.7000E 03    4.4500E 03   4.2000E 03   3.9500E 03   3.7000E 03   3.5000E 03   3.3000E 03   3.0500E 03
                2.8500E 03    2.6500E 03   2.4500E 03   2.3000E 03   2.1500E 03   1.9500E 03   1.8000E 03   1.6500E 03
                1.5000E 03    1.4000E 03   1.2500E 03   1.1500E 03   1.0500E 03   9.5000E 02   0.0
```

Figure B-1. ZODIAC II WGS Computer Program (continued)

```
EQUATIONS
    MODULE NAME = CONSTANTS
    COMMON PLO1,PLO2,PLO3,PLO9,PRAR,PRA2,PRA3,PRA4,PRAT,ETRN,ECBL,HTOW,ACBL
    COMMON DRAD,DIA,HCLR,HTOW.
    DRAD=DIA/2
    HTOW=HCLR+DRAD
    COMMON EGEN,ETRF
    PLO1 = PRAT/(15.32*PRAT**.21-1)
    ETRF = PRAT/(PRAT+PLO1)
    PRA2 = PRAT+PLO1
    PLO2 = .0033*(HTOW+200)/ACBL
    PRA3 = PRA2+PLO2
    ECBL = PRA2/PRA3
    PLO3 = PRA3/(4.53*PRA3**.215-1)
    EGEN = PRA3/(PRA3+PLO3)
    PRA4 = PRA3+PLO3
    PLO9 = .25*PRAT**.36
    PRAR = PRA4/ETRN+PLO9
```

```
EQUATIONS
    MODULE NAME = EFFECTIVE VELOCITY
    COMMON REWS,DRAD,DIA,AROT,HCLR,RCUT,HTOW,VTIP,VRAT,RAMX,SOL,KTIP,RVV,VBAR
    VRAT=VBAR*RVV
    RCLR=HCLR/DRAD
    KEFF=.9778+.1931*RCLR-.0528*RCLR*RCLR
    REWS=(DRAD/30)**.17*KEFF
    AROT = 3.14159*DRAD*DRAD
    RVMX=-3.261*SOL**2+1.491*SOL+.0741
    RAMX=1/RVMX
    VTIP=VRAT*REWS*1.4677*KTIP*RAMX
```

```
EQUATIONS
    MODULE NAME= ROTOR POWER
    COMMON DIA,VTIP,VWND,REWS,AROT,PROT,SOL,RDEN,EAER
    VEFF = 1.4677*VWND*REWS
    RADV=VTIP/VEFF
    EAER = TABLE EAER (SOL,RADV)
    PROT = 1.612E-6*EAER*RDEN*AROT*VEFF**3
```

```
EQUATIONS
    MODULE NAME=RATED WIND
    COMMON PRAR,PROT,RVV
    RVV=.25*RVV*(3+PRAR/PROT)
```

Figure B-1. ZODIAC II WGS Computer Program (continued)

```
EQUATIONS
   MODULE NAME = TRANSMISSION POWER
   COMMON PTRN,PROT,ETRN,PLO9
   PTRN=ETRN*(PROT-PLO9)

EQUATIONS
   MODULE NAME = GEN POWER
   COMMON PTRN,PGEN,PLO3,PRA3
   INITIALIZE DELT=0
   PGEN=PTRN-DELT
   P=PGEN/PRA3
   DELT=PLO3*(.511+.053*P+.436*P*P)
   R=DELT/PRA3
   ITERATE ON R,ATOL=.001 ,TIMES=20

EQUATIONS
   MODULE NAME = CABLE POWER
   COMMON PGEN,PCBL,PRA2,PLO2
   INITIALIZE DELT=0
   PCBL=PGEN-DELT
   P=PCBL/PRA2
   DELT=PLO2*P*P
   R=DELT/PRA2
   ITERATE ON R,ATOL=.001 ,TIMES=20

EQUATIONS
   MODULE NAME = TRANSFORMER POWER
   COMMON PCBL,PTRF,PRAT,PLO1
   INITIALIZE DELT = 0
   PTRF=PCBL-DELT
   P=PTRF/PRAT
   DELT=PLO1*(.2+.8*P*P)
   R=DELT/PRAT
   ITERATE ON R,ATOL=.001 ,TIMES=20

EQUATIONS
   MODULE NAME =INIT TO
   COMMON TO,VWND,VBAR,PTRF
   V = VWND-0.5
   TO = TABLE TWND (V,VBAR)

EQUATIONS
   MODULE NAME = INTEGRATE
   COMMON POUT
   COMMON VWND,PT,TO,TIME,VBAR,POUT
   V = VWND+0.5
   TI=TABLE TWND(V,VBAR)
   TIME=TO-TI
   TO=TI
   PT = POUT * TIME
```

Figure B-1. ZODIAC II WGS Computer Program (continued)

EQUATIONS

```
MODULE NAME = TOP WEIGHT
COMMON VTIP,DRAD,SOL,NBLD,RCUT,PRAT,EGEN,NGEN,REWS,HCLR ,PRA4
COMMON DCRD,RASP,DBLD,NROT,HTOW,DSID,QGEN,QROT,DFLP,PRA3,PRAR,PRA2,DIA
COMMON WTOP,ACBL,RSID,ATOW,D1,AENC,ABLD
COMMON HTOW,      STS,FHRZ,      DHUB,      MY,WRTG,WRGR,WCAB,WEL2,RTRN
COMMONWBLD,DBLD,WPCH,WRCL,WHSG,WSFT,WCP1,WTRN,WCP2,WCLH,WBPT,WYAW
COMMON WBLD,WFLP,WBLS,WPCH,WRCL,WHSG,WHUB,WROT,WSFT,WTRN,WCP1,WCP2,WBRK
COMMON WCLH,WSLP,WDRS,WGEN,WEL1,WELE,WCTR,WENC,WBPT,WYAW,WPTL,WRTG,WLAD
DCRD=3.14159*SOL*DRAD/NBLD
DCRD=1.443*DCRD
NROT=9.5493*VTIP/DRAD
RTRN=NGEN/NROT
DBLD=DRAD*(1-RCUT)
RASP=DBLD/DCRD
QROT=7.04*PRAR/NROT
QGEN=7.04*PRA4/NGEN
K9=1-(1-(2.772-.2314*RASP+.00651*RASP*RASP))*(2.5-.02222*DBLD)
WFLP=0
WBLD=.001*(.00091*DBLD**3.53)*K9
WBLS=NBLD*(WBLD+WFLP)
WHUB=1.187*(WBLD*(RCUT*DRAD))**.75
WPCH=WHUB*(.59-.68*RCUT)
WRCL=WHUB*(.09-.14*RCUT)
WHSG=WHUB-WPCH-WRCL
WROT=WBLS+WHUB
WSFT=.2825*QROT**.71
WTRN=.102*QROT**.93
WCP1=.014*QROT**.955
WCP2=.0147*QGEN**.93
WBRK=.15
WCLH=.0115*QGEN
WSLP=0
WDRS=WSFT+WCP1+WTRN+WCP2+WBRK+WCLH+WSLP
WGEN=.0426*PRA3**.79+PRA3*(1800-NGEN)*4E-7
WEL1=.0095*PRA3**.5
WELE=WGEN+WEL1
WCTR=.25+.000127*DIA**1.77
WRDG=WROT+WDRS+WELE+WCTR
WENC=.5325*WRDG**.51
WBPT=.3684*WRDG**.5*PRAT**.3
WYAW=.1656*WRDG**1.026
WPTL=WENC+WBPT+WYAW
WRTG=WRDG+WPTL
WLAD=.0564*HTOW
WCAB=.004*HTOW*ACBL
WEL2=.003*HTOW
WRGR=.36*WRTG**.27
WTOP=WRTG+WRGR
ABLD=DRAD*DCRD*NBLD
ABLD=(7/8)*ABLD
AENC=15*WRDG**.666
```

Figure B-1. ZODIAC II WGS Computer Program (continued)

```
EQUATIONS
MODULE NAME = TOWER WEIGHT
COMMON STS,K2,NFQB,NFQT,IP,DSID,ATOW
COMMON AT75,KCTS,FRO2,FEN2,FHZ2,MY2,MZ2,FRO1,FEN1,FTO1,FHZ1,MZ1,FHZ3,FTO2,VRAT
COMMON NROT,RTAS,REWS,ABLD,AENC,ATOW,DHUB,SOL,HTOW,VTIP,DRAD,RDEN
COMMON W1C,W1D,W1B,W2C,W2D,W2B,W3C,W3D,W3B,WC,WD,WB,W9,W4C,W5D,W5B
COMMON WGUS,WPTS,WBAS,WMSC,WCHD,WDIA,WBRC.,W1
COMMON WTOP,WTOW,WRTG,WRGR,WSTL,WTTW,WWGS,WEL2,WCAB,WLAD
INITIALIZE WTOW = 0
ITERATE ON WTOW,PTOL=0.1 ,TIMES=10
STS=18
K2=2
DSID=HTOW/RTAS
ATOW=.15*HTOW*DSID
D1=HTOW**2+DSID**2/2
IP=11+.00158*WTOP**2.4
W1=WTOP+WTOW/2
MY = 0
K3=0
K1=1.333
V = 120
SDYN=.00119*(V*REWS*1.4677)**2
FRO1 = .0012*SDYN*ABLD
FEN1 = .0012*SDYN*AENC
FTO1 = .0015*SDYN*ATOW
FHZ1 = FRO1+FEN1+FTO1/2
FHRZ = FHZ1
MZ1 = DHUB*FRO1
MZ = MZ1
  W2 = 3.4028*D1*W1/HTOW
  W3 = 9.6245*D1/DSID*(FHRZ/K1+K3*W1/1.333)
  W4 = 19.249*D1**.5*MY/(DSID*K1)
  W5 = 16.333*D1**.5*MZ/(DSID*K1)
 W1C = .00105*(W2+W3+W4+W5)/STS
 W1D =1.1*.054444*HTOW*MZ/(DSID*K1*STS)
 W1B = 2*.013611*HTOW*MZ/(DSID*K1*STS)
K3 =.25
K1 = 1.0
V = VRAT
VEFF = V*REWS*1.4677
SDYN=.00119*(V*REWS*1.4677)**2
RADV=VTIP/VEFF
AT75 = 50*SOL
KCTS = .067+.9/RADV -.007*AT75 -.8*SOL
FRO2 = KCTS*3.14*DRAD*DRAD*SOL*VTIP**2*.002378*RDEN/1000
FEN2 = .0012*SDYN*AENC
FTO2 = .0015*SDYN*ATOW
FHZ2 = FRO2+FEN2+FTO2/2
FHRZ = FHZ2
MY2 = .018*DRAD*DRAD
MZ2 = .022*DRAD*DRAD
MY =MY2
MZ = MZ2
  W2 = 3.4028*D1*W1/HTOW
  W3 = 9.6245*D1/DSID*(FHRZ/K1+K3*W1/1.333)
  W4 = 19.249*D1**.5*MY/(DSID*K1)
  W5 = 16.333*D1**.5*MZ/(DSID*K1)
 W2C = .00105*(W2+W3+W4+W5)/STS
 W2B = 2*.013611*HTOW*MZ/(DSID*K1*STS)
 W2D = 1.1*.054444*HTOW*MZ/(DSID*K1*STS)
```

Figure B-1. ZODIAC II WGS Computer Program (continued)

```
EQUATIONS CONT.
 60  K3 = 0
 61  K1 = 1.0
 62  FHZ3 = 2*FRO2+FEN2+FTO2/2
 63  FHRZ = FHZ3
 64  MY = 2*MY2
 65  MZ = 2*MZ2
 66    W2 = 3.4028*D1*W1/HTOW
 67    W3 = 9.6245*D1/DSID*(FHRZ/K1+K3*W1/1.333)
 68    W4 = 19.249*D1**.5*MY/(DSID*K1)
 69    W5 = 16.333*D1**.5*MZ/(DSID*K1)
 70   W3C = .00105*(W2+W3+W4+W5)/STS
 71   W3D = 1.1*.054444*HTOW*MZ/(DSID*K1*STS)
 72   W3B = 2*.013611*HTOW*MZ/(DSID*K1*STS)
 73  WC = W1C
 74  W   =W2C
 75  IF W3C IS GT W,W=W3C
 76  IF W IS GT WC,WC=W
 77  WD = W1D
 78  W=W2D
 79  IF W3D IS GT W2D,W=W3D
 80  IF W IS GT WD,WD=W
 81  WB = W1B
 82  W=W2B
 83  IF W3B IS GT W,W=W3B
 84  IF W IS GT WB,WB =W
 85  W9 = WC+WD+WB
 86  WGUS = .1*W9
 87  WPTS = .1*W9
 88  WBAS = .1*W9
 89  WMSC = .1*W9
 90  NFQB=1.5*NROT
 91  W4C = 1.5452 E-10*(WTOP+WTOW/3)*NFQB**2*K2*D1**2/DSID**2
 92  NFQT=2.5*NROT
 93  W5D = 1.1*.17926 E-6*K2*IP*(HTOW*NFQT/DSID)**2
 94  W5B = 2*.04484 E-6*K2*IP*(HTOW*NFQT/DSID)**2
 95  WCHD = WC
 96    IF W4C IS GT WC,WCHD=W4C
 97  WDIA = WD
 98  IF W5D IS GT WD,WDIA=W5D
 99  WBRC = WB
100  IF W5B IS GT WB,WBRC=W5B
101  WTOW = WCHD+WDIA+WBRC+WGUS+WMSC+WEL2+WCAB+WLAD
102  WTOP=WRTG+WRGR+WPTS
103  WSTL = WCHD+WDIA+WBRC+WGUS+WBAS+WMSC+WPTS
104  WTTW = WSTL+WRGR+WEL2+WCAB+WLAD
105  WWGS = WTTW+WRTG
```

Figure B-1. ZODIAC II WGS Computer Program (continued)

```
EQUATIONS
 MODULE NAME = COST
 COMMON WSTL,   HTOW,          WTOP,WTOW,    DSID,NBLD,DIA,PRAT,NGEN
 COMMONWBLD,CBLD,WPCH,WRCL,WHSG,WSFT,WCP1,WTRN,WCP2,WCLH,WBPT,WYAW
 COMMON WENC,WRGR,WFLP,      PRA2,PRA3,ACBL,FHZ1,FHZ2,FHZ3,MY2
 COMMON CBLD,CFLP,CBLS,CPCH,CRCL,CHSG,CHUB,CRCT,CSFT,CTRN,CCP1,CCP2,CBRK
 COMMON CCLH,CSLP,CDRS,CGEN,CCAB,CTRF,CEL3,CEL4,CELC,CENC,CBPT,CYAW,CPTL
 COMMONCCTR,CRGR,CSTL,CFNC,CTOW,CINS,CLND,CCLR,CFNC,CSHD,CPAD,CSIT,CDIR
 COMMON CMAR,CMAP,CMAT,CMAS,CMNT,COPS,COAM,CCAR,CYR,CKW
 COMMON CINR,CINP,CIND,CINE,CINC,CLAD
 CBLD=WBLD*23.5*1000+CBLD*62.1
 CFLP=0
 CBLS=NBLD*(CBLD+CFLP)
 CPCH=WPCH*2500
 CRCL=WRCL*2000
 CHSG=WHSG*3900
 CHUB=CPCH+CRCL+CHSG
 CRCT=CBLS+CHUB
 CSFT=WSFT*1000*(.20*2.00+.53*4.50+.27*2.00)
 CTRN=WTRN*2700
 CCP1=WCP1*4560
 CCP2=WCP2*4560
 CBRK=1500
 CCLH=WCLH*10150
 CSLP=0
 CDRS=CSFT+CCP1+CTRN+CCP2+CBRK+CCLH+CSLP
 CGEN=72.65*PRA3**.854+.01*PRA3*(1800-NGEN)
 CCAB=22.25*(HTOW+200)*ACBL**.72
 CTRF=54.23*PRAT**.74
 CEL3=21210
 CEL4=30*HTOW+1920
 CELC=CGEN+CCAB+CTRF+CEL3+CEL4
 CENC=WENC*2000
 CBPT=WBPT*660
 CYAW=WYAW*2700
 CRGR=WRGR*2700
 CPTL=CENC+CBPT+CYAW+CRGR
 CCTR=11350
 CLAD=106*HTOW
 CSTL=WSTL*(550+20)
   F1=1.0607*(HTOW*(FHZ1)                          )/DSID-(WTOP+WTOW)/4
   F2=1.0607*(HTOW*(FHZ2+.25*(WTOP+WTOW/2))+  MY2)/DSID-(WTOP+WTOW)/4
   F3=1.0607*(HTOW* FHZ3                     +2*MY2)/DSID-(WTOP+WTOW)/4
   K4=F1
   IF F2 IS GT K4,K4=F2
   IF F3 IS GT K4,K4=F3
   K5=1+2.19/(K4**(1/3))
 CFNC=.7055*K4*145*K5
 CTOW=CSTL+CFNC+CLAD
 CINR=3700
 CINP=2700
 CIND=4000
 CINE=1000+.133*(CELC-20000)
 CINC=2900
 CINS=CINR+CINP+CIND+CINE+CINC
 ALND=(5*DIA)**2
 CLND=ALND*500/43560
 ACLR=(3*DIA)**2
 CCLR=ACLR*1500/43560
 DFND=4*DIA
```

Figure B-1. ZODIAC II WGS Computer Program (continued)

```
EQUATIONS CONT.
CFNC=12*CFNC
CSHC=1600
CPAD=330
CSIT=CLND+CCLR+CFNC+CSHC+CPAD
CCIR=CRCT+CCRS+CELC+CPTL+CCTR+CTOW+CINS+CSIT
CCWN=.25*CCIR
CMAP=.035*(CCRS+CELC+CCTR)
CMAR=.1*CRCT
CMAT=.005*(CPTL+CTOW)
CMAS=CSIT*.01
CMNT=CMAR+CMAP+CMAT+CMAS
CCPS=.015*CCIR
CCAM=CCPS+CMNT
CCAR=.15*CCIR
CYR=CCAM+CCAR
CKW=CCIR/PRAT
```

```
EQUATIONS
MODULE NAME=PLANT EFFICIENCY
COMMON TTOT,KWHR,PRAT,RTIM,AKW,EPLT,YKWH,KWHK,CYR,CKWH
RTIM=TTOT/8760
KWHK=8760*(1.3+.015*PRAT**.5)
YKWH=KWHR-KWHK
AKW=YKWH/8760
EPLT=AKW/PRAT
CKWH=CYR/YKWH
```

```
EQUATIONS
MODULE NAME = INPUT
COMMON PRAT,RVV,SOL,VBAR,HCLR,KTIP,END,DIA
```

Figure B-1. ZODIAC II WGS Computer Program (continued)

```
EQUATIONS
  MODULE NAME = TOP WEIGHT
  COMMON VTIP,DRAD,SCL,NBLD,RCUT,PRAT,EGEN,NGEN,REWS,HCLR ,PRA4
  COMMON CCRD,RASP,DBLD,NROT,HTCW,DSID,QGEN,QROT,DFLP,PRA3,PRAR,PRA2,DIA
  COMMON WTOP,ACBL,RSID,ATOW,CI,AENC,ABLD
  COMMON HTCW,      STS,FHRZ,      DHUB,      MY,WRTG,WRGR,WCAB,WEL2,RTRN
  COMMONWBLD,DBLD,WPCH,WRCL,WHSG,WSFT,WCP1,WTRN,WCP2,WCLH,WBPT,WYAW
  COMMON WBLD,WFLP,WBLS,WPCH,WRCL,WHSG,WHUB,WROT,WSFT,WTRN,WCP1,WCP2,WBRK
  COMMON WCLH,WSLP,WDRS,WGEN,WEL1,WELE,WCTR,WENC,WBPT,WYAW,WPTL,WRTG,WLAD
  COMMON WGRP,WBRL
  DCRD=3.14159*SCL*DRAD/NBLD
  DCRD=1.443*CCRD
  NROT=9.5493*VTIP/DRAD
  RTRN=NGEN/NROT
  DBLD=DRAD*(1-RCUT)
  RASP=DBLD/CCRD
  QROT=7.04*PRAR/NROT
  QGEN=7.04*PRA4/NGEN
  WFLP=0
  WBLD=.000017*DBLD**2.86
  WRCL=.000107*DIA**2
  WBRL=17*QROT**.83/DIA
  WGRP=.00168*WBRL**1.48*DIA*(1+.1*DIA*(RCUT-.05))
  WHUB=WRCL+WBRL+WGRP
  WBLS=NBLD*WBLD
  WROT=WHUB+WBLS
  WSFT=.2825*QROT**.71
  WTRN=.102*QROT**.93
  WCP1=.014*QROT**.955
  WCP2=.0147*QGEN**.93
  WBRK=.15
  WCLH=.0115*QGEN
  WSLP=0
  WDRS=WSFT+WCP1+WTRN+WCP2+WBRK+WCLH+WSLP
  WGEN=.0426*PRA3**.79+PRA3*(1800-NGEN)*4E-7
  WEL1=.0095*PRA3**.5
  WELE=WGEN+WEL1
  WCTR=.25+.000127*DIA**1.77
  WRDG=WROT+WDRS+WELE+WCTR
  WENC=.5325*WRDG**.51
  WBPT=.3684*WRDG**.5*PRAT**.3
  WYAW=.1656*WRDG**1.026
  WPTL=WENC+WBPT+WYAW
  WRTG=WRDG+WPTL
  WLAD=.0564*HTCW
  WCAB=.004*HTCW*ACBL
  WEL2=.003*HTCW
  WRGR=.36*WRTG**.27
  WTOP=WRTG+WRGR
  ABLD=DRAD*DCRD*NBLD
  ABLD=(7/8)*ABLD
  AENC=15*WRDG**.666
```

Figure B-2. Revisions to the Parametric Model to Reflect Preliminary Design Weight and Cost Estimates for the Rotor.

EQUATICNS

```
MCCULE NAME = CCST
CCMMCN  WSTL,   HTCW,          WTOP,WTOW,    DSID,NBLD,DIA,PRAT,NGEN
CCMMCNWBLC,CELC,WPCH,WRCL,WHSG,WSFT,WCP1,WTRN,WCP2,WCLH,WBPT,WYAW
CCMMCN WENC,WRGR,WFLP,        PRA2,PRA3,ACBL,FHZ1,FHZ2,FHZ3,MY2
CCMMCN CELC,CFLP,CELS,CPCH,CRCL,CHSG,CHUB,CRCT,CSFT,CTRN,CCP1,CCP2,CBRK
CCMMCN CCLH,CSLP,CCRS,CGEN,CCAB,CTRF,CEL3,CEL4,CELC,CENC,CBPT,CYAW,CPTL
CCMMCNCCTR,CRGR,CSTL,CFNC,CTCW,CINS,CLND,CCLR,CFNC,CSHD,CPAD,CSIT,CCIR
CCMMCN CMAR,CMAP,CMAT,CMAS,CMNT,COPS,COAM,CCAR,CYR,CKW
CCMMCN CINR,CINP,CINC,CINE,CINC,CLAC
CFLP=0
CCMMCN WGRP,WERL,CGRP,CERL
CBLC=9200*WBLC+101*CELC
CRCL=2620*WRCL
CERL=3000*WERL
CGRP=2250*WGRP
CHUB=CRCL+CERL+CGRP
CBLS=NBLC*CBLC
CRCT=CHUB+CELS
CSFT=WSFT*1000*(.20*2.00+.53*4.50+.27*2.00)
CTRN=WTRN*2700
CCP1=WCP1*4560
CCP2=WCP2*4560
CBRK=1500
CCLH=WCLH*10150
CSLP=0
CCRS=CSFT+CCP1+CTRN+CCP2+CBRK+CCLH+CSLP
CGEN=72.65*PRA3**.654+.01*PRA3*(1800-NGEN)
CCAB=22.25*(HTCW+200)*ACBL**.72
CTRF=54.23*PRAT**.74
CEL3=21210
CEL4=30*HTCW+1920
CELC=CGEN+CCAB+CTRF+CEL3+CEL4
CENC=WENC*2000
CBPT=WBPT*660
CYAW=WYAW*2700
CRGR=WRGR*2700
CPTL=CENC+CBPT+CYAW+CRGR
CCTR=11350
CLAC=106*HTCW
CSTL=WSTL*(550+20)
  F1=1.0607*(HTCW*(FHZ1)                         )/DSID-(WTOP+WTOW)/4
  F2=1.0607*(HTOW*(FHZ2+.25*(WTOP+WTOW/2))+  MY2)/DSID-(WTOP+WTOW)/4
  F3=1.0607*(HTOW* FHZ3                     +2*MY2)/DSID-(WTOP+WTOW)/4
  K4=F1
  IF F2 IS GT K4,K4=F2
  IF F3 IS GT K4,K4=F3
  K5=1+2.19/(K4**(1/3))
CFNC=.7055*K4*145*K5
CTCW=CSTL+CFNC+CLAC
CINR=3700
CINP=2700
CINC=4000
CINE=1000+.133*(CELC-20000)
CINC=2900
CINS=CINR+CINP+CINC+CINE+CINC
ALNC=(5*DIA)**2
CLNC=ALNC*500/43560
ACLR=(3*DIA)**2
CCLR=ACLR*1500/43560
```

Figure B-2. Revisions to the Parametric Model to Reflect Preliminary Design Weight and Cost Estimates for the Rotor (continued)

```
EQUATIONS CONT.
  60   DFNC=4*DIA
  61   CFNC=12*DFNC
  62   CSHD=1600
  63   CPAD=330
  64   CSIT=CLND+CCLR+CFNC+CSHD+CPAD
  65   CDIR=CRCT+CDRS+CELC+CPTL+CCTR+CTOW+CINS+CSIT
  66   CCWN=.25*CDIR
  67   CMAP=.035*(CDRS+CELC+CCTR)
  68   CMAR=.1*CRCT
  69   CMAT=.005*(CPTL+CTOW)
  70   CMAS=CSIT*.01
  71   CMNT=CMAR+CMAP+CMAT+CMAS
  72   CCPS=.015*CDIR
  73   CCAM=CCPS+CMNT
  74   CCAR=.15*CDIR
  75   CYR=CCAM+CCAR
  76   CKW=CDIR/FRAT
```

Figure B-2. Revisions to the Parametric Model to Reflect Preliminary Design Weight and Cost Estimates for the Rotor (continued)

APPENDIX C

FAILURE MODES AND EFFECTS ANALYSIS

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM________________________________ INITIALS & DATE______________

SUBSYSTEM:____

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Brief description of one potential mode of subsystem disruption or performance degradation which would result from one or more types of subsystem failure. This is to be a description of the effect of failure, not the failure itself, e.g., "locked in drive position" "free-wheeling", etc.

CRITICALITY

☐ SAFETY ☐ OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

A statement of the evidence or symptoms of disrupted or degraded performance that could be used to detect its occurrence remotely, e.g., "excessive vibration", "rotor overspeed", "reduced efficiency", etc. Visual evidence or symptoms that would require on-site observation should not be listed.

POSSIBLE FAILURE CAUSES

A list of all pertinent hardware failures that could result in the subsystem disruption or performance degradation described above, e.g., "sheared shaft", "coupling failures", "hydraulic leak", "bearing jam", etc.

COMPENSATING PROVISIONS

The method(s), either automatic or remotely employed, which can be used to shut down the system or otherwise prevent further damage to equipment and injury to personnel, e.g., "feather rotor", "dump hydraulic pressure", etc.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

The state of the system after the compensating action discussed above has been taken, and a statement of any damage or hazards expected to result from having taken that action, e.g., "rotor feathered and locked; control system jammed. Jammed controls prevent rotor from being properly oriented, therefore damage may result in high winds, if not repaired, etc.".

DISCUSSION

Further amplification or qualification of any of the above. For failure modes not amenable to detection and/or compensation, a statement of the reason(s) and any recommended items for design changes to reduce the probability of that failure. For failure modes for which fault detection and/or compensation may not be entirely effective, design recommendations concerning ways in which the effects of failure might be further mitigated or the compensating provisions improved.

WGS EQUIPMENT FAILURES

CRITICALITY DEFINITIONS

SAFETY CRITICAL

Uncompensated failure could cause major equipment damage and/or personnel hazard. Steps must be taken to detect failure, and provide compensating features and/or backup redundancy. If detection or compensation is not practical or possible, probability of failure must be reduced to acceptably low levels. Immediate shutdown and lockout of WGS is mandatory until repaired.

OPERATIONAL

Uncompensated failure may cause minor damage to associated equipment or minor personnel hazard. Will require shutdown and lockout of WGS until failure has been repaired. May deteriorate to more critical failure if not detected and shut down.

NON CRITICAL

Uncorrected failure does not affect ability to safely operate WGS. May result in reduced efficiency, inconvenience, etc., but will not damage other associated equipment. WGS may be allowed to continue operation, and failure repaired at next scheduled maintenance interval. Many failures in this class will not be detectable except by inspection at WGS site.

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/21/75

SUBSYSTEM ROTOR/CONTROLS (FULL DRIVE)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Blade Pitch hardover to full drive position or lockup near full drive or severed pitch link which releases both blades to full drive or flat pitch position while operating.

CRITICALITY

[X] SAFETY [ ] OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50 - items 6 & 7a & b

all modes

SYMPTOMS/METHOD OF DETECTION

(1) Power or speed variations while operating, which occur without proper response of pitch controls or servo amplifier.
(2) Pitch servo trim does not match wind condition.
(3) Overtorque and/or overspeed of system.
(4) Higher than normal control forces due to jam.
(5) Failure to satisfy SSC timeout checks during startup/shutdown sequence.

POSSIBLE FAILURE CAUSES

(1) Hydraulic control valve hardover jam
(2) Electrical hardover failure in servo loop.
(3) Sensor hardover failure.
(4) Hydraulic piston jams in drive position
(5) Open line from RPM or torque limit sensor or open feedback line.
(6) Fracture of main pitch control link or pitch hub
(7) Pitch Controls jam due to:

a. Jam of main pitch control link or hub mechanism.
b. Main blade pitch bearing jams

COMPENSATING PROVISIONS

(1) Independent overspeed and/or overtorque detection devices to initiate shutdown independent of pitch servo/sensor electrical failures.
(2) Purely mechanical emergency control over power device to force controls to feather. Control power for shutdown taken from stored energy in rotor so it is independent of external power source and emergency battery, and introduced into pitch system so it bypasses all failure except possibly 6 and 7 above.
(3) Measure pitch control forces as indication of incipient jam.

COMPENSATING PROVISIONS (continued)

(4) Very conservative design of pitch control to reduce probability of failure.
(5) Pitch rate limited hydraulically so hardover cannot damage system.
(6) Servo simulator monitors operation during standby, synch, and operate.
(7) SSC monitors operation during startup/shutdown sequences.
(8) Rotor can be stopped by turning 90° to wind with yaw mechanism.

EFFECT ON SYSTEM BASED ON ABOVE PROVISIONS

System will be forced to shut down and lockout. Must be repaired before restart. Compensating provisions should prevent any significant secondary damage to system.

DISCUSSION

Rotor should be able to be properly parked in all cases except possibly Causes 6 and 7, where it may have to be parked in flat pitch. It is designed to take 120 mph flat pitch wind load in any case.

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/21/75

SUBSYSTEM ROTOR/CONTROLS (FULL FEATHER)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Blade Pitch Hardover to Feather position or lockup in full feather position.

CRITICALITY

EST FREQUENCY (MTBF - YEARS)

☐ SAFETY ☒ OPERATIONAL 1 2 5 10 20 50 >50 items 6 & 7a, b

all modes

SYMPTOMS/METHOD OF DETECTION

1. Loss of power and deceleration of rotor while operating.
2. Inability to restart if shut down.
3. Abnormally high pitch control forces

POSSIBLE FAILURE CAUSES

1. Hydraulic Control valve Hardover Jam.
2. Electrical Hardover failure in servo loop.
3. Hydraulic piston jams in feather position.
4. Emergency feathering device jams in feather.
5. Emergency feathering device actuates inadvertently due to improper failure indication.
6. Pitch Control Jams due to:
   a. Jam of main pitch control link or hub mechanism.
   b. Fault sensor or SSC initiates shutdown sequence inadvertently.
7. Fault sensor or SSC initiates shutdown sequence inadvertently.

COMPENSATING PROVISIONS

1. None required - Safe type of failure mode.
2. Should still try to minimize to prevent unnecessary shutdowns.
3. Sensing of abnormally high pitch control forces helps to prevent ultimate jam of pitch control in less favorable position.
4. Pitch rate limited to prevent reverse thrust in case of hardover.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

This is inherently a safe type of failure, since it results in shutdown of the rotor if operating, and inability to start up if shut down. Hardover is rate limited by inherent design of pitch actuation system so that rate of hardover will not overstress system. Loss of operating time until repaired.

DISCUSSION

Regardless of cause, SSC will sense negative power flow and initiate normal shutdown sequence. If failure was inside SSC itself, then rotor will still shut down, but may not be properly parked (i.e., may be rotating at low RPM). (See Sheet C7 for this case).

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/21/75

SUBSYSTEM ROTOR/CONTROLS - (ONE BLADE FREE)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Blade Pitch failure such that controls are broken free and blade is released - (one blade only) - Other Blade still under control.

CRITICALITY

[X] SAFETY [ ] OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Very heavy vibration levels at tower head (1/rev.) if in standby, or if operating above rated power.
2. Sudden overtorque if operating above rated wind, or overspeed if in standby mode.
3. One blade moves to flat pitch position while other blade remains at commanded angle, while operating above rated power.
4. One blade released from feathered position if shut down, but no speedup of rotor (rotor lock will prevent startup).
5. Abnormally high pitch control forces.

POSSIBLE FAILURE CAUSES

1. Breakage or separation of blade pitch control, e.g., pitch horn breaks, control rod shears, or rod separates from pitch horn or main control hub. (One blade only).

COMPENSATING PROVISIONS

1. Sensing of control forces may provide advance warning before failure.
2. Other blade will likely be able to stop or slow down rotor if resulting vibratories and blade loads do not destroy system.
3. Other blade will be driven to feather at a controlled rate by control system.
4. Failing to flat pitch prevents immediate loss of failed blade.
5. Conservative design of pitch controls to reduce probability of this failure occurring.
6. Overspeed will initiate shutdown.
7. Excess vibration levels sensed and used to initiate shutdown.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Possible structural damage to rotor and/or tower, although good probability of survival if final design takes this type of failure into account from a dynamic loading standpoint. Careful inspection of structure in rotor, hub, pintle, and tower area needed before system is run again; plus isolate cause of failure and redesign to prevent recurrence.

DISCUSSION

Detailed analysis beyond scope of preliminary design required to determine if remaining blade can bring system to a stop or at least down to low RPM with other blade driving, without structural failure of rotor or tower. Normal moments on blades under operation are expected to drive blade pitch away from feather position if blade is released. This is safest direction since driving toward feather at a high rate would cause high flat plate loads and probably fracture blade at root. Pitch control linkages must be conservatively designed and lightly stressed to minimize this failure mode.

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/21/75

SUBSYSTEM ROTOR/CONTROLS (MID RANGE LOCKUP)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Blade pitch lockup in partially feathered position while operating.

CRITICALITY

☐ SAFETY ☒ OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50 items 3 & 4a, b
ALL MODES

SYMPTOMS/METHOD OF DETECTION

1. Large power or speed variations while operating, which occur without proper response of pitch controls or servo amplifier.
2. Abnormally large pitch control forces.

POSSIBLE FAILURE CAUSES

1. Hydraulic control valve jam in neutral or open valve coil.
2. Electrical failure, open wire, dead amplifier, loss of servo power.
3. Hydraulic piston jams in partial feather
4. Pitch Controls Jam due to:
   a. Jam of main pitch control link or hub mechanism
   b. Blade pitch bearing jams

COMPENSATING PROVISIONS

1. Generator Trips off if overtorqued beyond limit.
2. Independent RPM Limiter initiates shutdown (redundant).
3. Pure mechanical control overpower forces controls to feather (see sheet C1, Compensating Provision (2).
4. Conservative design of pitch controls reduces probability of failure.
5. Sensing of pitch control forces may provide advance warning of failure.
6. Also see provisions 5 through 8 of Sheet C1.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Similar to lockup in full drive described on Sheet C1.

DISCUSSION

None

SHEET C5

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/21/75

SUBSYSTEM ROTOR/CONTROLS (NORMAL SHUTDOWN, NO DECELERATION)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Failure of rotor to decelerate properly in RPM after receipt of shutdown command from normal shutdown source.

CRITICALITY

☐ SAFETY ☒ OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 (20) 50 >50

SYMPTOMS/METHOD OF DETECTION

1. SSC system indicates deceleration fault (failure of rotor to decelerate).
2. Large power or speed variations with no pitch control or servo amplifier response.
3. Main Breaker fails to trip within 5 sec of Normal shutdown command to SSC.

POSSIBLE FAILURE CAUSES

1. Feathering of Rotor not sufficient due to controls failure (see Sheet C4, causes 1, 3, and 4).
2. SSC or associated interface circuits fail to react to shutdown command (i.e., failure of SSC or associated circuits relating to shutdown).

COMPENSATING PROVISIONS

1. Symptoms (1) and (3) above will be detected and used to initiate emergency shutdown.
2. Symptom (2) will be used to initiate emergency shutdown (overtorque and RPM limit sensor).
3. Using yaw servo, rotor will be turned out of wind and rotor brake applied.
4. Also see compensating provisions 1 through 5 of sheet C4.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Loss of operating time until repaired - No hazard since all causes can be detected and overcome with above compensating provisions.

DISCUSSION

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/20/75

SUBSYSTEM ROTOR/CONTROLS (EMERGENCY SHUTDOWN, NO DECELERATION)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Failure of Rotor to decelerate properly after receipt of shutdown command from safety related sensor (via hydraulic pressure dump).

CRITICALITY

☒ SAFETY ☐ OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. SSC system indicates deceleration fault (failure of rotor to decelerate).
2. Large power or speed variations with no pitch response.
3. Buildup of pitch control forces in normal operation.

POSSIBLE FAILURE CAUSES

1. Jam of main pitch control link or hub mechanism.
2. Main blade pitch bearing jams.
3. Hydraulic Cylinder jams.
4. Both emergency feathering springs fail, or both cams do not engage (double failure).

COMPENSATING PROVISIONS

1. Sense pitch control forces for advance warning of jams.
2. Yaw servo can be used to turn rotor 90° to wind, and then stop with rotor brake.
3. Simple and conservative design of emergency feathering mechanism (springs and brakes are redundant).
4. No external power required. Feathering continues as long as rotor is rotating. Large forces overcome jams.
5. Emergency feathering device overpowers hydraulic cylinder.
6. Emergency system backed up by normal pitch actuator and SSC.
7. Cause (4) above, extremely unlikely because it involves double failure.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Possible damage if above provisions are not able to overcome pitch control jams.

SHEET C7

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/20/75

SUBSYSTEM DRIVE/ROTOR/CONTROLS (FAILS TO STOP COMPLETELY)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Failure of rotor to come to a complete stop after decelerating to low RPM during shutdown cycle and after applying rotor brake.

CRITICALITY

[ ] SAFETY [X] OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Rotor Brake limit switch does not trip in specified time.
2. Rotor does not stop in specified time after brake applied.
3. Blade pitch angle does not reach full feather and winds are high. (e.g. high residual torque combined with worn brake or degraded actuator).
4. Brake overheats due to continued rotation while applied.
5. Generator and Rotor RPM do not match.

POSSIBLE FAILURE CAUSES

1. Power Loss (failure of sta service and emergency power sources).
2. Brake Actuator: Electrical failure, Mechanical Jam, sheared drive train.
3. Rotor Brake failure (mechanical) degraded/sheared, etc.
4. Limit switch failure
5. Drive system: gearbox/shaft/coupling broken or sheared.
6. Feathering of rotor not sufficient due to controls failure (see sheet No. C4).
7. Failure of SSC system
8. Loss of power to Rotor Brake Actuator.
9. Brake worn or degraded.

COMPENSATING PROVISIONS

1. SSC automatically releases brake and reverts to lockout mode if rotation continues after brake is applied.
2. Mechanical/Electrical fault telemetered to master station.
3. Periodic inspection of brake/actuator to preclude degradation.
4. Brake highly redundant, multiple disc wet type which should not wear.
5. Emergency battery backup for station service power to rotor brake actuator.
6. Separate speed sensors on generator and rotor detect shaft/drive failure.
7. Brake friction reduces with increasing temperature to prevent burnout if brake remains applied.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Rotor may cumulate additional fatigue cycles due to gravity bending moment loads while rotating at low speed. Additional secondary damage not likely, even in high winds. Fatigue cycles may also cumulate due to flat plate loading since blades are not parked horizontally.

DISCUSSION

As long as blades are locked in pitch near the feathered position, it is impossible for the rotor to reach any significant RPM, even in very high winds. Slight reduction in rotor fatigue life, depending on how long system remains in this state, and size of wind loads.

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/20/75

SUBSYSTEM DRIVE/ROTOR/CONTROLS

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Inability to park rotor properly after stopping.

CRITICALITY EST FREQUENCY (MTBF - YEARS)

[ ] SAFETY [X] OPERATIONAL 1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Shaft angle does not reach parked position in prescribed time (SSC indicates blade parking failure).

POSSIBLE FAILURE CAUSES

1. Drive system - Gearbox jammed or Shaft bearings jammed or degraded.
2. Inching drive - Electrical Failure, Mechanical Jam, Sheared Drive.
3. SSC system failure.
4. Rotor shaft position indicator sensor failure/wiring failure.
5. Loss of Power to inching drive (Loss of station service power)

COMPENSATING PROVISIONS

1. SSC automatically reverts to lockout mode and reports failure to master station.
2. Periodic inspection of inching drive to preclude failures.
3. Overtemperature device on inching drive to prevent overheat/fire.
4. Rotor can withstand parking in any position if feathered.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Rotor may cumulate additional fatigue cycles due to wind loads while parked in random orientation. Additional secondary damage not likely even in high winds, because WGS is designed to take up to 120 MPH flat plate loads. Loss of operating time until repaired.

DISCUSSION

Restart not recommended till after repair, because of item (1) under possible causes, although risk may be small.

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/20/75

SUBSYSTEM CONTROLS

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Inadvertent shutdown command from any source - without lockout or with lockout.

CRITICALITY

☐ SAFETY ☒ OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. No power output with favorable wind conditions.
2. Failure indication via telemetry link.

POSSIBLE FAILURE CAUSES

1. Failure of any sensor which could normally initiate shutdown in a direction to cause shutdown of system.
2. Failure of SSC system into shutdown mode.
3. Failure of lockout relay to lockout mode.
4. Failure of hydraulic disengage valve to disengage position.

COMPENSATING PROVISIONS

Make above sensors and data link as reliable as possible; but sensors should be designed so that their normal failure mode is to induce a shutdown rather than fail to initiate a shutdown when one is actually needed.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Loss of operating time and generated output - No Hazard. System will generally remain shut down until repaired, unless non-lockout sensor fails intermittently.

DISCUSSION

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/20/75

SUBSYSTEM CONTROLS (LOSES LOCKOUT)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

SSC does not remain in lockout state after operational or safety critical shutdown.

CRITICALITY

[X] SAFETY [ ] OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Telemetry or dispatcher indicates power being delivered after failure indication, and no repair action has taken place.

POSSIBLE FAILURE CAUSES

1. Failure of both SSC and electrical lockout relays (double failure).
2. Maintenance error - operator resets lockout without repairing original failure.

COMPENSATING PROVISIONS

1. Backup Lockout Relays as part of Generator Protective Controls and SSC.
2. Most sensors pass through latching device which will hold fault signal until reset.
3. System will shut down again if failure repeats.
4. Telemetry may be used to command shutdown of system.
5. Feeder may be disconnected to shutdown system remotely if SSC has failed completely.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Probability of occurrence very low, because of extensive backup provisions. Secondary damage may occur if this type of failure does occur.

DISCUSSION

Lockout devices and sensors must be highly reliable and must be designed to fail in safe mode, if possible. Redundancy should be arranged so that either latched sensor signals or lockouts can all independently initiate shutdown and hold system locked out.

SHEET C11

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/21/75

SUBSYSTEM CONTROLS (LOSS OF SUPERVISORY)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Loss of supervisory link from master to remote units. (Inability to remotely command shutdown).

CRITICALITY

[ ] SAFETY [X] OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Telemetry system indicates power being generated after master station has commanded shutdown via telemetry link.
2. Same as (1) above, except indicated by despatcher.

POSSIBLE FAILURE CAUSES

1. Failure of Remote Station SSC equipment and associated WGS interface.
2. Failure of Master Station SSC equipment or Power supply.

COMPENSATING PROVISIONS

1. Personnel can be sent to station to shut down manually.
2. Feeder can be disconnected and generator protective gear will shut down WGS.
3. System may shut down automatically if reason for commanded shutdown was high winds (automatically shuts down if wind speed above limit).

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Although WGS system safety and protective functions are still operative; inability to command shutdown remotely could compromise system safety. Probably best to shut down system until supervisory link is restored.

DISCUSSION

SHEET C12

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/21/75

SYSTEM ROTOR CONTROLS

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Yaw system goes dead, no control motion of yaw servo. Yaw damping still functional.

CRITICALITY

[ ] SAFETY [X] OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Increased wind errors and vibratories when operating.
2. Once shut down, will not be able to untwist cable or orient head for start-up - (check as part of cable untwist sequence).
3. Check for wind error before start-up (check for reverse wind).
4. Redundant wind direction sensors do not agree (sensor failure).
5. Yaw servo and monitor do not agree (yaw servo failure).

POSSIBLE FAILURE CAUSES

1. Failure in SSC, no yaw control signal output.
2. Solenoid coils open or solenoid jammed in neutral.
3. No hydraulic pressure, burst line, auxiliary pump control failure
4. Loss of electric power to servo.
5. Failed wind sensor or open wire to same.
6. Comparator fails dead or open wiring.
7. Solenoid driver fails dead or open.
8. Demod fails dead or open wiring.

COMPENSATING PROVISIONS

1. Yaw servo operation continuously checked by monitor circuit.
2. SSC will not allow start unless wind is properly oriented.
3. Redundant wind direction sensors continuously monitored.
4. Redundant check of proper wind direction outside of SSC and shutdown if wind reverses.
5. Adequate margin on blade/tower clearance for reverse wind.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Some loss of operating time once shut down, since restart is not possible. Compensating provisions 2,3,4 prevent possibility of starting or operating with wind 180° off and resultant reverse thrust on rotor. Rotor includes "no back" device to prevent reverse rotation under these conditions in any case. Only potential hazard is possible blade/tower intersection if rotor operated with reverse wind (due to reverse thrust). Redundant compensating provisions above prevent this possibility.

SHEET C13

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/21/75

SUBSYSTEM ROTOR CONTROLS (YAW FREE)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Yaw system fails completely to free wheeling condition (no damping, no yaw restraint).

CRITICALITY

[ ] SAFETY [X] OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Increase in vibratiories at tower head if operating.
2. Large periodic motion of pintle at 2/revolution if operating.
3. Large random motions of tower head if shut down.
4. Cannot untwist cable or orient head for restart after shut down.
5. Check for wind error before start-up (check for reverse wind).
6. Yaw rates in excess of design values.

POSSIBLE FAILURE CAUSES

1. Shear of shaft or coupling from yaw motor to worm screw (not likely).
2. Structural failure in worm screw or gear (not likely).
3. Loss of fluid from yaw motor due to burst hydraulic line, seal failure, or ruptured housing.

COMPENSATING PROVISIONS

1. System designed to take associated vibration, but will cumulate same fatigue cycles. Vibration sensor should cause shutdown.
2. Loss of hydraulic fluid will also initiate shutdown and lockout.
3. See also compensating provisions for sheet (C12) (all apply here).
4. High yaw rate will be sensed and cause shutdown and lockout.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Bad vibration environment for equipment mounted on tower head and excessive wear of worm gear. Should shut down until repaired. No significant wear after shutdown.

DISCUSSION

The one additional item which is susceptible to damage after shutdown of system is the control cable, which does not pass through slip rings at the tower head, and which could be damaged if the free-wheeling tower head manages to cumulate too many turns. Also see Discussion of sheet C12, which applies here.

SHEET C14

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/20/75

SUBSYSTEM ROTOR CONTROLS (YAW LOCKED)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Yaw servo jams mechanically - no motion of tower head in yaw in spite of control inputs. (Mechanical equivalent of sheet C12).

CRITICALITY

☐ SAFETY ☒ OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Increase in vibration at tower head if operating as wind error builds up. (Jam does not allow any reorientation as wind shifts).
2. Cannot untwist cable or orient head for restart after shutdown.
3. Large wind error while operating with no corrective motion of yaw servo.
4. Null check indicates reverse wind (rotor upwind of tower).
5. Yaw servo monitor indicates failure.

POSSIBLE FAILURE CAUSES

1. Mechanical jam of yaw motor.
2. Hydraulic blockage prevents flow.
3. Tower head bearings jam.
4. Tower head worm gear set jams or worm screw bearings jam.

COMPENSATING PROVISIONS

1. System designed to take vibration, some fatigue cycling of rotor. Vibration sensor may initiate shutdown.
2. Continuous check of yaw servo by monitor circuit will detect failure.
3. SSC will not allow restart unless wind properly oriented.
4. Redundant check for proper wind direction independent of SSC before starting, and during operation.
5. Redundant wind direction sensors continuously monitored.
6. Power loss will initiate shutdown if wind shifts.
7. Adequate margin on blade/tower clearance for reverse wind.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Compensating provisions 3,4, and 5 prevent possibility of starting or operating with wind 180° off, and resultant reverse thrust on rotor. Loss of operating time till repaired. Cannot untwist cable or orient head for restart. See sheet C12 also (applies here).

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/22/75

SUBSYSTEM ROTOR CONTROLS (YAW HARDOVER)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)
Yaw system fails Hardover

CRITICALITY

☐ SAFETY ☒ OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

Same as Sheet C14, except for item 3 where yaw servo now moves in direction to increase wind error.

POSSIBLE FAILURE CAUSES

1. Wind sensor locks up (possible icing)
2. Synchro jams, shorted winding
3. Electrical hardover in servo loop.
4. Solenoid valve jams mechanically or electrically in hardover position.

COMPENSATING PROVISIONS

Same as Sheet C14

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Same as Sheet C12

DISCUSSION

Sheet C16

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/20/75

SUBSYSTEM CONTROLS (NO STARTUP)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Will not enter startup mode

CRITICALITY

[ ] SAFETY [X] OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. No KWHRS output with favorable wind conditions.
2. Failure indication via telemetry link (SSC system).

POSSIBLE FAILURE CAUSES

1. Failure indication from any of the sensors which cause shutdown of the system.
2. Failure of wind velocity and/or direction sensors.
3. Inoperative pitch and/or yaw servo and/or associated controls and hydraulics.
4. Inoperative auxiliary pump, motor, motor control, or loss of station service power.
5. Failure of SSC system (locks up in shutdown or monitor mode).

COMPENSATING PROVISIONS

No compensating provisions to allow startup in event of above failures.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Loss of operating time and generated output - No hazard.

DISCUSSION

Better to remain shut down if any of the above described failures have occurred than to allow startup with failed equipment.

SHEET C17

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/23/75

SUBSYSTEM ROTOR/CONTROLS (NO ACCEL)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Enters startup mode but rotor will not accelerate up to proper operating speed range with favorable winds.

CRITICALITY		EST FREQUENCY (MTBF - YEARS)
[ ] SAFETY	[X] OPERATIONAL	1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Failure to accelerate up to speed after entering startup mode.
2. No power output with favorable wind conditions.

POSSIBLE FAILURE CAUSES

1. Failure of Pitch servo or pitch control jam (see Sheet C1 thru C4 for details).
2. Failure of starting programmer or associated sensors.
3. Insufficient wind to complete acceleration of rotor (most likely cause).
4. Failure of SSC or associated interface with WGS (starting mode).
5. Failure of yaw servo prevents proper orientation (see Sheets C12 thru C15).

COMPENSATING PROVISIONS

1. SSC detects failure to accelerate due to low wind and initiates shutdown (no lockout).
2. Ability to remotely monitor power being produced indicates no output.
3. All compensating provisions of above sheets (C1 thru C4, and C12 thru C15 apply).
4. Low RPM limit check automatically enabled 5 minutes after startup, and will shut system down without lockout if wind is too low.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Loss of operating time. No hazard to system, unless rotor remains at resonant RPM for extended time causing some loss of fatigue life. This is redundantly protected against by compensating provisions (1) and (3) above.

SHEET C18

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/23/75

SUBSYSTEM CONTROLS (MISSED ALARM)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Missed Alarm - No failure indication at Master when failure has actually occurred.

CRITICALITY

☐ SAFETY ☒ OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 (20) 50 (>50) double failure mode

all modes

SYMPTOMS/METHOD OF DETECTION

1. Loss of expected energy output from system as reported by SSC.
2. Loss of expected energy output from system as reported by despatcher.
3. Periodic inspection of WGS reveals failures which were not reported.
4. Detect loss of link by automatic periodic telemetry check.

POSSIBLE FAILURE CAUSES

1. Failure of SSC master or remote stations.
2. Loss of interface components/wiring, etc. between WGS and SSC.

COMPENSATING PROVISIONS

1. WGS includes inherent backups and safeguards to take appropriate action in nearly all cases to safeguard system against secondary damage without external intervention.
2. SSC system includes feature to periodically confirm existence of proper link between remote and master and alarm if link is lost.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

No hazard to system, except for the remote chance that compensating provision (2) above does not function, and simultaneous failure occurs on WGS which requires immediate attention of an operator to avoid secondary damage to WGS.

DISCUSSION

Probability of this type of double failure is extremely remote.

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/23/75

SUBSYSTEM CONTROLS (DEGRADED TELEMETRY)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Degraded/improper performance data reported at master by telemetry link.

CRITICALITY — EST FREQUENCY (MTBF - YEARS)

☐ SAFETY ☒ OPERATIONAL — 1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Performance data does not make sense (obviously improper).
2. Performance data does not follow previously established trends.
3. Comparison of recorded and telemetered data shows inconsistencies.

POSSIBLE FAILURE CAUSES

1. Failure of SSC prevents proper telemetering of data.
2. Failure of sensing device or interface between sensor and WGS.

COMPENSATING PROVISIONS

1. Performance data also recorded at site and is therefore recoverable.

2. Previous experience can flag improper data.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

No effect on system unless reporting of improper data initiates supervisory shutdown of system, and results in lost operating time until repaired.

DISCUSSION

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 7/7/75

SUBSYSTEM ELECTRICAL - SSC

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Microprocessor fails to sequence or alarm properly.

CRITICALITY

[X] SAFETY [ ] OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Time spent in any given mode exceeds that normally allowed by up timeout sub routines internal to SSC.
2. Does not enter shutdown sequence (hangs up in operate, standby, or sync modes).
3. Hangs up in other modes (startup, monitor, or shutdown).
4. Transfers to wrong mode (e.g., inadvertent startup).
5. Oscillation between modes.

POSSIBLE FAILURE CAUSES

1. Failures internal to SSC electronics.
2. Output relay hangup for particular mode involved.

COMPENSATING PROVISIONS

1. Separate timers and logic circuits outside of SSC initiate shutdown if SSC fails to complete sequence in required time, or if SSC deviates from required sequence, or fails to alarm when sequence is not completed.
2. Separate interlocks for critical items outside of SSC where practical., e.g., interlock rotor brake with rotor RPM so it will not come on at high RPM.
3. Shutdown will be initiated external to SSC whether or not SSC enters proper sequence.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

No hazard to system so long as external logic detects failure and initiates shutdown. Shutdown can still be initiated even if SSC does not enter proper sequence. Loss of operating time until SSC is repaired.

DISCUSSION

NON CRITICAL FAILURES

SYSTEM WGS INITIALS & DATE DHB 7/4/75

SUBSYSTEM CONTROLS

FAILURE DESCRIPTION	METHOD OF DETECTION
Loss of Voice Link	Audible Silence
False Alarm at Master Station	Visual inspection at site
Failure of Reference Wind Sensor or comparator	Electrical failure alarm at master or visual self test during inspection
Failure of Pitch servo monitor	SAME AS ABOVE
Failure of Yaw servo monitor	SAME AS ABOVE
Failure of SSC monitor	SAME AS ABOVE
Failure of Reference power supplies	SAME AS ABOVE
Failure of one overspeed sensor (redundant sensors) or associated comparator	SAME AS ABOVE
Failure of torque monitor	SAME AS ABOVE

SHEET D1

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS/DRIVE INITIALS & DATE CW 6/20/75

SUBSYSTEM MAIN POWER PATH

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Excessive noise and vibration, loss of efficiency, tendency to run hot, possible loss of drive (shaft or coupling failure).

CRITICALITY

☐ SAFETY ☒ OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 50

SYMPTOMS/METHOD OF DETECTION

Vibration, overheating, loss of power, output for given wind conditions. Onset likely to be gradual. Detect by temperature readings or by periodic oscillograph traces of vibration behavior. Independent speed sensors at generator and rotor will detect catastrophic failures of shaft/couplings.

POSSIBLE FAILURE CAUSES

1. Material flaws in gears, bearings, or couplings.
2. Usually high shock loads.
3. Fretting due to small displacement high frequency motion while shut down.
4. Corrosion or wear due to contaminated lubricant.
5. Catastrophic failure of shaft or couplings.

COMPENSATING PROVISIONS

1. Conservative design with substantial overload capacity.
2. Large capacity fine filters in the lubrication system.
3. Well defined and facilitated maintenance program.
4. Inclusion of temperature, vibration, lubrication pressure and filter condition sensors to detect problems in early, non-serious stages.
5. Design so shaft and/or coupling failure will not interfere with operation of pitch controls and will cause immediate shutdown.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Failures requiring replacement of rotor bearings or main gearbox will require heavy lifting equipment and much time to repair. Drive couplings, quill shaft, rotor brake can be repaired fairly readily. System shutdown and lockout required when above type failures detected. Loss of operating time until repaired.

DISCUSSION

Gearbox should be given run prior to installation to disclose quality problems in material or assembly. Secondary systems such as gearbox lubrication and cooling systems should be continually monitored for proper function to preclude this malfunctioning leading to a major drive system problem.

SHEET E1

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/24/75

SUBSYSTEM ELEC./CONTROLS

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Loss of station service power (all loads or partial loss, emergency batteries still intact).

CRITICALITY

☐ SAFETY ☒ OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50
Cause #1 Causes 2,3,4

SYMPTOMS/METHOD OF DETECTION

1. No utilities (lights, heat, service power, etc.).
2. Alarm from 24 VDC emergency charger (no-charge alarm).
3. Alarm from main 110/220 supply (station service) combined with (2) above.

POSSIBLE FAILURE CAUSES

1. Transmission line down or damaged due to weather, lighting, etc.
2. Station service transformer failure.
3. Shorts or blown breakers in station service distribution system within WGS.
4. Failures in 24 VDC charger.

COMPENSATING PROVISIONS

1. 24 VDC emergency batteries for all critical loads (see schematics).
2. Rotor can still be shut down even with no battery power (pure mechanical backup).
3. Alarms (symptom (2) above) cause immediate shutdown and lockout.
4. Alarm (symptoms (2) and (3) simultaneously) causes shutdown after time delay. (1 minute).

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

No effect on system except for possible loss of operating time. Time delay on provision (4) above allows for possibility of temporary outage and prevents unnecessary shutdowns. System holds in standby mode during time delay interval (standby interrupt sent to SSC).

DISCUSSION

Note that if line to WGS is down, there is no way to generate power since link with utility is broken. If no power available from either battery or station serv., rotor can be shut down, but cannot be properly parked. However, this is a double failure situation. (Extremely remote). Note that overspeed protection and emergency shutdown devices still operate in event of double failure (e.g., both station service and emergency power failure).

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE AK 6/23/75

SUBSYSTEM ELECTRICAL - LIGHTNING ARRESTOR

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

No lightning current protection.

CRITICALITY EST FREQUENCY (MTBF - YEARS)

☒ SAFETY ☐ OPERATIONAL 1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Mechanically damaged Arrestor.
2. Carbon Tracks on insulation.
3. Blown vent cap on Arrestor.
4. Bad grounding connectors/wiring.

POSSIBLE FAILURE CAUSES

1. Too strong lightning stroke
2. Mechanical damage

COMPENSATING PROVISIONS

Other arrestors in vicinity

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Reduction in lightning protection.

DISCUSSION

Arrestors and wiring will require periodic visual inspection, especially after severe winds or thunderstorms.

SHEET E3

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE WAS 6/20/75

SUBSYSTEM ELECTRICAL

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

System will not synchronize with grid. (No pull-in of breaker).

CRITICALITY

☐ SAFETY ☒ OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Low generator voltage.
2. Loss of field power (sync. Gen.)
3. One or more open phases
4. Synchronizer and verifier disagree.
5. Main breaker will not close.
6. Speed error meter will not come within 2% of network for induction generator.
7. Fault indication from supervisory and sequencing control (SSC) monitor.

POSSIBLE FAILURE CAUSES

1. Failed voltage regulator, field breaker, or field power transformer (sync. gen.).
2. Failed generator.
3. Defective synchronizer or verifier (sync. gen.).
4. Defective speed matching interlock (ind. gen.).
5. Defective pitch control.
6. Open enable signal line to breaker or failed breaker.
7. Failed SSC.
8. See sheet C17 also.
9. Insufficient wind to maintain RPM.

COMPENSATING PROVISIONS

1. SSC monitor will interrupt startup sequence and initiate shutdown if a fault is indicated, or if synch process is not complete in 20 minutes.
2. Overtemperature detectors and ground fault relay detects generator failures.
3. Synch verifier (synch generator)
4. See Sheet C17.
5. Insufficient wind should trip low RPM limit monitor after 5 minutes, and should not lock out.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

When system is shut down and locked out, there is no further danger of damage. However, operating time will be lost until repaired.

DISCUSSION

It is considered extremely remote that any serious failure in the electrical system would escape detection and cause serious damage during the twenty minute period allowed by the SSC for the main breaker to close.

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE AK 6/23/75

SUBSYSTEM ELECTRICAL

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Generator mechanical failure causing loss of efficiency and difficulty in system startup.

CRITICALITY EST FREQUENCY (MTBF - YEARS)

☐ SAFETY ☒ OPERATIONAL 1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Excessive vibration and noise.
2. Reduced output.
3. Bearing overtemperature.

POSSIBLE FAILURE CAUSES

1. Generator bearing failure.
2. Loose mounting.
3. Bent or broken shaft.
4. Armature failure.
5. Frame failure.
6. Corrosion.

COMPENSATING PROVISIONS

1. Bearing temperature relays.
2. Mechanical failure may induce electrical failure which will be detected by provisions of sheet E-5.
3. Vibration pickups in tower head may initiate shutdown.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

No further damage if detected and tripped off the line. Damage generally limited to generator if not detected. Probability of this type failure very low if generator is properly maintained.

DISCUSSION

SHEET E5

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE AK 6/23/75

SUBSYSTEM ELECTRICAL - POWER GENERATING APPARATUS

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Excessive heating due to electrical fault inside WGS protective loop.

CRITICALITY EST FREQUENCY (MTBF - YEARS)

☐ SAFETY ☒ OPERATIONAL 1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

Overheating of affected component and/or abnormal current flow through ground fault or diff. relay.

POSSIBLE FAILURE CAUSES

1. Insulation failure of wiring or other components due to mechanical damage.
2. Loss of power transformer coolant.
3. Generator electrical failure.
4. Failed voltage regulator and/or field circuits.

COMPENSATING PROVISIONS

1. Differential current relay.
2. Ground fault relay.
3. Over voltage relay.
4. Overtemperature devices in generator and transformer.
5. Standard utility protective practices used throughout system.
6. Loss of field relay and fuses on field power transformer and PTs.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Trip off - lock out - no further damage. Redundancy in protective devices precludes possibility of failure going undetected.

DISCUSSION

Standard utility protective practices used throughout electrical system.

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE AK 6/23/75

SUBSYSTEM ELECTRICAL

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Excessive power reversal (watts or vars)

CRITICALITY

☐ SAFETY ☒ OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. KWH Meter countdown.
2. Trip of reverse power and/or excitation loss relay (sync. gen.)
3. Trip of pitch control system failure sensors (see Sheets C1 thru C4)
4. Trip of yaw control system failure sensors (see Sheets C12 thru C15).

POSSIBLE FAILURE CAUSES

1. Loss of field or field excitation, regulator/exciter failure (sync. generator only).
2. Failure of pitch control system (sheets C1 through C4).
3. Failure of yaw control system (sheets C14 and C15)

COMPENSATING PROVISIONS

1. Reverse power relay and excitation loss relays will trip.
2. Winding overtemperature relays.
3. Pitch and yaw control monitoring sensors should prevent power reversal (See above referenced sheets).

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Trip and lockout, highly redundant sensing provisions should prevent secondary damage.

DISCUSSION

Standard Utility protective practices used throughout electrical system.

SHEET E7

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/24/75

SUBSYSTEM ELECTRICAL/CONTROLS

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Loss of emergency battery power (24 VDC) to critical loads. (Station service power still intact).

CRITICALITY

[ ] SAFETY [X] OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Alarm from battery breaker (auxiliary contacts)
2. Low battery voltage alarm.
3. Failure indication from associated loads, e.g., pitch controls, servos, etc.
4. Advance warning via No-charge alarms on battery chargers.

POSSIBLE FAILURE CAUSES

1. Failure of batteries due to improper maintenance, overload, age, mechanical.
2. Open battery connectors and/or cables.
3. Inadvertent trip of battery protective breakers.
4. Failure of emergency distribution lines or tripped breakers in lines to critical loads.

COMPENSATING PROVISIONS

1. Loads supplied by chargers until shutdown completed (Causes 1,2, and 3).
2. Rotor can still be shut down even with no battery power (pure mechanical) and overspeed protection is still operative.
3. Critical loads such as breaker trip lines can be made redundant from battery supply to trip coil.
4. All battery alarms cause immediate shutdown and lockout.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

No hazard to system with above compensating provisions except loss of operating time until repaired. Most significant might be inability to trip main breaker. Recommend redundant trip lines direct from independent power sources (one from battery and one from station service).

DISCUSSION

Loss of power from both batteries and station service means rotor can still be shut down, but cannot be parked. This is double failure situation (extremely remote). No hazard to rotor. Backup breaker at Utility Company end of WGS feeder line would be desirable if it can be provided.

NON CRITICAL FAILURES

SYSTEM WGS INITIALS & DATE AK 6/23/75

SUBSYSTEM ELECTRICAL

FAILURE DESCRIPTION	METHOD OF DETECTION
Shorted pf capacitor or Capacitor switch.	Blown fuse, low line voltage due to vars being drawn from network.
Worn or pitted slip ring.	rf noise in communication or signal system, ozone odor, during slew.
Failed switchboard indicators.	Periodic inspection checks of instruments.
Failed electrical housekeeping equipment.	Visual inspection - space heaters should be checked monthly.

SHEET E9

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 6/30/75

SUBSYSTEM ELECTRICAL (SYNCHRONIZER/VERIFIER)

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Main synchronizer allows closure of breaker when generator is not in sync with network (Verifier prevents closure for synchronous).

CRITICALITY

[X] SAFETY [X] OPERATIONAL

SYNCHRONOUS INDUCTION

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Periodic inspection of proper operation of both synchronizer and verifier (both include self check provisions) for synchronous.
2. Overcurrent and overtorque trip either machine off line if inadvertent closure does occur.
3. Backup speed matching interlock provided for induction machine through primary controls (generator tachometer).

POSSIBLE FAILURE CAUSES

Failure in main synchronizer (synchronous generator).

Failure of speed matching device (induction generator).

COMPENSATING PROVISIONS

1. Verifier prevents inadvertent closure to line (Backs up synchronizer).
2. Periodic inspection using self check features on both units assures proper operation. (Not likely both units would fail between inspection intervals).
3. Synchroscope available as further backup for periodic checks.
4. In event of double failure, generator protective relays will trip.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

None given above compensating provisions. Mechanical and/or electrical damage to system could occur if synchronous generator was engaged out of phase. Induction generator not critical. Would be tripped off by overcurrent if engaged with large speed error before damage could occur.

DISCUSSION

Failure of verifier between checkout intervals would be backed up by main synchronizer. Double failure required to allow inadvertent closure for either machine.

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 7/5/75

SUBSYSTEM ELECTRICAL - UTILITY

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Electrical failures outside of WGS on utility company equipment.

CRITICALITY

[ ] SAFETY [X] OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

Standard generator protective relaying equipment for external faults.

POSSIBLE FAILURE CAUSES

1. Phase-to-ground faults or phase-to-phase faults.
2. Three phase faults.
3. One or two lines open (partial loss of load).
4. All three lines separated from grid - WGS may be isolated with block of feeder load.

COMPENSATING PROVISIONS

1. Phase unbalance and redundant overcurrent relays detect cause (1) above.
2. Redundant overcurrent relays detect cause (2) above.
   NOTE: Overcurrent relays are voltage restrained to allow more sensitive settings.
3. Overvoltage on open lines and phase unbalance detects (3) above.
4. Under and over frequency detects cause 4 above for synchronous, not critical for induction.
5. Utility senses presence of WGS on line, and prevents out-of-phase reclosure.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Loss of operation until fault is cleared. Automatic reclosure by utility after a fault should only occur if WGS line is dead. Under and over frequency relays at WGS provide backup protection for synchronous machine. Induction not critical for reclosure, but needs overvoltage protection to prevent resonance if separated from network.

DISCUSSION

Utility recommends WGS should always wait at least one minute before coming back on line if tripped due to external fault, to allow time for reclosers to finish cycling. All generator relaying equipment for WGS follows standard utility practices and will not be analyzed in detail here.

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB 7/7/75

SUBSYSTEM ELECTRICAL - UTILITY

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Feeder recloser recloses on WGS in out of sync condition.

CRITICALITY EST FREQUENCY (MTBF - YEARS)

☐ SAFETY ☒ OPERATIONAL 1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Frequency of WGS should drift off 60 Hz (No speed regulation in operate mode). Under or over frequency relays will trip WGS breaker.
2. Utility recloser will see excited line if WGS breaker fails to trip.
3. Possible overcurrent and/or unbalance will trip WGS breaker when recloser trips off initially.

POSSIBLE FAILURE CAUSES

WGS may fail to trip off line when separated from power grid if WGS is isolated with a block of load which roughly matches its power output.

COMPENSATING PROVISIONS

1. Under and over frequency relays should sense frequency drift and open WGS breaker.
2. Reclosers on Utility feeder to WGS should look for voltage on line before reclosing.
3. Overcurrent and unbalance relays may also trip WGS breaker.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

No hazard to system unless there is double failure - redundant compensating provisions.

DISCUSSION

Utility should provide voltage sensing on reclosers looking toward WGS.

SHEET R1

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM ROTOR INITIALS & DATE HG 6/20/75

SUBSYSTEM ROTOR BLADE AND HUB

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

CRITICALITY

[x] SAFETY [ ] OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

Vibration sensors responsive to frequencies from 1/rev. to 10/rev. (0.5 - 5 Hz).

POSSIBLE FAILURE CAUSES

1. Out of track due to pitch control wear.
2. Loss of blade stiffness due to fatigue camage or high wind overload.
3. Unbalance due to F.O.D.
4. Unbalance due to water entry.

COMPENSATING PROVISIONS

Automatic shutdown at pre-set vibration level.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

System will be shut down until corrective action is taken.

DISCUSSION

All significant modes, if failure associated with the rotor, will produce 1/rev. vibrations. Accordingly, suitable vibration pickups should be mounted in these axes at the outboard rotor shaft bearing support, (non rotating), as close to the hub as possible. Furthermore, a means of measuring pitch control loads in the rotating system (via instrumented pitch limbs) is recommended for the development rotor. In production rotors, pitch control loads may be monitored in the non-rotating system. Rotor shutdown should be

<u>DISCUSSION</u> (continued)

initiated automatically when control loads exceed a specified value, (possibly caused by jams, etc.).

Monitoring of vibration levels and control loads can also detect failures in other parts of the system as a side benefit.

SHEET R2

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE HG 7/1/75

SUBSYSTEM ROTOR

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Rotor overspeed.

CRITICALITY

[x] SAFETY [ ] OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

Overspeed exceeding 50%

SYMPTOMS/METHOD OF DETECTION

1. RPM increase over limit.
2. Sudden generator load reduction at maximum torque.

POSSIBLE FAILURE CAUSES

1. Torque overload limit, rotor control malfunction.
2. Sudden sustained high wind gust.

COMPENSATING PROVISIONS

1. Primary pitch control system capability to prevent overspeed due to gusts and generator load loss.
2. Backup mechanical pitch control overspeed correction independent of electrical and hydraulic servo failures.
3. Rotor blade design to take overspeed (50% recommended).

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

1. Permanent rotor shutdown until malfunction is corrected.
2. Temporary shutdown until wind abates.

DISCUSSION

SHEET R3

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE HG/WAS 7/1/75

SUBSYSTEM ROTOR

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Catastrophic Loss of Blade/Blade-Tower Intersection.

CRITICALITY

[X] SAFETY [ ] OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

1. Extreme vibration indicated by monitoring system before failure.
2. Loss of all power output, telemetering control and data reporting.

POSSIBLE FAILURE CAUSES

1. Foreign object collision with rotor.
2. Collapse of tower.
3. Blade fatigue failure.
4. Violent wind gust in reverse direction.
5. Severe meteorological condition such as tornadoes.

COMPENSATING PROVISIONS

1. Conservative rotor/tower design to minimize probability of catastrophic failure.
2. Incipient failure warning devices to detect failure onset before a catastrophic failure results.
3. Adequate visibility and minimum terrain masking to minimize probability of aircraft collision.
4. Should consider blade tip lights or WGS illumination at night for collision avoidance.
5. Regular weather monitoring to avoid WGS operation under severe or violent wind condition.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

System would be destroyed or would sustain major structural damage.

DISCUSSION

WGS system design will be sufficiently conservative to make this type or failure extremely improbable. This includes design features in rotor and control system to initiate system shutdown and lockout before a catastrophic failure occurs.

SHEET R4

NON CRITICAL FAILURES

SYSTEM ROTOR SYSTEM INITIALS & DATE HG 6/23/75

SUBSYSTEM HUB AND BLADES

FAILURE DESCRIPTION	METHOD OF DETECTION
Fatigue cracks on blade and hub delamination	Visual, dye check, ultrasonic inspections
Corrosion	
Oil leaks	
Control bearings looseness	
Erosion of LE	
F.O.D. (minor)	
Lightning damage	
Lightning anode erosion	

SHEET T1

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE RGP/ DHB 6/16/75

SUBSYSTEM TRUSS TOWER

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Excessive vibration of tower head while operating WGS under moderate load conditions.

CRITICALITY

[X] SAFETY [ ] OPERATIONAL

EST FREQUENCY (MTBF - YEARS)

1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

Lateral and longitudinal vibration sensors located at appropriate points on tower rotor head to initiate automatic shutdown in event of excessive vibration.

POSSIBLE FAILURE CAUSES

1. Loss of most of diagonals or horizontal braces in one or more bays due to excessive static loads.
2. Same as (1) above, but due to successive fatigue failures.
3. Loss of one leg or main member due to excessive static load.
4. Loss of one leg or main member due to fatigue failures.

COMPENSATING PROVISIONS

Redundancy in structure for moderate loads.
Vibration sensing and automatic shutdown.
Periodic inspection for early detection of fatigue failures.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

Loss of availability until repaired.

DISCUSSION

System still susceptible to more extensive damage even after shutdown if failure is such that tower could not withstand high static wind loading. This type of failure should be repaired ASAP if high winds are expected.

SHEET T2

FAILURE MODES AND EFFECTS ANALYSIS

SYSTEM WGS INITIALS & DATE DHB/RGP 6/16/75

SUBSYSTEM TRUSS TOWER

EFFECT ON SUBSYSTEM OPERATION (UNCOMPENSATED)

Tower collapses while operating WGS. (Also see sheet R3).

CRITICALITY

EST FREQUENCY (MTBF - YEARS)

☒ SAFETY ☐ OPERATIONAL 1 2 5 10 20 50 >50

SYMPTOMS/METHOD OF DETECTION

Loss of all power output and telemetering control and data reporting. Loud noise followed by many telephone calls.

POSSIBLE FAILURE CAUSES

1. Static failure of critical structure under higher than design loads.
2. Static failure of critical structure due to mechanical damage.
3. Fatigue failure of critical structure undetected by inspection.

COMPENSATING PROVISIONS

Conservative design and periodic inspection to minimize probability of this type failure.

EFFECT ON SYSTEM (BASED ON ABOVE PROVISIONS)

System would be destroyed.

DISCUSSION

Design will be sufficiently conservative to make this type failure extremely unlikely. This includes design features in rotor and control system to minimize the probability of a blade/tower intersection; or loss of a blade which could cause excessive tower loads and possible tower failure.

NON CRITICAL FAILURES

SYSTEM WGS INITIALS & DATE TC 6/16/75

SUBSYSTEM TRUSS TOWER

FAILURE DESCRIPTION	METHOD OF DETECTION
Fatigue cracks	Visual Inspection
Loose, missing rivets, bolts	Visual Inspection
Broken weldments	Visual Inspection
Surface deterioration/corrosion	Visual Inspection
Excessively worn stair treads	Visual Inspection
Cracked, broken concrete	Visual Inspection
Excessive settling of foundation	Visual Inspection

APPENDIX D

SYMBOLS

APPENDIX D
SYMBOLS

SECTION 3 - SYSTEM ANALYSIS

A_{ROT}	Rotor Disc Area, m^2 (ft^2)
Dia	Diameter, m (ft)
E_{AER}	Aerodynamic Efficiency of the Rotor
h	Height Above Ground, m (ft)
h_{ref}	Reference Ground Height, 9.1m (30 feet)
r/R	Spanwise Blade Station, Fraction of Total Radius
P	Power, kw
P_R	Rated Power (Output), kw
R	Rotor Radius, m (ft)
RPM	Rotor Speed in Revolutions Per Minute
V	Wind Velocity, m/s (fps)
V_{WND}	Wind Speed at Reference Altitude, m/s (mph)
V_{REF}	Wind Speed at Reference Altitude, m/s (fps)
V_R	Rated Wind Speed (Quoted at Reference Altitude), m/s (mph)
$\overline{V}$	Median Wind Speed (Quoted at Reference Altitude), m/s (mph)
V_D	Design Wind Speed for Rotor Optimization, m/s (mph)
V_{EFF}	Effective (Integrated) Wind Speed as Seen by the Rotor Disc, m/s (fps)
V_{TIP}	Rotor Tip Speed, m/s (fps)

SYMBOLS (Continued)

σ	Rotor Solidity
ρ	Air Density, kg/m^3 (slug/ft^3)
$\theta_{.75}$	Blade Pitch Angle at .75 Spanwise Blade Station, Deg
ψ	Blade Azimuth Position, radians
Ω	Rotor Angular Velocity, radians/sec

SECTION 4 - ROTOR SUBSYSTEM

a	Lift Curve Slope
a	Virtual Lag Hinge Offset, m (ft)
a.c.	Aerodynamic Center
b	Distance From Virtual Lag Hinge Offset to Blade c.g., m (ft)
c	Blade Chord Length, m (ft)
C_{D_o}	Blade Profile Drag Coefficient
C_L	Lift Coefficient
c.g.	Center of Gravity
C(k)	Theodorsen Lift Deficiency Function
EI	Blade Bending Stiffness, kg-cm^2 (lb-in^2)
GJ	Blade Torsional Stiffness, kg-cm^2 (lb-in^2)
I_b	Blade Moment of Inertia About its Virtual Lag Hinge
$I_{\beta\beta}$	Blade Flapping Inertia, kg-m^2 (slug-ft^2)
$I_{\beta\theta}$	Blade Product of Inertia, kg-m^2 (slug-ft^2)

SYMBOLS (Continued)

$I_{\theta\theta}$	Blade Feathering Inertia, kg-m^2 (slug-ft^2)
m_b	Blade Mass, kg (slugs)
M	Total Effective Mass of Blades and Tower, kg (slugs)
n	Number of Blades
N	Rotor Speed, RPM
r	Blade Radial Station, cm (in)
R	Rotor Radius, m (ft)
V_{ci}	Minimum Wind Speed at Which the Rotor Will Achieve Operating RPM, m/s (mph)
V_{cutout}	Maximum Operating Wind Speed, Beyond Which the Rotor Would be Shut Down, m/s (mph)
V_{rated}	Rated Wind Speed m/s (mph)
V_{tip}	Rotor Tip Speed, m/s (ft/sec)
V_{wind}	Wind Speed, m/s (mph)
X_A	Distance From the Blade Section Feathering Axis to the Section Aerodynamic Center, Positive When the Aerodynamic Center is Forward, Percent Chord
X_I	Distance From the Blade Section Feathering Axis to the Section Center of Gravity, Positive When the Center of Gravity is Forward, Percent Chord
α	Blade Angle of Attack, Degrees
β	Blade Flapping Angle, Degrees
γ	Lock Number = $\frac{\rho a c R^4}{I_{\beta\beta}}$
θ	Blade Pitch Angle, Degrees

SYMBOLS (Continued)

λ	Inflow Ratio = $\frac{V_{wind}}{\Omega R}$, Negative for WGS
μ	Forward Flight Velocity Ratio for Helicopters, $\mu \approx 0$ for WGS
ρ	Air Density, kg/m^3 ($slugs/ft^3$)
σ	Rotor Solidity Ratio = $\frac{ncR}{\pi R^2}$, dimensionless
ω	Tower Frequency Associated With Mass M, rad/sec
ω_o	Static Frequency of Blade About its Virtual Lag Hinge, rad/sec
ω_θ	Blade Natural Feathering Frequency, rad/sec
Ω	Rotor Speed, rad/sec

Superscripts $\dot{x}$ and $\ddot{x}$ denote differentiation with respect to time, $\frac{d}{dt}$, $\frac{d^2}{dt^2}$

SECTION 5 - CONTROLS SUBSYSTEM

m	Exponent Which Relates Blade Pitch Angle to Wind Speed Above Rated - Dimensionless
σ	Rotor Solidity - Dimensionless

SECTION 6 - STRUCTURAL SUBSYSTEM

A	Projected Area, m^2 (ft^2)
C_D	Drag Coefficient of a Body
EI	Bending Stiffness, $N \cdot m^2$ ($kip \cdot ft^2$)

SYMBOLS (Continued)

F_{ty}	Yield Tensile Stress, MPa (ksi)
F_{tu}	Ultimate Tensile Stress, GPa (ksi)
F'_c	30 Day Ultimate Strength of Concrete, MPa (ksi)
F_x	Horizontal Force in Direction of Shaft Axis, kN (kips)
F_y	Horizontal Force Normal to Shaft Axis, kN (kips)
F_z	Vertical Force, kN (kips)
JG	Torsion Stiffness, N•m^2 (k•ft^2)
L	Effective Length of Member, m (ft)
M_x	Moment About the x Axis, Essentially Rotor Torque, kN•M (ft-kips)
M_y	Moment About the y Axis, Tower Overturn Moment, kN•M (ft-kips)
M_z	Moment About the z Axis, Tower Torsion, kN•M (ft-kips)
ρ	Mass Density of Air, kg/m^3 (slug/ft^3)
ρ	Radius of Gyration, m (ft)
V	Wind Velocity, m/sec (ft/sec)

APPENDIX E

PROTOTYPE WIND GENERATOR SYSTEM COSTS

PROTOTYPE WIND GENERATOR SYSTEM COSTS

INTRODUCTION

The purpose of this Appendix is to examine and derive realistic costs for the first prototype WGS systems of the size range developed in the Design Study (NAS3-19404). The Design Study addressed future production costs for such machines. Subsequent evaluations carried out by Kaman during the proposal phase for an actual first unit of a similar WGS (Mod-1 proposal), and in connection with other studies noted below, give an indication of the probable cost of the first pre-production systems. Although these cost levels appear excessively high when compared with the low cost of future machines projected in the Design Study, there is, in fact, no real conflict.

The costs of a present day prototype system are higher than those of a full production first unit due to a number of factors which are discussed and evaluated herein. The cost influence of each of these factors is estimated.

The several factors to be examined in this study include:

- Inflation
- Size changes
- Contractor capability and subcontracting
- Competitive procurement environment
- Production design improvements
- R & D non-recurring activities
- Learning curves

The effects of these factors on system costs are used to bring the Mod-1 Study cost estimate for the 1000th production unit (180 ft diameter system in constant 1975 dollars and an established and growing market) to the expected cost of a near term, prototype first unit Mod-1 system (200 ft diameter in 1977 dollars). The expected end result prototype system costs are not affected by the order in which these effects are estimated and applied; for illustration purposes, the effects are applied in the order presented. Figure E-1 summarizes these costs at the several levels considered. In addition, Tables E-1 and E-2 are given showing subsystem costs for both the Mod-1 1500 kW system and for a 500 kW system.

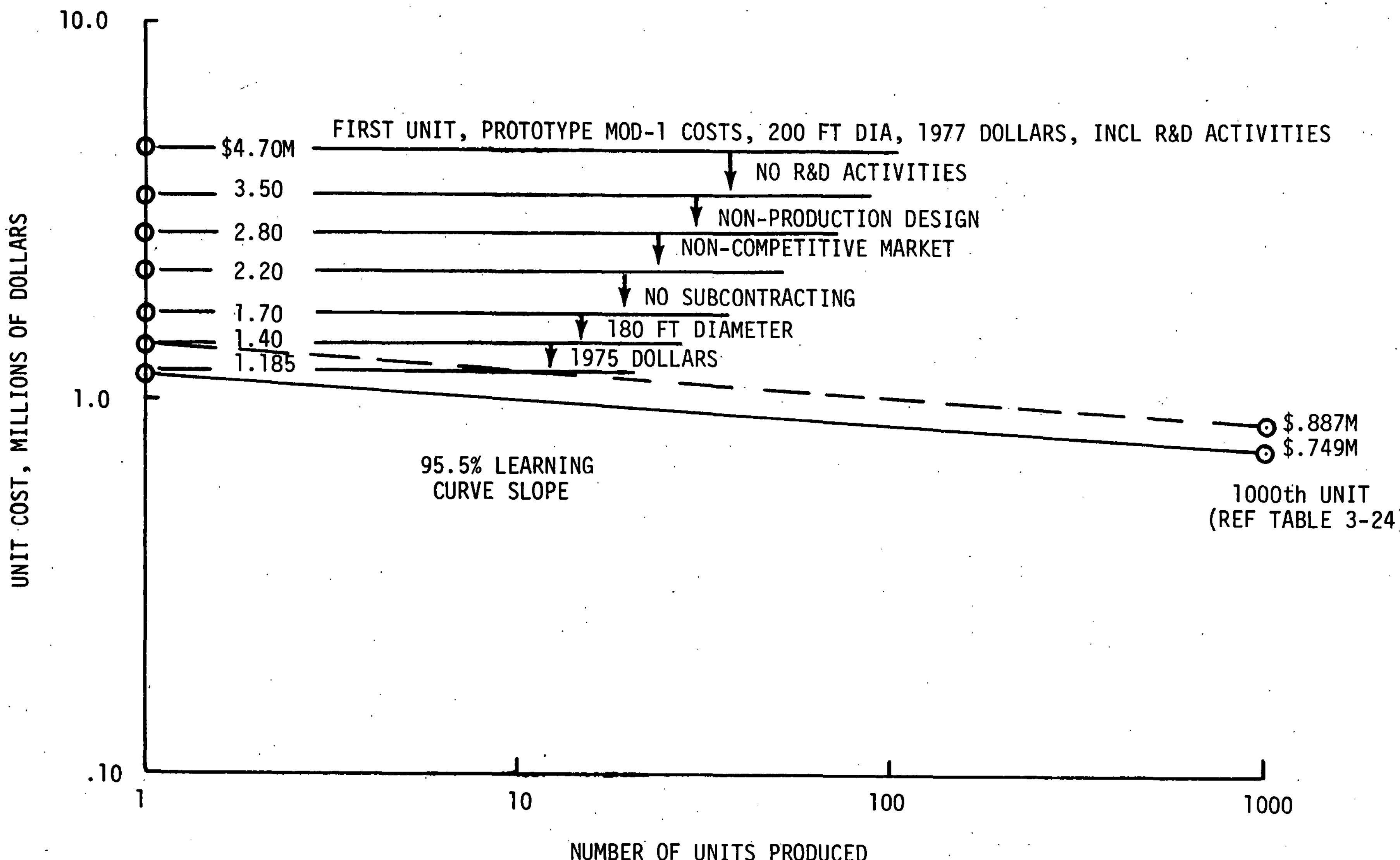


Figure E-1. Estimated 1500 kW Mod-1 System Costs.

		PRIME CONTRACTOR				SUBCONTRACTORS					PRIME CONTRACTOR BURDEN (5)	TOTALS
		LABOR (1)		PURCHASED PARTS (2)	OTHER (3)	LABOR (1)		PURCHASED PARTS (2)	OTHER (3)	BURDEN (4)		
		(HOURS)	($1000)	($1000)	($1000)	(HOURS)	($1000)	($1000)	($1000)	($1000)	($1000)	($1000)
PROGRAM MANAGEMENT & SYSTEMS INTEGRATION		1000	22	---	6	---	---	---	--	---	10	38
EQUIPMENT	ROTOR	9000	174	124	16	3000	64	83	7	39	190	697
	NACELLE	---	---	---	--	5000	100	335	--	95	198	728
	ELECTRICAL	---	---	---	--	600	13	145	--	30	70	258
	CONTROLS	---	---	---	--	5000	99	50	--	39	70	258
	STRUCTURE	---	---	---	--	6000	120	86	--	43	93	342
ASSEMBLY & INSTALLATION	NACELLE	---	---	---	--	3000	60	12	--	20	34	126
	TOWER & FOUNDATION	---	---	---	--	29000	582	54	13	126	290	1065
	ERECTION	---	---	---	--	7000	144	49	--	39	87	319
	CHECKOUT	---	---	---	--	20000	414	7	53	98	213	785
SITE COSTS	LAND PREPARATION	---	---	---	--	2000	39	---	--	5	16	60
	CONTROL BUILDING	---	---	---	--	500	10	4	--	2	6	22
TOTALS		---	196	124	22	---	1645	825	73	536	1277	4698

NOTES:

(1) Labor costs based on typical aerospace industry composite rates of $9.00 per hour and 125% overhead

(2) Purchased parts include materials and catalog & off-the-shelf components, as purchased

(3) Other costs include computer time, publications, travel, etc., as appropriate

(4) Subcontractor burden rates based on typical component and subassembly industry burdens and profits

(5) Prime contractor burden rates based on typical aerospace industry rates of 25% General and Administrative and 10% profit

TABLE E-1. ESTIMATED 1500 kW SYSTEM PROTOTYPE COSTS, (THOUSANDS OF DOLLARS)

		PRIME CONTRACTOR				SUBCONTRACTORS					PRIME CONTRACTOR BURDEN (5)	TOTALS
		LABOR (1)		PURCHASED PARTS (2)	OTHER (3)	LABOR (1)		PURCHASED PARTS (2)	OTHER (3)	BURDEN (4)		
		(HOURS)	($1000)	($1000)	($1000)	(HOURS)	($1000)	($1000)	($1000)	($1000)	($1000)	($1000)
PROGRAM MANAGEMENT & SYSTEMS INTEGRATION		1000	22	--	6	---	---	---	--	---	10	38
EQUIPMENT	ROTOR	7000	140	85	10	2000	39	47	3	22	129	475
	NACELLE	---	---	--	--	4000	83	211	--	65	134	493
	ELECTRICAL	---	---	--	--	400	9	97	--	21	47	174
	CONTROLS	---	---	--	--	3000	67	33	--	26	47	173
	STRUCTURE	---	---	--	--	4000	82	58	--	29	63	232
ASSEMBLY & INSTALLATION	NACELLE	---	---	--	--	2000	41	8	--	13	23	85
	TOWER & FOUNDATION	---	---	--	--	20000	395	37	8	86	196	722
	ERECTION	---	---	--	--	4000	81	50	--	27	59	217
	CHECKOUT	---	---	--	--	14000	282	5	36	67	145	535
SITE COSTS	LAND PREPARATION	---	---	--	--	1000	22	---	--	3	9	34
	CONTROL BUILDING	---	---	--	--	500	10	4	--	2	6	22
TOTALS		---	162	85	16	---	1111	550	47	361	868	3200

NOTES:

(1) Labor costs based on typical aerospace industry composite rates of $9.00 per hour and 125% overhead

(2) Purchased parts include materials and catalog & off-the-shelf components, as purchased

(3) Other costs include computer time, publications, travel, etc., as appropriate

(4) Subcontractor burden rates based on typical component and subassembly industry burdens and profits

(5) Prime contractor burden rates based on typical aerospace industry rates of 25% General and Administrative and 10% profit

TABLE E-2. ESTIMATED 500 kW SYSTEM PROTOTYPE COSTS, (THOUSANDS OF DOLLARS)

DISCUSSION

Table 3-24 of the Design Study Final Report presented estimated WGS costs, including a subsystem breakout, for the 1000th unit (500 kW and 1500 kW systems). The cost level for this 1000th unit was projected by the parametric model in the Design Study final report as approximately $749,000. The cost/performance model that generated this cost level utilized a 95.5% unit learning curve to reflect costs at different production points. On the basis of the production price given above and the learning curve slope, the cost of the first production unit would back-figure to approximately $1,185,000. In actual fact, however, the cost of early engineering prototype units may be expected to be considerably higher than this figure.

Based on work done by Kaman subsequent to the Design Study, specifically the Mod-1 proposal effort, the WECS Off-Shore Study for ERDA, and other proprietary Kaman work, it is possible to estimate the cost of a representative initial prototype unit which reflects the realistic factors for market situation, development stage, and procurement scenario. Accordingly, these factors will be examined in the following paragraphs as they influence the $1,185,000 first-unit production cost figure given above. Note that for reference purposes, the configuration which will be utilized is the 1500 kW, 200 ft rotor diameter system which has been established by NASA as the Mod-1 system.

Inflation. The $1,185,000 first production unit cost, derived in the subject study, is based on 1975 dollars. Using an average inflation rate of 7% per year, this is equivalent to a $1,400,000 first-unit price in the mid-1977 mid-point of the actual Mod-1 development program. Inflation also brings the 1000th unit cost up to $887,000.

Size Changes. Cost sensitivity analyses indicate that total system costs in this size category will increase approximately $15,000 per foot of rotor diameter, increasing the first-unit price to approximately $1,700,000 for the 200 ft diameter final Mod-1 configuration noted above.

Contractor Capability and Subcontracting. The Mod-1 Study cost analysis assumed a wind turbine manufacturer with a breadth of capability to fabricate and erect a complete system without major subcontracting. For near term prototype wind turbines, however, such a corporate structure does not yet exist and some subcontracting will be necessary, the actual amount depending on the specific prime-subcontractor arrangement made.

The typical procurement arrangement used in this analysis assumes that a specialized manufacturer carries out the rotor system development and fabrication and another contractor is responsible for the remainder of the system (nacelle, drive train, tower and site preparation). It is also assumed that the rotor manufacturer is the prime contractor for the total WGS system who subcontracts the remainder of the system.

For this program, the additional burdening effect resulting from the above procurement arrangement is estimated to be $500,000, raising the expected first-unit production cost to approximately $2,200,000.

Competitive Procurement Environment. At the present time, a competitive market in wind turbine components still does not exist. Requests for vendor quotes and cost estimates are complied with on only a routine basis. Given a future market where manufacturers of generators, gearboxes, towers, etc., are aware of the potential for future contracts, lower and more competitive subsystem prices are likely. However, during the more limited market environment situation which exists at present, it is expected that costs will remain higher by an estimated 25%, resulting in approximately $2,800,000 for the first prototype in a non-competitive market.

Production Design Improvements. The first prototype system cost in a non-competitive market is also based on the assumption that a production design has been evolved. At the present time, such a production design does not truly exist and the first prototype unit design will not have had the advantage of experience that will only come with operating time. It is believed that this present lack of experience leads to conservative design loads and heavier and more costly components.

The prototype status of the Mod-1 system also results in the adoption of components which are available, but not necessarily ideally sized or configured for full production use. The gearbox, generator, bearings, electrical system components, etc., are selected for prototype use based on their availability and general suitability for such use.

Additionally, again because of the prototype aspects of the Mod-1 system, some of the components designed and/or selected will not be ideally suited, from a produceability standpoint, for full production use. Given a production status with sufficient units to amortize non-recurring costs, it is strongly believed that more cost effective components could be utilized.

It is believed that all of the non-production aspects of the prototype system just discussed would add an additional 25% to the costs, leading to an estimated pre-production, first-unit prototype cost of $3,500,000. This represents the cost of a wind turbine designed and built to today's state-of-the-art and in today's competitive market. It is a 200 ft diameter, 1500 kW system. The cost is expressed in 1977 dollars, and covers recurring costs primarily, with a level of program management commensurate with the fabrication and erection of a pre-production prototype.

R & D Non-recurring Activities. The above cost, however, does not yet reflect the special costs for a design and development program as procured by a Government agency, such as is the case for Mod-1. Non-recurring program elements such as the presence of the Engineering Data System, the incorporation of test instrumentation and planning, developmental testing, full technical and financial reporting and the R & D program management costs for the above work, are expected to add an estimated $1,200,000 to the Mod-1 program costs. It may be expected, then, that a Mod-1-sized, Government-procured system would cost approximately $4,700,000.

Learning Curves. The learning that will occur as the production progresses to hundreds of units is difficult to estimate with precision. Actual learning will occur only on those components such as blades and other rotor elements which are of new design and where production is just beginning. On the other hand, such items as gearboxes and generators are now in production and little "learning" is likely to occur. Additionally, vendors are likely to pass on the cost reductions resulting from learning only when competition from other suppliers forces them to. Ordering quantities, tooling concepts and line breaks all affect the learning to be expected. Analysis of component statuses and pricing concepts for components and subsystems during the Mod-1 Study derived a conservative 95.5% composite learning curve for the system. There is no evidence to date leading to a change in that conclusion.

CONCLUSION/SUMMARY

The first-unit 1500 kW prototype cost has been estimated at $4,700,000, of which $3,500,000 represents the cost of a pre-production first-unit. Figure E-1 summarizes the foregoing steps leading to this value. Table E-1 shows a breakout of estimated costs by major subsystem and by prime and major subcontractor. The costs include the prime contractor's burden of subcontractor work. Typical component industry average hourly, overhead, general, and administrative and profit rates are used throughout.

Table E-2 shows a similar breakout of estimated costs for the 500 kW prototype first-unit system, including R & D non-recurring costs.

In summary, this cost analysis has discussed the differences between prototype development program costs and full production first unit costs and has estimated the various cost elements therein. A breakout of prototype Mod-1 1500 kw and of 500 kW system estimated costs has been presented, utilizing typical industry rates and fees. The learning that may be expected as production proceeds and the conditions that affect learning have also been discussed; the 95.5% learning curve of the Design Study has been retained as the most likely slope for learning projection which is foreseen at this time.

REFERENCES

2-1. Thomas, R., Puthof, R. and Savino, J., PLANS AND STATUS OF THE NASA-LEWIS RESEARCH CENTER WIND ENERGY PROJECT, NASA TMS-71701, September 1975.

3-1. Metzger, R. F., Plaks, A., Meier, R. C., Berman, A., DEVELOPMENT OF A METHOD FOR THE ANALYSIS OF IMPROVED HELICOPTER DESIGN CRITERIA, USAAMRDL TR 74-30, July 1974.

3-2. Plaks, A., Metzger, R. F., MacDonald, H. I., Meier, R. C., IMPROVED COST EFFECTIVENESS OF HELICOPTERS THROUGH A TWO-POINT DESIGN CRITERION, American Helicopter Society Preprint No. 811, Presented at the Annual National Forum of the AHS, May 1974.

3-3. Abbott, I. A., Von Doenhoff, A. E., SUMMARY OF AIRFOIL DATA, NACA Report No. 824, 1945.

3-4. Wenzinger, C. J., Harris, T. A., WIND TUNNEL INVESTIGATION OF THE NACA 23012, 23021 and 23030 AIRFOILS WITH VARIOUS SIZES OF SPLIT FLAPS, NACA Report No. 668, 1939.

3-5. Goett, H. J., Bullivant, W. K., TESTS OF NACA 0009, 0012 AND 0018 AIRFOILS IN THE FULL SCALE TUNNEL, NACA Report No. 647, 1938.

3-6. Bullivant, H. J., TESTS OF THE NACA 0025 and 0035 AIRFOILS IN THE FULL SCALE TUNNEL, NACA Report No. 708, 1941.

4-1. Rosenbrock, H. H., VIBRATION AND STABILITY PROBLEMS IN LARGE WIND TURBINES HAVING HINGED BLADES, TR C/T113, The British Electrical and Allied Industries Research Association, 1955.

4-2. Coleman, Robert P., Feingold, Arnold, M., THEORY OF SELF-EXCITED MECHANICAL OSCILLATIONS OF HELICOPTER ROTORS WITH HINGED BLADES, NACA TN 3844, February 1957.

4-3. Miller, R. H., Ellis, C. W., BLADE VIBRATION AND FLUTTER, American Helicopter Society Journal, July 1956.

5-1. Sholes, J., RECOMMENDED ASSUMPTIONS FOR ECONOMIC MODEL AND WIND GUST MODEL, Letter from NASA Lewis to Kaman Aerospace Corporation, March 11, 1975.

1. Report No. NASA CR-134937	2. Government Accession No.	3. Recipient's Catalog No.

4. Title and Subtitle	5. Report Date
DESIGN STUDY OF WIND TURBINES, 50 kW TO 3000 kW, FOR ELECTRIC UTILITY APPLICATIONS - ANALYSIS AND DESIGN	February 1976
	6. Performing Organization Code
7. Author(s)	8. Performing Organization Report No. R-1382
	10. Work Unit No.
9. Performing Organization Name and Address Kaman Aerospace Corporation Bloomfield, Connecticut 06002	11. Contract or Grant No. NAS 3-19404
	13. Type of Report and Period Covered Contractor Report
12. Sponsoring Agency Name and Address U.S. Department of Energy Division of Solar Energy Washington, D.C. 20545	14. Sponsoring Agency ~~Code~~ Report no. DOE/NASA/9404-76/2

15. Supplementary Notes

Final Report. Prepared under Interagency Agreement E(49-26)-1010. Project Manager, John E. Sholes, Wind Energy Project Office, NASA Lewis Research Center, Cleveland, Ohio 44135

16. Abstract This report presents the results of a program to develop preliminary designs of low power (50-500 kW) and high power (500-3000 kW) wind generator systems (WGS) for electric utility applications. These designs provide the bases for detail design, fabrication, and experimental demonstration testing of these units at selected utility sites. The program include four tasks: a conceptual design task; an optimization task; a preliminary design task; and a utility requirements evaluation task. In the conceptual design task, several feasible WGS configurations were evaluated, and the concept offering the lowest energy cost potential and minimum technical risk for utility applications was selected. In the optimization task, the selected concept was optimized utilizing a parametric computer program prepared for this purpose. In the preliminary design task, the optimized selected concept was designed and analyzed in detail. The utility requirements evaluation task examined the economic, operational, and institutional factors affecting the WGS in a utility environment, and provided additional guidance for the preliminary design effort. Results of the conceptual design task indicated that a rotor operating at constant speed, driving an AC generator through a gear transmission is the most cost-effective WGS configuration. The optimization task results led to the selection of a 500 kW rating for the low power WGS and a 1500 kW rating for the high power WGS. It was also determined that these two machine designs could be installed at utility sites with yearly median wind speeds from 8 to 20 mph, and provide energy at costs which approach those of machines optimized for each specific site. The preliminary design task produced a detailed refinement of the optimized selected concept, which utilizes a rotor with two variable pitch, filament wound composite blades, mounted on a rigid hub, driving a standard AC synchronous generator through a commercial gearbox. The system designs were prepared for both a conventional steel truss tower, and a precast, post-tensioned concrete shell tower. The utility requirements analyses indicate that conventional electric utilities can operate and maintain WGS units with no substantial change in normal operating and maintenance procedures.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Wind energy conversion systems Wind turbine technology	Unclassified - unlimited STAR Category 44 DOE Category UC-60

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	535	

* For sale by the National Technical Information Service, Springfield, Virginia 22161

☆ U.S. GOVERNMENT PRINTING OFFICE: 1978— 757-139/6330