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DEVELOPMENT OF AN 8 KILOWATT WIND TURBINE GENERATOR FOR RESIDENTIAL TYPE APPLICATIONS

Phase I. Design and Analysis. Volume II—Technical Report

By
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L. G. Angeloff
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Work Performed Under Contract No. AC04-76DP03533

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Grumman Energy Systems, Inc.
Bohemia, New York

U.S. Department of Energy

Solar Energy

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Phase I

Design and Analysis
VOLUME II - TECHNICAL REPORT

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Prepared For
Rockwell International Corporation
Energy Systems Group
Rocky Flats Plant
Wind Systems Program
P. O. Box 464
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Project Abstract

This report is an account of the work performed by Grumman Energy Systems in accordance with the Phase I (Design Evaluation) Work Statement of Contract No. PF71787-F, "Development of an 8 kW Wind Turbine Generator (WTG)", issued on 3 January 1978. This represents an element of an overall program entitled "Technical and Management Support for the Development of Wind Systems for Farm and Rural Use", managed by the Wind Systems Program, Rocky Flats Plant, Energy Systems Group of Rockwell International on behalf of the Department of Energy. The goal of this program is to develop Wind Energy Conversion Systems (SWECS) with potential for rapid commercialization, and which will be economically viable in rural and residential areas. The objective of the 8 kW WTG program is to develop and provide a practical prototype capable of being manufactured and sold at prices competitive with alternative energy sources. The task has been a challenging one, and the detailed trade-off studies performed in Phase I have resulted in a configuration which differs somewhat from the original concept. Grumman is confident, however, that the final design at the completion of the contract has the capability of meeting the overall program goals.

Phase I has consisted of detailed design evaluation and trade-off studies, which are necessary to finalize the SWECS configuration so as to meet the following general design and performance specifications.

Design and Performance Specifications

Power Output	8 kW (minimum) at a wind speed of 9 m/s (20 mph) at sea level
Cut-In Wind Speed (of generator)	Minimize with regard to power production and system cost
Cut-Out Wind Speed	Maximize with regard to power production and system cost
Survival Wind Speed (maximum gust velocity)	74 m/s (165 mph)
Rotor Speed Control	Optional Design
Yaw Control	Optional Design
System Life Goal	25 years (minimum)
Target Capital Cost Goal (for mass produced unit consisting of rotor, gear train, controls and tower)	$750/1977 dollars (maximum)

This design period has lasted 13 months (3 January 1978 - 31 January 1979) and will lead into a 5 to 7 month fabrication and testing (Phase II) period. The contract is currently funded through delivery and installation at the buyer's facility (at Rocky Flats).

VOLUME II

CONTENTS

VOLUME II

CONTENTS (Cont)

VOLUME II

CONTENTS (Cont)

FIGURES

TABLES

TABLES (Cont)

NOMENCLATURE

AAWS	Annual Average Wind Speed
ac	Alternating Current
AkWh	Annual Energy in Kilowatt hours
C	Count of Hall Effect Device used in measuring blade angle
CDR	Critical Design Review
COE	Cost of Energy
C_p	Power Coefficient
C_T	Thrust Coefficient
D	Rotor Diameter
dc	Direct Current
DOE	Department of Energy
ECU	Electronic Control Unit (as used on Windstream 25)
FCR	Fixed Cost Ratio
FDR	Final Design Review
f_n	Natural Frequency
F_x	Force in the x direction (lateral)
F_y	Force in the y direction (fore-aft)
F_z	Force in the z direction (vertical)
GESI	Grumman Energy Systems, Inc.
Hz	Hertz (cycles per second)
H_z	Hours of exceedance of wind speed V annually
IC	Initial Cost
kW	Kilowatt
kWh	Kilowatt hour
ksi	1000 lb. per square inch
LEROM-UV	Light Erasable Read Only Memory - Ultra Violet
MCU	Main Control Unit (as used on 8 kW WTG)
MS	Margin of Safety
M_x	Moment about lateral axis
M_y	Moment about fore-aft axis
M_z	Moment about vertical axis
N	Number of revolutions per minute
NASTRAN	NASA Structural Analysis
P	Power
PDR	Preliminary Design Review
PH	Precipitation Hardened
PMG	Permanent Magnet Generator
RFP	Request for Proposal
rpm	Revolutions per minute
ScF	Stress Concentration Factor
SF	Safety Factor
SOW	Statement of Work
SWECS	Small Wind Energy Conversion Systems
V	Wind Velocity
$\overline{V}$	Average Mean Wind Speed
V_{REF}	Wind Speed at which a rotor develops maximum power coefficient
V_z	Annual Average Wind Speed at elevation z

1.0 INTRODUCTION

The Department of Energy (DOE) initiated a program for the development and study of Small Wind Energy Conversion Systems (SWECS) in 1977. As a part of this program, Request For Proposals (RFP) were issued for design and development of an 8 kW Wind Turbine Generator (WTG) in a 20 mph wind. In January, 1978, Grumman Energy Systems Incorporated (GESI) was awarded a contract to proceed with Phase I of the 8 kW WTG design.

The objectives of this development, as shown in Section 1 of the Statement of Work (SOW) were:

1. To develop a technology capable of designing, building, and selling (at a cost competitive with alternative energy costs) WTGs in this size range for use in a number of rural and residential applications

2. To provide fabrication cost data which may be utilized in determining the economic viability of WTGs in the 8kW size range

3. To demonstrate wind energy systems of the 8 kW size range which are technically practical.

During the three years preceding this design, GESI had obtained considerable information and experience on its Windstream 25 WTG. This machine showed a good match with the requirements of the SOW for the 8 kW WTG. It was decided that the Windstream 25 (WS-25) would form a good starting point for the 8 kW WTG design. However, as a result of several studies conducted during the design phase, the following improvements were made:

- A dual actuator control system replaced the single motor with tip speed brake system of the WS-25

- The hard-wired Electronic Control Unit of the WS-25 was replaced by the micro-processor Main Control Unit (MCU) of the 8 kW WTG, thereby introducing flexibility to the control of parameters

- The MCU was positioned at the base of the tower or in a control shed, giving ease of access, as compared to the ECU, which was positioned inside the nacelle

- The blade structure was improved by using 6061-T6 instead of 6063-T6, giving greater strength

- Rotor diameter was increased from 25 ft. to 33½ ft., giving an 80% increase in wind collecting area

- Rotor blades were coned to allow a centrifugal relief of bending loads to the blades

- A dedicated design was made for the 8 kW WTG hub, as compared to the adapted DC3 hub design used on the WS-25

- Material selection for the 8 kW WTG and material finishes included the requirements of a 25 year service life

- Detailed attention was paid to fatigue loads

- An Induction Generator replaced the Alternator system of the WS-25

- Detail design of the vertical shaft allows an easier installation in the tower

- Weight balance of the rotor and nacelle about the vertical shaft and support by a roller thrust bearing allows greater freedom in yaw

- Improved accessibility to equipment in nacelle was achieved for ease of maintenance

- The Pitch Control Actuator was placed in the nacelle, thus eliminating the need for drive shaft slip rings.

The results of the Phase I work are detailed in this report, giving the history and results of the technical and cost studies conducted.

2.0 CONFIGURATION OVERVIEW

Table I presents the Final Design Review design features. The design predictions meet or exceed the design requirements in every area but cost per kW.

Table II shows the evolution of the design from that initially proposed to the current configuration approved at FDR, while Figs. 1 through 3 present the inboard profile associated with the various designs.

Further cost reductions can be realized through additional refinement of the design, and through the utilization of less expensive manufacturing techniques that can be adopted in volume production. However, Grumman believes that the final cost reduction attempts should be made after the results of the test program are analyzed.

Table I 8 kW Prototype Design Features

POWER OUTPUT	10-11 kW @ 20 MPH WIND SPEED
SURVIVAL WIND SPEED	165 MPH WITH A SAFETY FACTOR OF 2 (ON YIELD)
SYSTEM LIFE	25 YEAR DESIGN FATIGUE LIFE WITH A TIMES 4 FACTOR
PREDICTED COST PER kW	\$ 990 – \$ 1,050
ON LINE AVAILABILITY	HIGH DUE TO: SIMPLICITY REDUNDANCY
POWER GENERATION	
3ϕ AC	INDUCTION GENERATOR (PROTOTYPE SYSTEM)
1ϕ AC*	INDUCTION GENERATOR
DC*	ALTERNATOR – RECTIFIER/BATTERY SYSTEM
TEMPERATURE EXTREMES (-50°C TO +60°C)	ACTUATOR HEATER STRIP FOR COLD ENVIRONMENTS CONTINUOUS DUTY MOTORS (IN ACTUATORS) SYNTHETIC HYDROCARBON LUBRICANTS LOW TEMPERATURE GREASES FOR COLD ENVIRONMENTS GEAR BOX FAN FAN COOLED INDUCTION GENERATOR RAM AIR VENTILATION
*DEVELOPMENTAL OPTIONS	
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Table II Design Evolution

DESIGN EVENT	PROPOSED DESIGN 1/3/78	PDR-1 5/8/78	PDR-2 8/8/78	CDR 10/31/78 & 2/1/79
ROTOR TYPE/DIA	DOWNWIND/28 FT (8.5 M)	DOWNWIND/32 FT (9.8 M)	DOWNWIND/33.5 FT (10.1 M)	DOWNWIND/33.5 FT (10.1 M)
ROTOR CONING ANG	0°	0°	7°	3½°
BLADE TYPE/CHORD	AL EXT/24 IN. (61 CM)	AL EXT/24 IN. (61 CM)	AL EXT/18 IN.	AL EXT/18 IN. (45.7 CM)
TOWER/GUY WIRE	55 FT CONC/NO (16.8 M)	55 FT CONC/NO (16.8 M)	55 FT COR-TEN/YES (16.8 M)	55 FT COR-TEN/YES (16.8 M)
POWER GENERATION	ALTERNATOR	ALTERNATOR	IND GEN	IND GEN
SYSTEM OUTPUT RATED kW/MAX kW	8/NOT DET.	11.1/12	11.1/20	10-11/20
UTILITY INT	SYNC INVERTER	SYNC INVERTER	DIRECT	DIRECT
WEIGHT	NOT DEF.	3,200 LBS	2,540 LBS	2,589 LBS
COST/kW AT 1,000 UNITS/YR	$750	$1082	$1020	$ 990 – $1,050

NOTE: AL EXT = ALUMINUM EXTRUSION; CONC = CONCRETE; IND. GEN = INDUCTION GENERATOR; NOT DET = NOT DETERMINED; INT = INTERFACE; NOT DEF = NOT DEFINED

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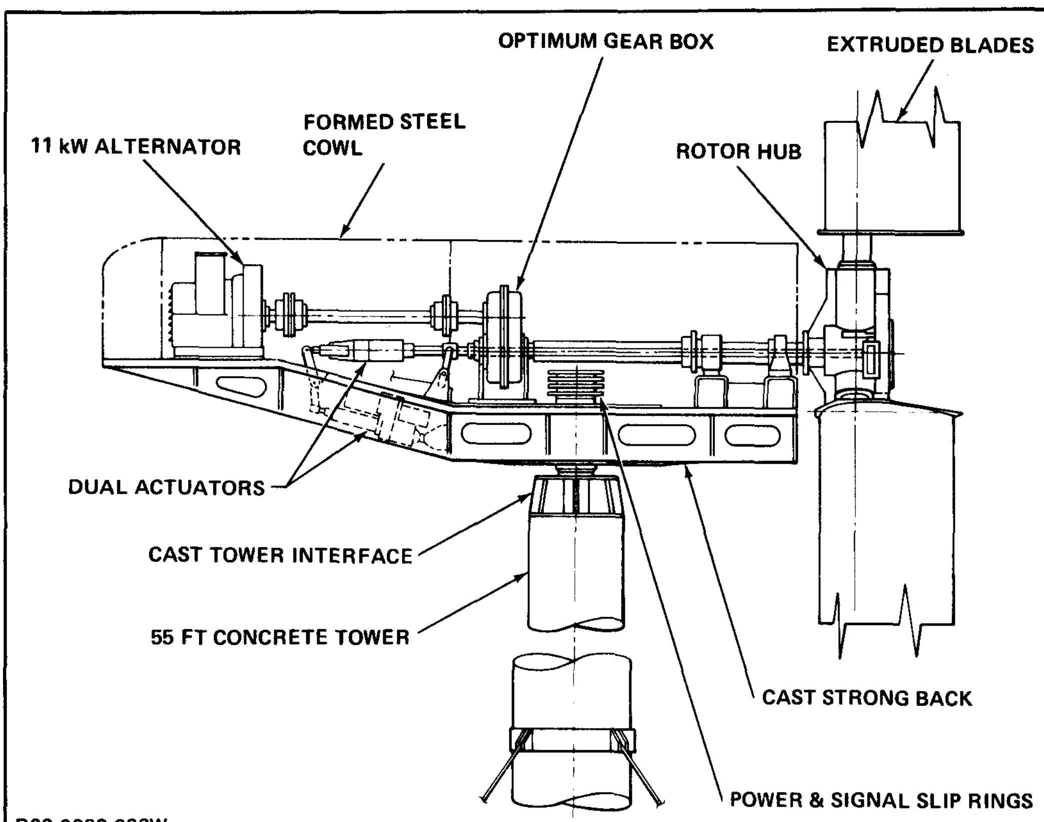

Fig. 1 Preliminary Design Review I Production System

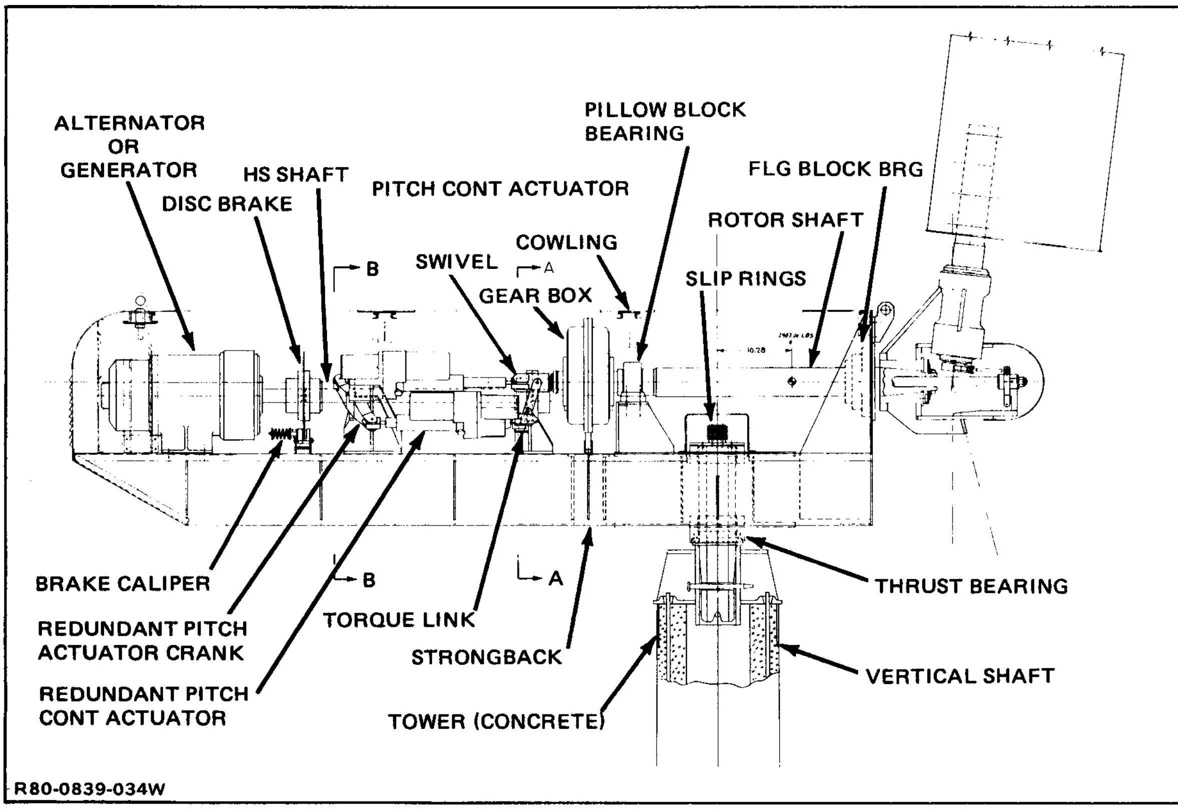

Fig. 2 Preliminary Design Review II Prototype System

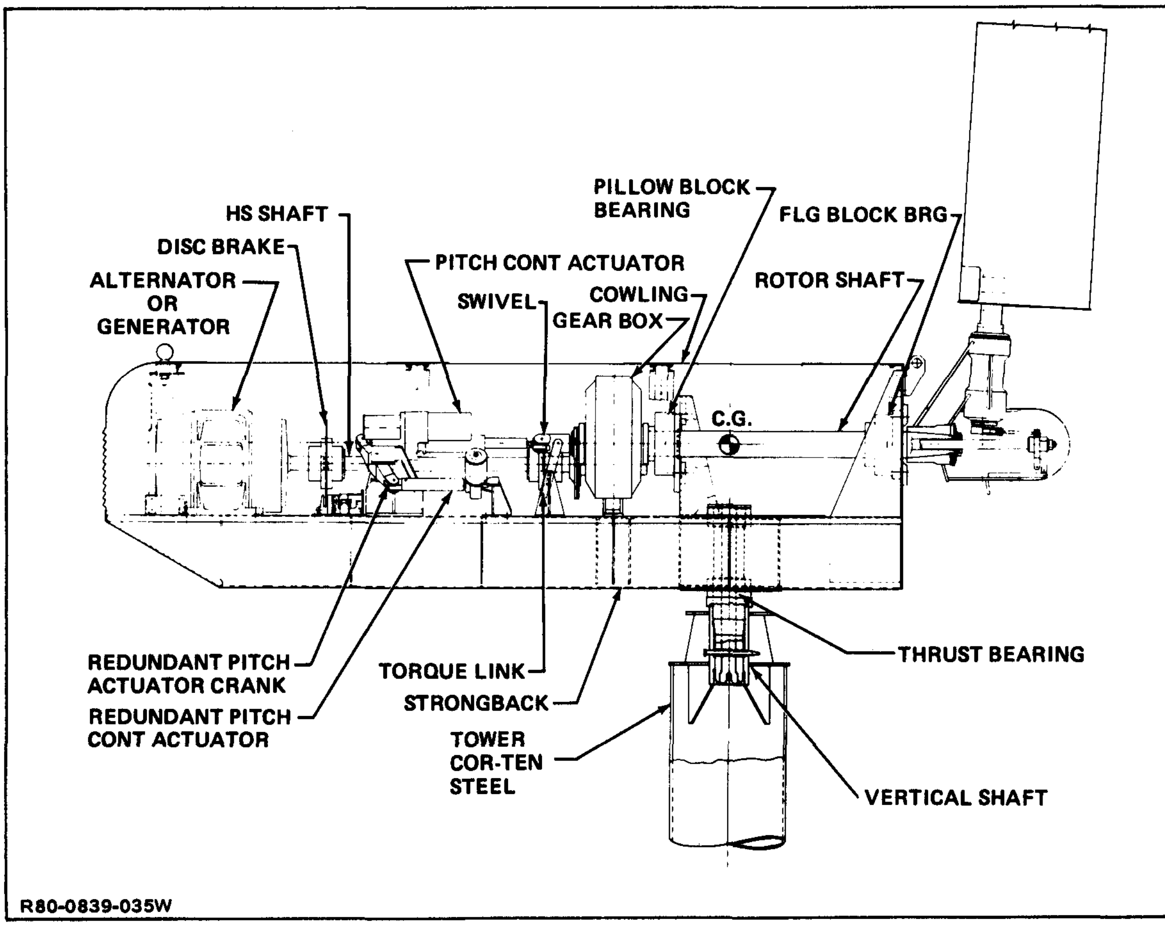

Fig. 3 Critical and Final Design Review Prototype System

3.0 COST ANALYSIS

Two methods are available in assessing the cost effectiveness of a wind generation system, and are based on the cost of power produced and the cost of energy produced.

The first method used to determine cost effectiveness of the 8 kW WTG used the cost of power approach, setting a target of \$750/kW (1977 dollars). A target of 8 kW in a 20 mph wind was set which the Grumman WTG exceeds when using a V_{REF} = 16.63 mph (10.8 kW in a 20 mph wind; see Fig. 4). From Table III the cost of the unit is \$11,010 in 1978 dollars, yielding a cost of 11,010/10.8 = \$1,019/kW (1978). The target cost of \$750/kW becomes \$802/kW with 7% inflation, and thus the unit exceeds the target cost by 27%. If the WTG had been designed with a gearbox which optimized the 20 mph wind speed performance (V_{REF} = 20 mph), then its output would have been 11.4 kW, and would have exceeded the target cost by 20%.

The cost of energy (COE) approach taken uses the expression COE = $\{(IC \times FCR) + AOM\}/AkWh$. The installed cost (IC) is the sale price (see Table IV) of \$11,010 x 1.5 + \$3,800 = \$20,315 (1978). The 1.5 factor is the burden for transportation, dealer, fees, etc., and the \$3,800 is the installation cost. A fixed charge rate (FCR) of 8.7% is used, and \$281 per year is allowed for annual operating maintenance (AOM). The annual energy production in kWh (AkWh) is obtained from Fig. 5, where its variation with annual average wind speed is shown. This yields the costs in Table IV, which are plotted in Fig. 6.

Grumman feels that comprehensive testing of the prototype WTG will lead to further design improvements aimed towards lowering the production costs.

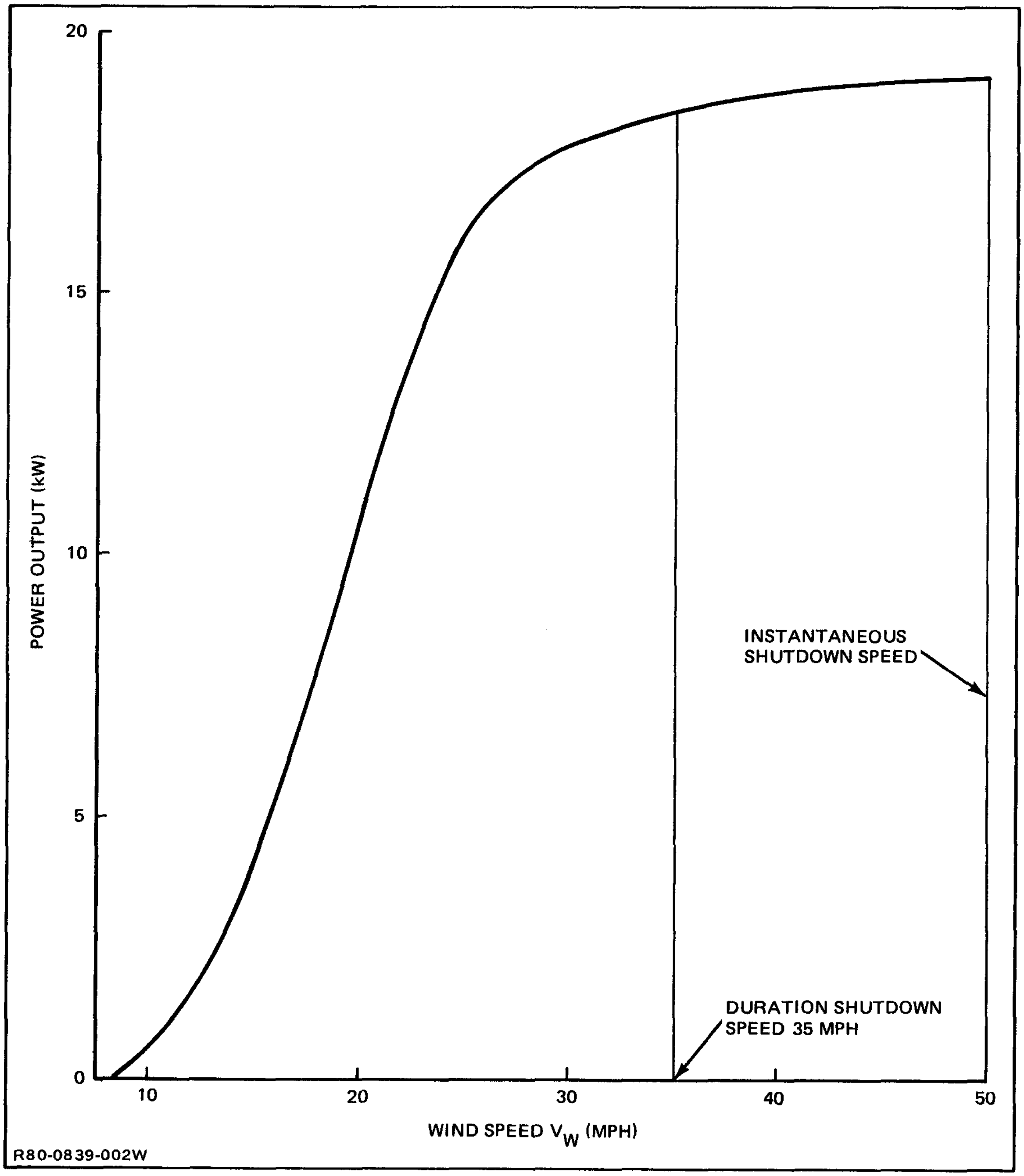

Fig. 4 Induction Generator Output vs Wind Speed (V_{REF} = 16.63 MPH)

Table III Production Cost Summary (1978 Dollars)

ASSEMBLIES COST APPROACH				
BASE LINE EQUIPMENT	PURCHASED PARTS	RAW MATERIAL AND/OR CASTINGS	FAB & ASSY LABOR	SUB TOTALS
1 LOW SPEED DRIVE	1063	242	174	1479
2 HIGH SPEED DRIVE	288	–	6	289
3 ROTOR HUB ASSY	355	339	530	1224
4 BLADES	884	12	98	994
5 VERTICAL SHAFT/PEDESTAL	36	154	186	376
6 STRONG BACK	–	137	400	537
7 COWLING ASSY	–	86	143	229
8 PITCH CONTROL MECH	8	58	120	186
9 ELECTRICAL EQUIP	1601	–	136	1737
10 MISC HARDWARE	50	–	50	100
11 PAINT	–	50	60	110
12 FINAL ASSY	–	–	184	184
13 TOWER	–	880	379	1259
	4280	1958	2466	8704
G & A & PROFIT (26.5%)				2306
SELLING PRICE				11,010
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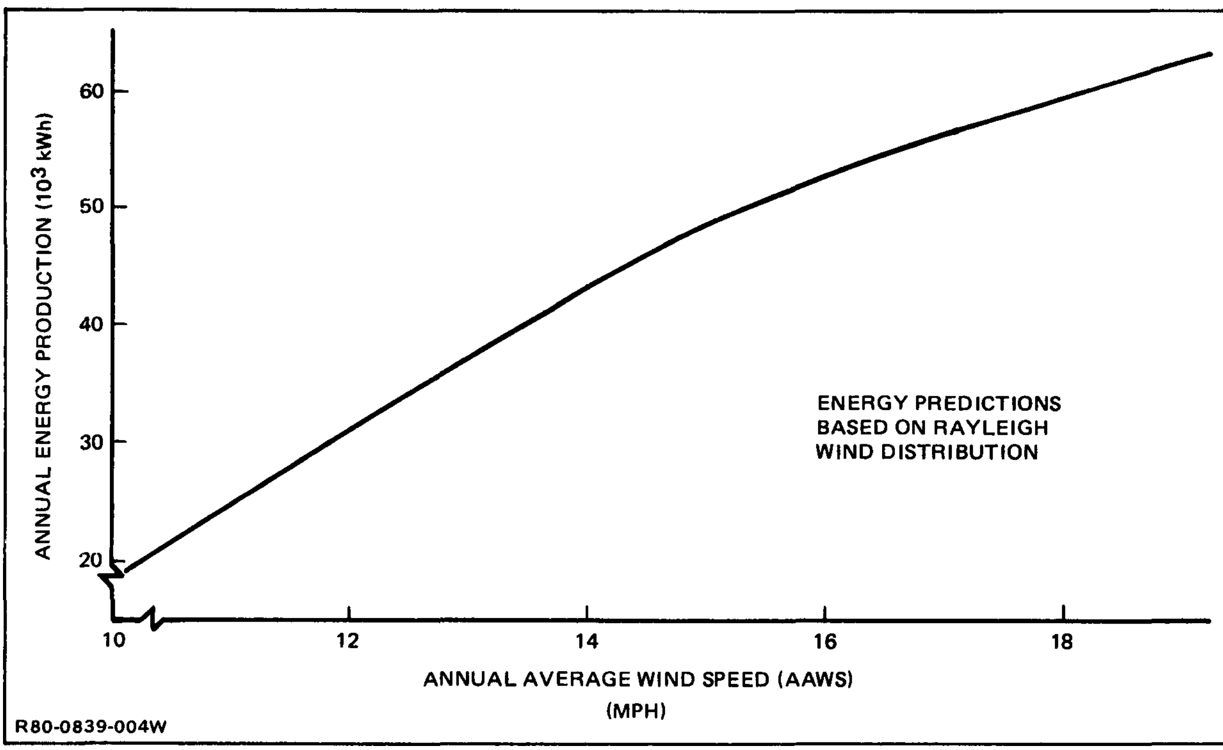

Fig. 5 Annual Energy Production vs AAWS (V_{REF} – 6.63 MPH)

Table IV Cost of Energy vs. Annual Average Wind Speed

AAWS (MPH)	ANNUAL ENERGY kWh	COST OF ENERGY (1978) PER kWh
10	18,500	11.1 ¢
12	31,500	6.5 ¢
14	43,600	4.7 ¢
16	53,000	3.9 ¢
18	59,500	3.4 ¢

THESE PREDICTIONS ARE COMPETITIVE WITH ENERGY COSTS IN THE NEW YORK CITY/LONG ISLAND AREA, WHERE ELECTRICITY IS CURRENTLY 7½ TO 10½ CENTS PER kWh.

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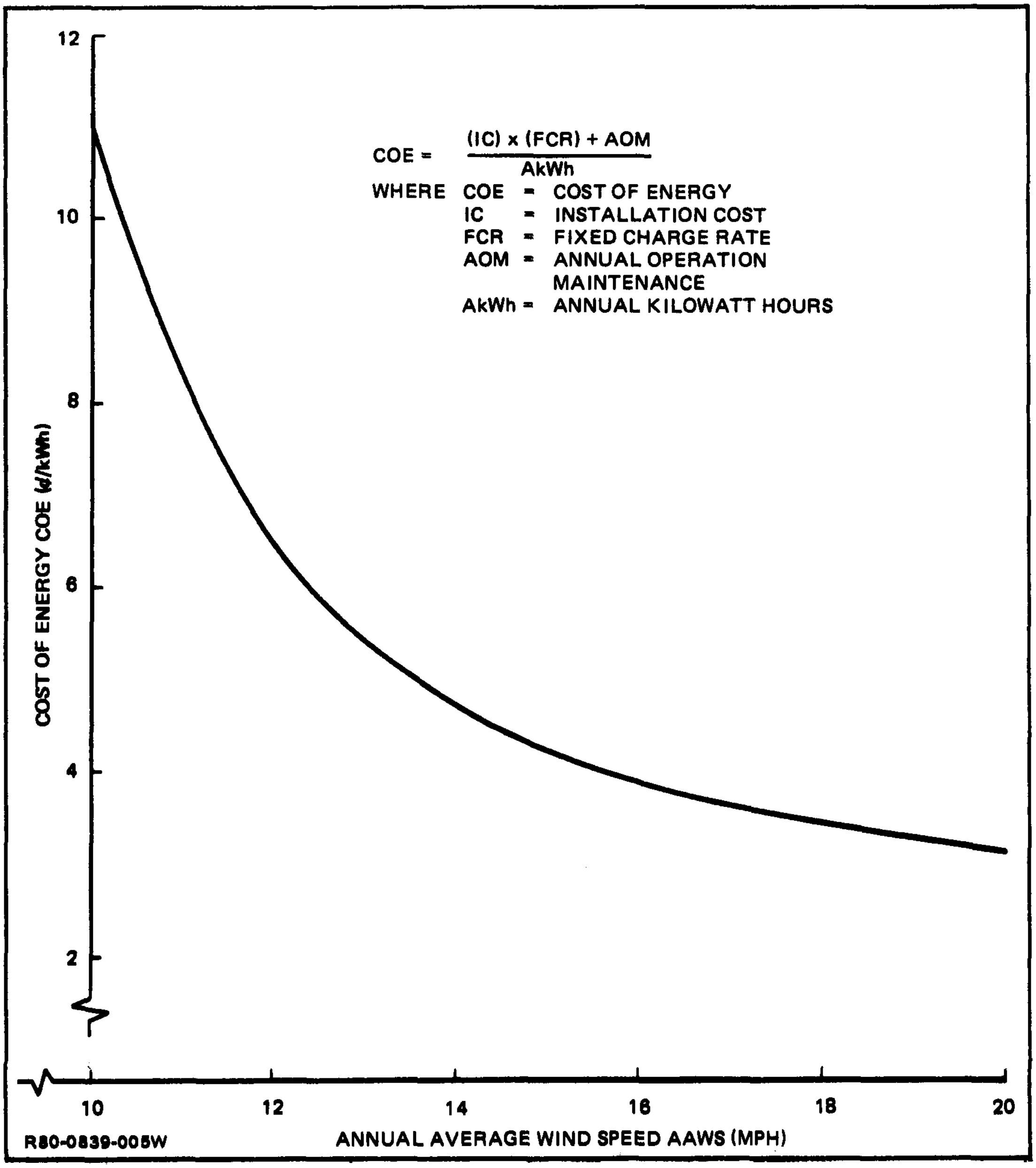

Fig. 6 Cost of Energy vs. Annual Average Wind Speed

4.0 DESIGN CRITERIA

The following criteria are the primary drivers in the design of the 8 kW WTG.

4.1 ON LINE AVAILABILITY

To fulfill the requirement of being operational for 99% of the time, the design:

- Emphasized simplicity by keeping the number of parts to a minimum
- Eliminated the need for adjustments wherever possible
- Minimized the number of greasing operations needed in routine maintenance
- Selected components with high reliability (e.g., gearbox and Pitch Control System Actuators)
- Provided ease of access to components within the nacelle by use of three quick-release doors
- Highlighted machine problem areas by using flags on the Annunciator Panel, which is situated on the ground
- Self tested the Main Control Unit which is also on the ground.

4.2 LIFETIME OF 25 YEARS

To assure an operational life of 25 years, careful attention was paid to the problems of fatigue, corrosion, wear and fretting.

4.2.1 Fatigue Loading

Fatigue loading effects on the structure were limited by the introduction of three criteria:

1. The safety factor of 2.0 for yield strength on critical loading cases automatically established a level of safety in fatigue. It was readily seen that for a machine running for 25 years, the endurance strength of a given component defines its fatigue strength. Since the endurance strength of steel and aluminum are approximately half the ultimate strength the fatigue strength is shown to be adequate.

2. Where stress concentrations were necessarily introduced into the design, effort was made to position these concentrations at areas of low stress (e.g. blade attachment points).

3. Where it was not possible to position the sections with stress concentrations away from sections of maximum stress full concentration factors were applied with the safety factors (e.g. bolt hole through vertical shaft).

4.2.2 Corrosion

Corrosion can be expected to be a significant problem for structural materials exposed to a harsh environment for over 25 years. Three methods were used for limiting the effects of corrosion:

1. Inhibition. Steels such as COR-TEN steel which develop a surface oxide layer, which further inhibits corrosion, were used for the strongback, tower pedestal, and guy rods.

2. Resistance. Stainless steel (15-5 PH) was chosen for the blade stub shafts, the low speed shaft, and the vertical shaft.

3. Protection. This is generally expensive, its reliability over 25 years not high, and it needs refurbishing. It is used on the blades (duranodic finish), and on the hub and nacelle covers (paint).

Further criteria which influenced the selection of the method for limiting the corrosion of a given structure were:

- Cost. Where competing methods of limiting corrosion were available, the lower cost method was selected.

- Maintaining cleanliness in the vicinity of bearings. Priority is given to avoiding ingress of any loosened protective particles in the bearings. This is effected by choosing corrosion resistant shafts, and also by supplying the bearings with seals.

- Accessibility. Where a component has areas which are not readily accessible, surface protection will be avoided. This is to avoid "pockets of corrosion" developing where subsequent refurbishing of the protective coating could be incomplete.

4.3 DESIGN LOADING CASES

The following maximum loading conditions were determined as the basis for combinations forming the design loading cases listed in Table V.

Table V Design Loading Cases

CONDITIONS	1	2	3	4	5	6	7	8	REMARKS
1. CUTOUT WIND WITH GUST	X	X	X	X			X	X	UP TO V_W = 50 MPH
2. TOWER SHADOWING		X		X					BLADE THRUST REMOVED
3. GYROSCOPIC FORCES	X	X	X	X					73 RPM & $\dot{\theta}_{MAX}$ = 60°/SEC
4. GRAVITY FORCES	X	X	X	X	X	X	X	X	
5. CENTRIFUGAL FORCES	X	X	X	X			X	X	AT 73 RPM
6. UNBALANCED ICE LOADS	X_x	X_x	X_z	X_z					10 LB AT TIP
7. MAX WIND					X				165 MPH WITH GUST
8. UNIFORM 2½ IN. ICE LAYER						X			UNIT STATIONARY
9. 1 IN. DIA HAILSTONE IMPACT							X		TERMINAL VELOCITY ≈ 73 FT/SEC
10. ROTOR LOCKING								X	WIND GUSTS UP TO 50 MPH

NOTE: THE TEN CONDITIONS LISTED ARE CONSIDERED TO BE THE MOST SEVERE TO WHICH THE WHOLE WTG IS SUBJECTED. VARIOUS CONDITIONS ARE RATIONALLY COMBINED TO FORM THE EIGHT DESIGN LOADING CASES. AN X DENOTES A CONDITION WHICH IS INCLUDED IN A LOADING CASE, I.E., LOADING CASE 1 COMBINES CONDITIONS 1, 3, 4, 5, AND 6. THE SUFFIXES X AND Z FOR CONDITION 6 DENOTE THE DIRECTION OF THE ROTOR ICE IMBALANCE; X IS LATERAL AND Z IS VERTICAL. WHERE LOADS CAN OCCUR IN TWO DIRECTIONS (I.E., Z UP OR Z DOWN), THE MORE ADVERSE IS CHOSEN.

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4.3.1 Cut-out Wind, Including Gusts

The Main Control Unit (MCU) signals the Rotor Blade to move from the High Speed run position to the feather position, in either of two conditions:

1. A steady wind above 35 mph for a specified period

2. A gust which exceeds 50 mph

The second condition gives the more critical loads. Associated with this condition are rotor thrust forces of 890 lbs. and torque of 2,174 ft.-lb. These loads occur at a rotor speed of 73 rpm which corresponds to an induction generator speed of 1,835 rpm.

4.3.2 Tower Shadowing Effects

When a blade passes behind the tower, it will be subjected to off-loading from the reduced wind velocity in the wake of the tower. This results in a cyclic thrust and pitch couple about the hub which is three times the rotational speed of the rotor. Assessment of the magnitude of the reduction is difficult for an isolated tower, and when rotor interference is considered, it becomes even more complicated. For the purpose of structural analysis, it was conservatively assumed that the blade completely off-loaded in the tower shadow.

4.3.3 Gyroscopic Forces

Changes of wind direction will cause the 8 kW WTG to respond in yaw. The maximum yaw rate is estimated at 60°/sec. This yaw rate, when combined with a rotor speed of 73 rpm, produces a pitching moment ("nose-up" or "nose-down") of 176,000 in.-lb. at the hub. Even though this is a transient condition, the structure is designed to withstand this load.

4.3.4 Gravity Forces

The static forces resulting from the weight of the unit and the weight of supported components are included in all analyses.

4.3.5 Centrifugal and Coning Forces

The centrifugal forces acting on the rotor are determined for the operating speed of 73 rpm. Coning of the rotor blades was introduced, so that the eccentricity of the centrifugal forces produce moments on the blade which partially relieve moments produced from the thrust forces. These coning forces are included in the design loading cases.

4.3.6 Unbalanced Ice Forces

To allow for uneven build-up or shedding of ice on the rotor, an equivalent imbalance of 10 lbs. on the tip of one blade was considered.

4.3.7 Maximum Wind with Gust

A maximum wind of 165 mph is analysed for the machine in the feathered position. In this condition, the rotor will not be rotating, and the WTG is considered under static pressure only.

4.3.8 Uniform 2½ In. of Ice

A build-up of a uniform 2½ in. layer of ice of density 54 lb./ft.^{3} is considered for a non-operational machine. The increase of gravitational force from this condition does not have a significant design impact.

4.3.9 One In. Diameter Hailstone Impact

The effect of impact of 1 in. diameter hailstones is compared with tests listed in the literature. The only sensitive element of the machine is the leading edge of the blade, at the tip. It is found that the 0.16 in. thick aluminum leading edge can easily withstand this hailstone impact. (See Refs. 1 and 2.)

4.3.10 Yaw Locking of Nacelle to Tower

During maintenance operations on the nacelle, a pin is inserted between the nacelle and the tower pedestal, preventing yawing. Maintenance operations are not expected in winds which gust above 50 mph. The wind direction is considered normal to the blades with a drag coefficient of 1.3 effective.

4.4 SAFETY FACTORS

Safety factors are introduced into the design of the 8 kW WTG to provide structural protection against: (1) Random material defects, (2) Manufacturing defects, (3) Minor service damage to parts, (4) Loads above specification.

Presently, there are no industry-wide safety factors established for WTG's. An extensive review was made of other industries safety factors to guide the selection for WTG's. These included those used in: (1) Aircraft design, (2) Buildings and other structures, (3) Bridges, (4) Structural supports for Highway Signs, Luminaires, and Traffic Signals, (5) Aluminum structures, Steel structures, Concrete structures.

Based on information obtained from these sources, a general safety factor of 2.0 on yield strength was extracted. This was judged to be a good compromise between supplying adequate strength and maintaining a cost effective machine. Zero, or positive margins of safety {Yield strength of material/Factored stress on material - 1 $\geq$ 0} are obtained. A further factor of 1.15 is applied to joints.

For ground handling equipment, where personnel safety is involved, a safety factor of 5.0 is applied and compared to the ultimate strength of the components.

4.5 MATERIAL SELECTION

The following criteria are used in determining materials for the 8 kW WTG:

4.5.1 Strength

Strength is obtained by material selection (steel, aluminum, etc.), heat treatment, and section size.

4.5.2 Corrosion Protection

All components will be subjected to moisture during the 25 years life of the machine. Methods of dealing with corrosion are:

1. ACCEPTANCE. Allow a limited protective film of corrosion to develop such that further corrosion is inhibited.

2. AVOIDANCE. Use of materials, such as stainless steels, fiberglass, plastic, etc., which show limited or no response to corrosion.

3. PROTECTION. Avoid ingress of corrosion to the surface of components by paint, chrome plating, epoxy coating, etc.

   - Cost. The least expensive materials which are compatible with other criteria will be used.

   - Weight. Weight of parts will be minimized by material selection, again, as compatible with other criteria. Generally, material removal will not be used as a means to weight saving, as this is not cost effective.

   - Weldability. This criteria applies where the selected method of construction is welding.

4.6 STRUCTURAL DYNAMICS

The structure of the 8 kW WTG supports rotating machinery in the form of the rotor, low and high speed shafts, and the induction generator. The possibility exists that the rotating machinery will excite the structure if the rotation speed is close to the natural frequencies or harmonics of these structures. Operation of the rotor/Induction Generator, other than during start up or shut down, is over a very limited range, close to 75 rpm at the rotor, which limits the range of critical excitations. To assure that no excessive vibration problems exist, an extensive analysis was conducted, (see Section 6.4). The following excitations were analysed:

1. Blade excitation (once per revolution) from tower shadowing

2. Blade excitation (three times per revolution) as rotor response to tower shadowing

3. Blade excitation from an unbalanced ice load

4. Blade flutter occurring where the torsional and bending frequencies closely match, causing a coupling between these two modes

5. Tower excitation (three blade passages per revolution) from tower shadowing

6. Transmission torsional excitation from unbalanced blade weight

7. Tower vortex shedding frequency matching the tower lateral bending frequency

8. Guy rod excitation from the wind.

5.0 PARAMETRIC AND TRADE-OFF STUDIES

During the Preliminary Design Phase, a number of parametric and trade-off studies were conducted. These were in the areas of Power and Energy Optimization, Power Conversion, System Control, and Structural Optimization. Details of these areas are found under the following headings:

The parametric and trade-off studies were addressed towards:

- Minimizing material and labor costs
- Optimizing the performance and minimizing the cost per kW and/or cost per kWh
- Improving the reliability of the WTG
- Simplifying installation
- Eliminating dynamic problems.

5.1 OPTIMIZATION OF ROTOR DIAMETER

In selecting the rotor design, consideration was given to a two-bladed rotor. However, it was seen that a two-bladed rotor has both a varying yaw and pitch inertia as a function of blade position. For rotors of three blades or more, the yaw and pitch inertias are equal and independent of blade position. Therefore, the three-bladed rotor was expected to run smoother than a two-bladed rotor. Also, the cyclic thrust load amplitude would be greater for the two bladed rotor, again making for uneven running. From these considerations a three-bladed rotor was chosen.

The rotor diameter went through three stages during design. In the first stage of the design the existing Windstream 25 blade extrusion was chosen. A rotor diameter of 28.3 ft. was found to satisfy the requirement of delivering 8 kW in a 20 mph wind. In fact, a predicted power of 8.8 kW was delivered at 20 mph through an Alternator/Synchronous Inverter system. At this time, the system had a rating of 20 kW, thus allowing the rated wind speed of 26.3 mph to be set. In winds above this speed, blade pitch control would limit power output to 20 kW up to the cut-out wind speed. The rotor diameter was limited to 28.3 ft. by blade root strength. Although the only criteria used at this time was power production in a 20 mph wind, the WTG obviously could produce more annual energy with the 26.3 mph rated wind speed.

Subsequently, the rated wind speed was fixed at 20 mph, allowing the rotor diameter to increase to 32 ft. Power output in a 20 mph wind for a 32 ft. diameter rotor was 11.1 kW, still using the Alternator/ Synchronous Inverter system. In these first two stages of design, a 2 ft. chord 6063-T6 aluminum blade was considered, with a maximumum rotor speed of 125 rpm for the 28.3 ft. rotor diameter and 88 RPM for the 32 ft. rotor diameter. The dominant design consideration was from gyroscopic forces on the blade, with associated thrust and centrifugal forces. No coning was considered.

During the third stage of design, the power conversion system was changed to a three phase induction generator. This had the structural advantage of limiting the maximum rotor speed to 73 rpm, thus limiting gyroscopic forces. Further changes in blade construction from 2 ft. chord to 1 1/2 ft. chord, and from 6063-T6 to 6061-T6 aluminum, impacted maximum rotor diameter. Coning of the rotor was also introduced (Section 5.11) to partially relieve the effects of thrust forces on the blade. This resulted in a maximum diameter of 33.5 ft., and an output of 11.2 kW in a 20 mph wind when the maximum power coefficient is developed at this speed. During this stage of the design it was recognized that annual energy production was a more effective method of evaluating the WTG than obtaining a particular power at a particular wind speed. For optimum energy production in a 12 mph annual average wind speed, it was found that at the best reference wind (wind at which maximum power coefficient is developed), the power at 20 mph reduces to 10.8 kW. This is discussed in more detail in Section 5.3.

5.2 OPTIMIZATION OF BLADE SIZE

Development of the blade was intimately connected with the rotor diameter development (see 5.1). Following Grumman experience with the Windstream 25, it was decided to make one-piece extruded aluminum blades.

The airfoil section which was developed after investigating 15 sections for the Windstream 25 was continued. It was found from the WS-25 work, and also supported by the literature (see p. 147 of "Power from the Wind" by P. C. Putman), that the output from a rectangular blade is relatively insensitive to the amount of twist. In the early stage of the design the Windstream 25 blade was used. This was a 2 ft. chord, 5.04 in. maximum depth 6063-T6 extrusion (yield strength = 25,000 psi). The extrusion manufacturers informed us that 6061-T6 (yield strength = 35,000 psi) could not be used for this size blade, but that it could be used for a 1.5 ft. chord. By including a coning angle with the blades to partially relieve the thrust terms on the blade, this lighter blade was found preferable and selected. The change in the blade chord reduced the solidity and had only a small impact on performance. Further advantages in using the lighter bladed, coned rotor was that it allowed a better weight balance of the WTG about the vertical shaft, and that it allowed a reduction in the nacelle length. The airfoil type selected was NACA 64_4 - 421 (modified).

5.3 OPTIMIZATION OF REFERENCE WIND SPEED

The reference wind speed is defined as the wind speed at which the maximum power coefficient is developed. For a given blade setting (β), the power coefficient versus tip speed ratio will typically follow the "humped" shape of Fig. 7. For the blade setting shown in Fig. 7, the maximum power coefficient occurs at a tip speed ratio of 5.25. When the rotor speed has been established, the reference wind speed will be obtained from the expression $V_{REF} = N \pi D/\lambda$. For the 8 kW, these values are: rotor speed N = 73/60 = 1.217 revolutions per second; rotor diameter = 33.5 ft.; and tip speed ratio at Cp max. = 5.25. The reference wind speed V_{REF} = 24.4 ft/sec = 16.63 mph.

Since power production is through an induction generator which has an approximately constant speed, then the WTG rotor similarly follows a constant speed. The gearbox chosen for the WTG then determines the rotor speed. A change of gearbox ratio will change rotor speed, and thus the reference wind speed. Changes of the reference wind speed have a significant effect on power production versus wind speed, as shown in Fig. 9. Power production (allowing for the Induction Generator efficiencies shown in Fig. 8, and a constant gearbox efficiency) is seen to be higher for the lower reference wind speeds than the higher reference wind speeds, in winds up to about 14.5 mph. Above 14.5 mph winds the situation reverses and higher power is produced for the higher reference wind speeds. This reversal of power production is a direct reflection of the shape of the power coefficient versus tip speed ratio, as shown in Fig. 7.

It is noted in Fig. 9 that for V_{REF} above approximately 17 mph, the power output can exceed the rated power of the Induction Generator of 20 kW in winds above approximately 28 mph. To protect the WTG, the blades are feathered when power exceeds the rated power. For this reason, there will be a loss in annual energy where V_{REF} above 17 mph is chosen in the high wind periods.

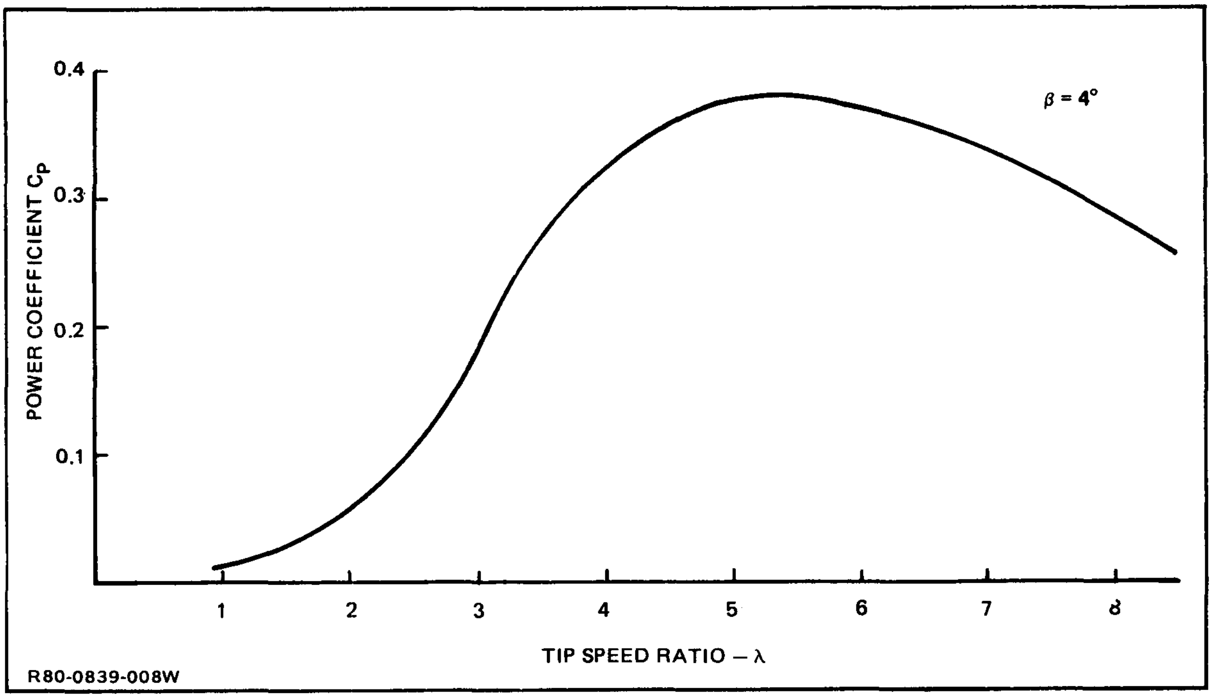

Fig. 7 Power Coefficient vs. Tip Speed Ratio

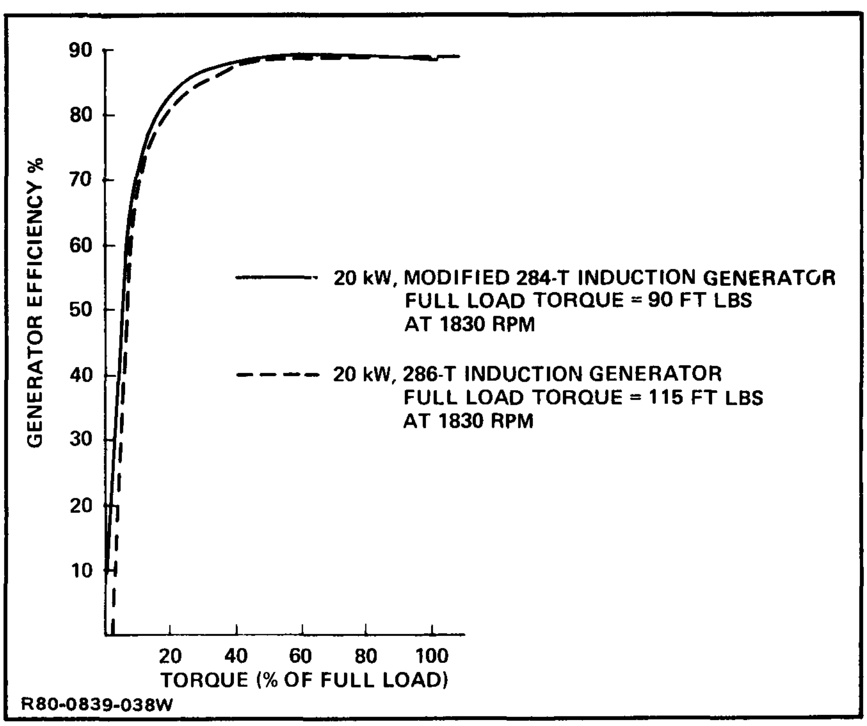

Fig. 8 Generator Efficiency vs. Generator Torque

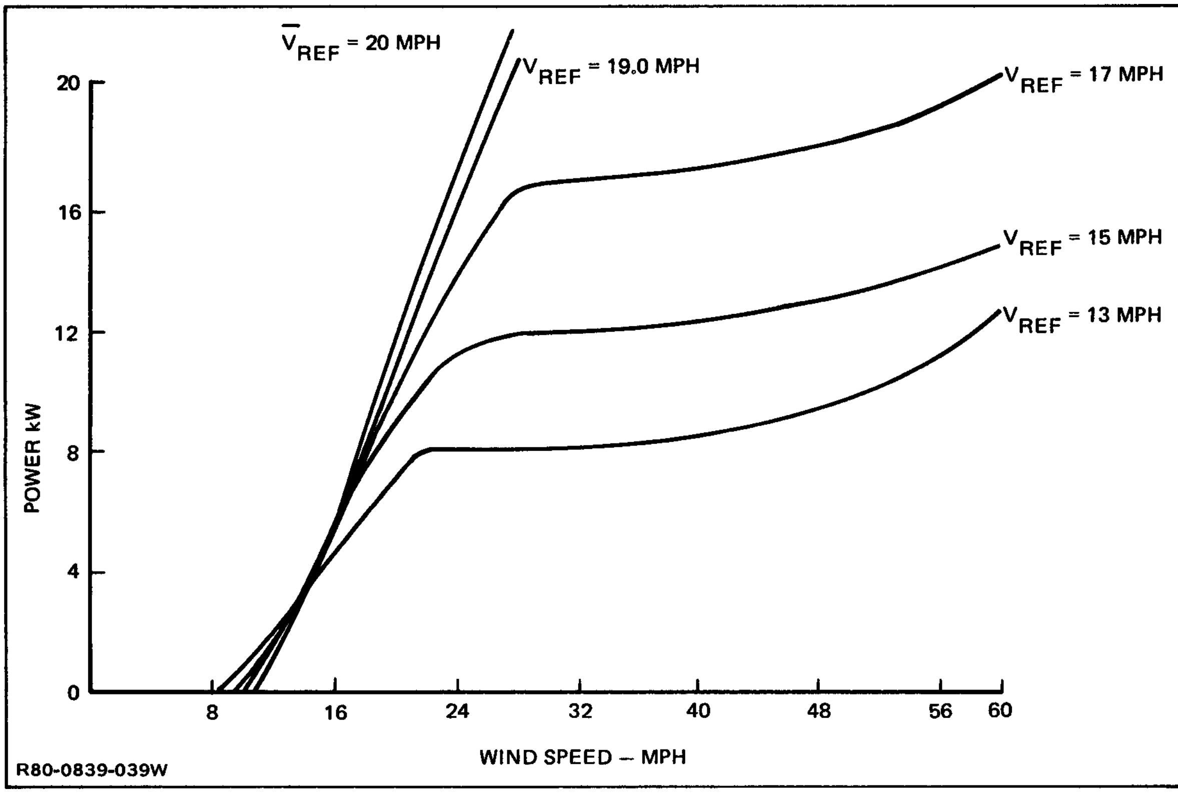

Fig. 9 Power Output at Variable V_{REF} Includes Generator and Gearbox Efficiency 33¼ Diameter Rotor, 1.5 Chord, C_p Max = 0.385

As explained in Section 5.1, the rotor design proceeded in three stages. From various program runs, the maximum values of power coefficient ranged from 0.38 to 0.42. These are shown in Table VI.

The results from the program show a small range of maximum power coefficients. It was conservatively decided to use the power coefficient plot derived for the 32 ft. rotor, and apply these coefficients to the area of the 33.5 ft. rotor in predicting the power and energy production of the 8 kW WTG.

To determine the optimum annual energy production of the 8 kW WTG, four values of Annual Average Wind Speed (AAWS) are chosen, and the energy production is obtained for V_{REF} from 14 to 20 mph. The associated gearbox ratio range is from 29.85 to 20.90. Results of this analysis are shown in Fig. 10.

Three of the values of AAWS chosen are 12, 15 and 18 mph at a height of 30 ft. above the ground. These values are converted to a height of 55 ft. above the ground using the relationship

$$V_{55} = \left\{ \frac{55}{30} \right\}^{1/7} V_{30}$$

Table VI Comparative Rotor Characteristics

ROTOR DIA (FT)	BLADE CHORD (FT)	CP MAX	SOLIDITY RATIO	ROTOR RPM
28.3	2.0	0.41	0.0964	125
32.0	1.5	0.38	0.0639	88
33.5	1.5	0.42	0.0615	73
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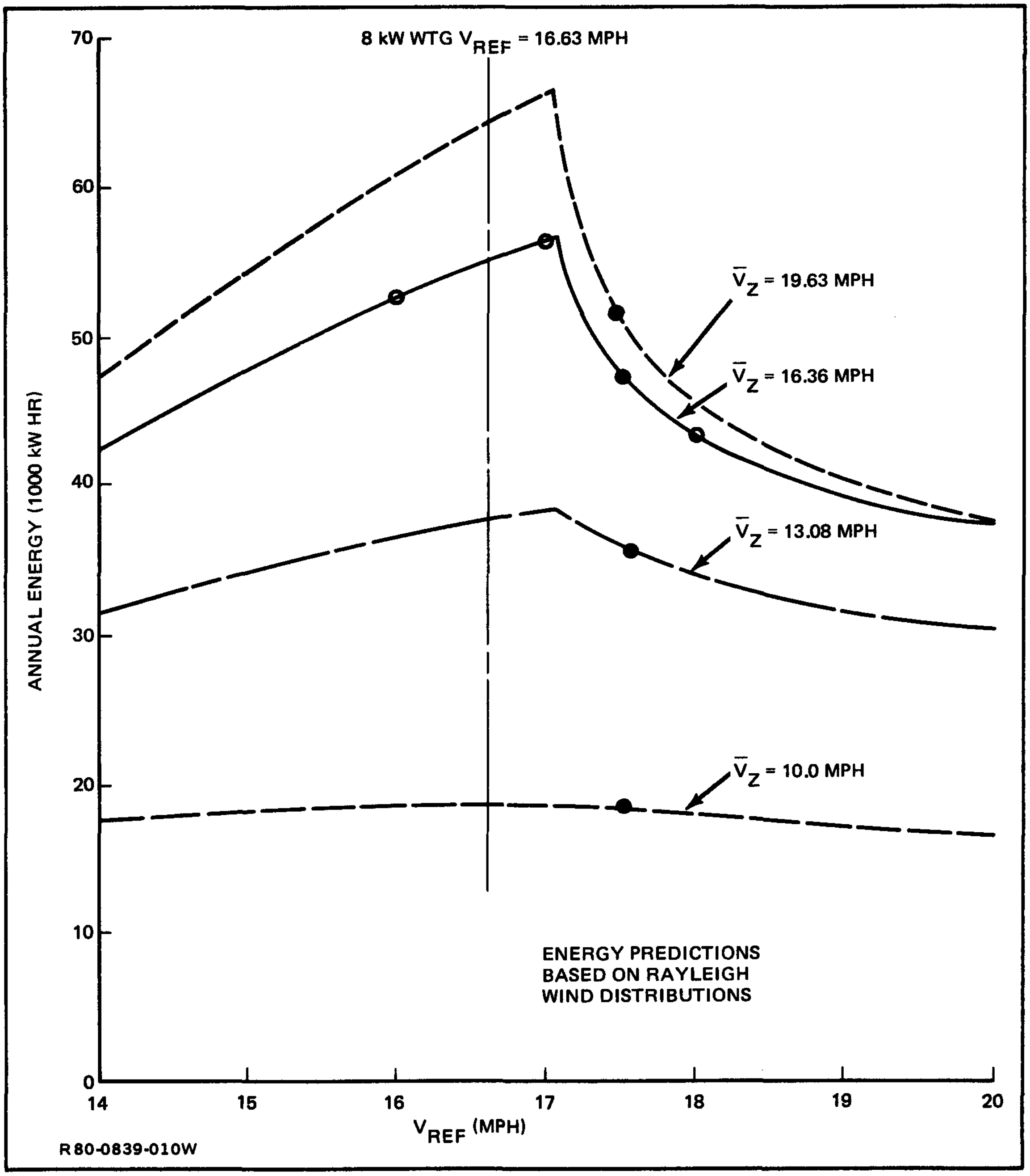


Fig. 10 Annual Energy vs. Reference Wind Speed for Four Annual Average Wind Speeds

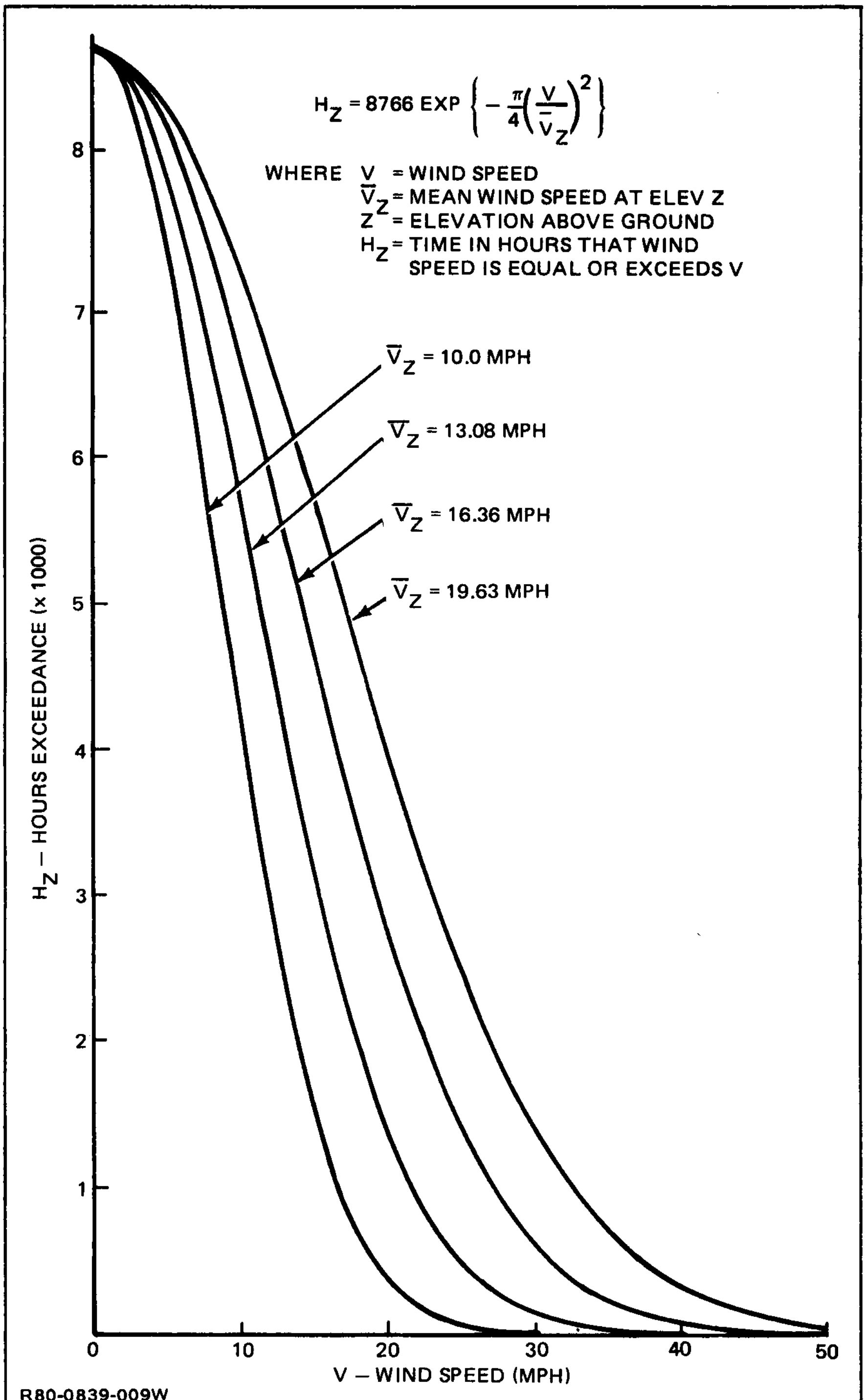


Fig. 11 Wind Speed Duration Profiles

This give AAWS's of 13.08, 16.36 and 19.73 mph respectively. A fourth AAWS of 10 mph is added.

Annual energy production for a given AAWS and V_{REF} is obtained from the expression $E = \Sigma\ (\Delta H \times \eta \times P_R)$, where ΔH is the number of hours in a given wind speed range; η is the combined efficiency of the gearbox and Induction Generator in this range; and P_R is the power from the rotor.

The wind speed from cut-in to cut-out speed (35 mph) is divided into 1 mph bands. The number of hours per year in each band is obtained from the Rayleigh distribution.

$$H_z = 8766_{EXP} \left\{ - \frac{\pi}{4} \left(\frac{V}{V_z} \right)^2 \right\} \quad \text{(see Fig. 11)}$$

Efficiency of the Induction Generator vs. percentage of full load is plotted in Fig. 8. The gearbox efficiency is a constant 95%. Rotor power is derived from expression $\rho = \rho/2\ V^3 C_p A$, where ρ is the air mass density; V is the free stream wind speed; C_p is the power coefficient; and A is the rotor diameter. For a given rotor, this can be reduced to $P = K\ V^3 C_p$, where K is a constant. From Fig. 12, it is seen that for increasing wind speeds (decreasing tip speed ratios) the C_p decreases rapidly after the maximum is achieved. For the range of V_{REF}'s considered, it is found that for winds above approximately 25 mph the decrease of C_p roughly matches the increase of V^3. Thus, above 25 mph the power output approximately holds constant, up to the cut-out wind speed (see Fig. 9). This feature exerts a strong influence on annual energy production for the 8 kW WTG, where an upper limit is set on power production at the Induction Generator rating of 20 kW. For V_{REF} greater than about 17 mph, the annual energy production reduces because the Induction Generator is not allowed to exceed its rating in winds above approximately 25 mph (see Fig. 10). As the AAWS decreases, there is less duration at the higher wind speeds, and the Induction Generator cut-off has a diminishing influence on reducing energy at the higher values of V_{REF}.

It is also seen that annual energy production increased for each of the four values of AAWS as V_{REF} increases up to the 17 mph level, (Fig. 10). This is reflected in Fig. 9, where at low wind speeds higher power is produced at the lower V_{REF}'s, but reverses in wind speeds above about 14.5 mph. Since the power produced in wind speeds above 14.5 mph is significantly greater than below, and also since the duration in these winds is greater for the chosen AAWS's, more energy is produced for the higher V_{REF}'s. The selection of V_{REF} = 16.63 mph for the prototype 8 kW is close to the optimum for energy production.

It is possible to obtain an Induction Generator of higher rating than 20 kW in order to allow greater annual energy production at the higher V_{REF}'s. However, this would have a cost impact on the Induction Generator, gearbox, slip rings, and power wiring. The current I.G.

rating of 20 kW is well above the 8 kW Work Statement requirement at 20 mph, and represents close to optimum energy production at the selected V_{REF}.

It is noted that the proceeding work is all based on prediction and needs the support of test data for confirmation. The influence of the C_p vs. λ curve is dominant in obtaining optimum energy production, and variation from the predicted values, especially at low values of λ, could change the optimum values of V_{REF}. The preceeding energy predictions did not include the influence of longitudinal or lateral gusts.

5.4 START-UP CHARACTERISTICS

During the early phase of the control system design where an Alternator system was envisaged, the blades were to be held at a fixed pitch angle up to wind speeds which would develop rated power. In winds above this rated value, the blades would move towards feather to maintain rated power until the wind reached the cut-off value, when the blades would be signaled into full feather. Subsequently, an induction generator system was chosen where the blades were to be held in one angular position from start-up through to cut-off wind speeds. Performance predictions showed that a shallow blade angle of 4° to the plane-of-rotation gave an optimum power at the high tip speed ratios. However, at low tip speed ratios of about 3.0 and less, it was found that for blade angles above 4°, the power coefficients increased, indicating superior start-up characteristics in low wind speeds (see Fig. 12).

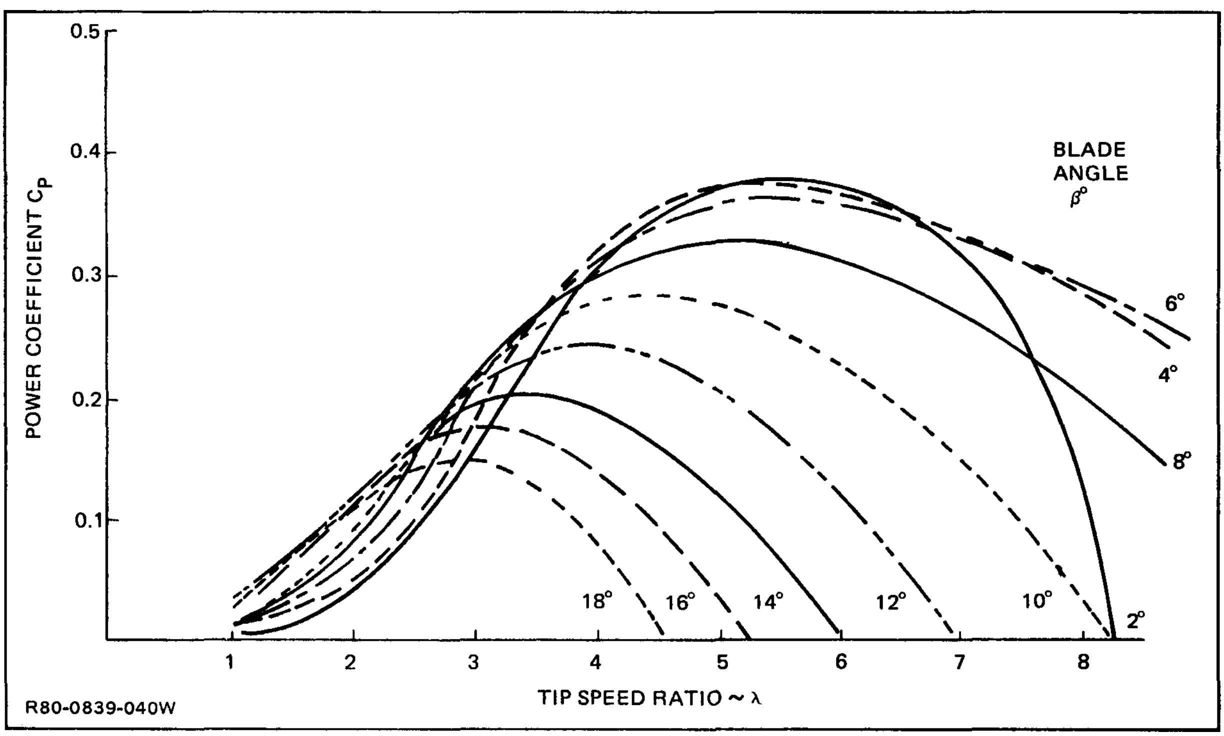


Fig. 12 Power Coefficient vs. Tip Speed Ratio

Since the lower power coefficient at 4° factored into V^3 would give very low power (not enough to overcome system friction in very cold temperatures), a start-up blade angle of 27° was chosen.

The blade setting was based upon obtaining 15 rpm in a 7 mph wind with no power extraction. The corresponding tip speed ratio is 2.54, which requires a blade setting of 27° (see Fig. 13). The selection of 15 rpm was a compromise between having sufficient rotational speed and limiting the blade travel to its high speed run position.

A further advantage in having a high value of β in low wind speeds is improved yaw characteristics. This is a result of two effects:

1. A larger blade area presentation to winds normal to the long face of the nacelle

2. The higher yaw responsiveness of the rotor because of higher rotor thrust forces

The sequence of blade positions in the start-up mode is as follows:

1. In the shut-down condition, the blades will be in the fully feathered position

2. When the "system-ready" signal is received (primary actuator extended, secondary actuator retracted, the system is ready to operate on the anemometer and rotor rpm signals

3. In winds below 7 mph, the blades remain in feather

4. In winds above 7 mph and below 35 mph, the blades are moved to the low speed run position of $\beta = 27°$

5. When the signal is received that rotor speed is above one quarter synchronous speed (18 rpm) for greater than 20 consecutive seconds, the blades are moved to the high speed run postion of $\beta = 4°$.

This sequence is presently programmed into the controller (MCU), but is easily adjusted should field test results indicate the need.

5.5 TRADE-OFF BETWEEN ALTERNATOR/SYNCHRONOUS INVERTER AND INDUCTION GENERATOR POWER SYSTEMS

A study was made between an Alternator/Synchronous Inverter and Induction Generator power system. Comparisons between the two systems are as follows:

- Efficiencies - Although a one-to-one comparison between the two systems is not possible, a survey of induction generator efficiency vs. percentage of full load shows it to be higher than the alternator with Synchronous Inverter efficiency vs. Alternator speed. Transmission efficiencies were the same for the two systems.

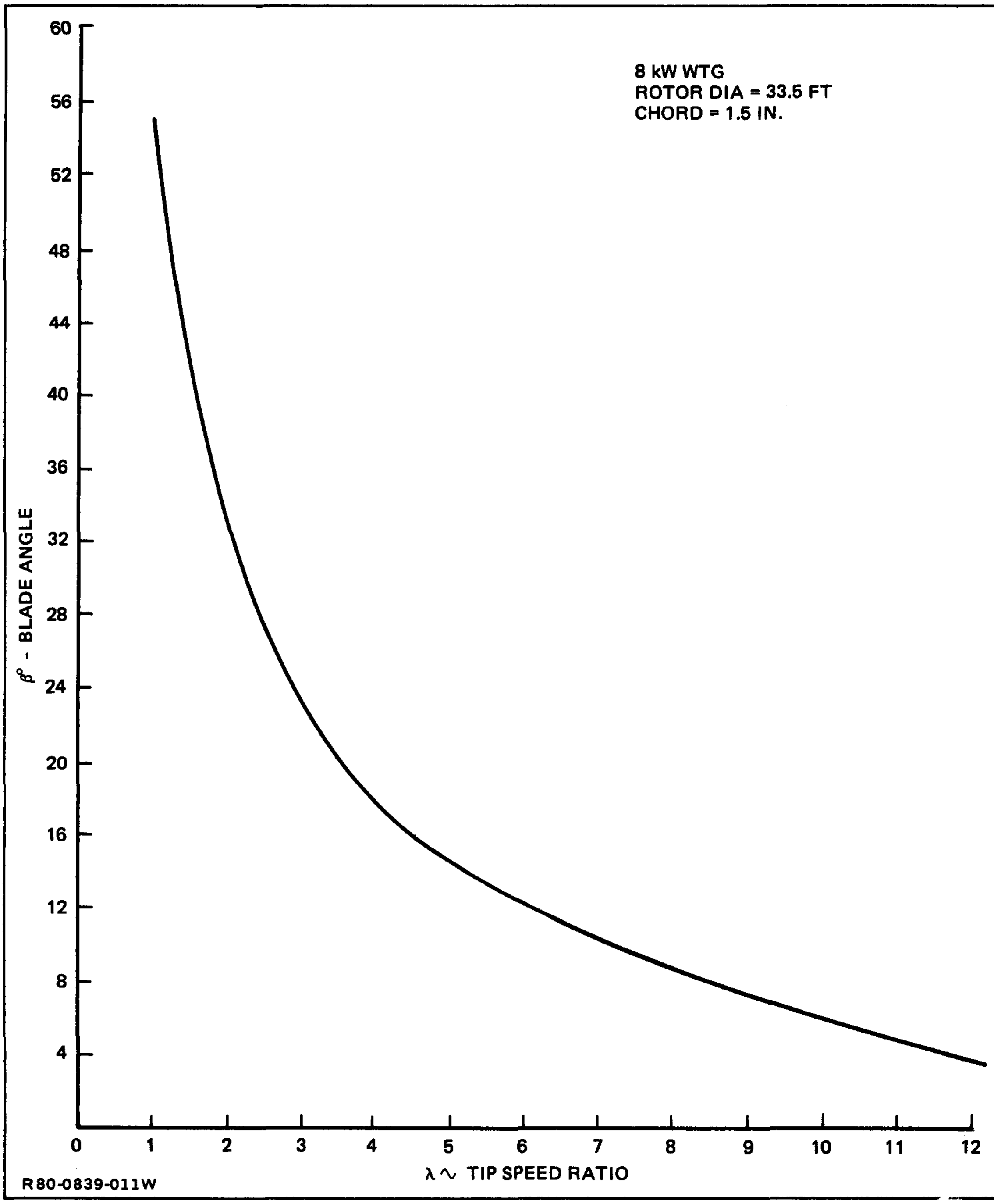


Fig. 13 Tip Speed Ratio for Zero Power Coefficient vs. Blade Angle

- Power Output vs. Wind Speed - In winds below 20 mph, the alternator produced more power than the induction generator; above 20 mph the reverse held true.

- Annual Energy Production - Based upon the annual energy equation (Section 5.3), the two systems produced the same energy for an annual average wind speed of about 11.5 mph. Above this speed the Induction Generator produced more energy than the Alternator/Synchronous Inverter system.

- Cost of Unit and Cost of Energy - The Alternator/Synchronous Inverter installation cost of $18,533 exceeded that of the Induction Generator by $1,192 (see Table VII). For comparison of the cost of energy reference wind speeds of 16 and 20 mph were chosen for the generator. In annual average winds of 11.8 mph the 16 mph reference wind speed showed a greater annual energy production, resulting in a 1¢/kWh saving over the Alternator system. In annual average winds of 14.5 mph this saving increased to 2¢/kWh (8¢ vs. 10¢/kWh). There was a subsequent system refinement of the reference wind speed to 16.63 mph.

Table VII Cost of Energy

MOD 4 WORK STATEMENT

$$COE = \frac{(IC)\ (FCR)\ +\ AOM}{AKWH}$$

IC = "TURNKEY" COST

FCR = FIXED CHARGE RATE = 0.18/YR

AOM = OPERATION AND MAINTENANCE COST

AKWH = ANNUAL KW-HR PRODUCED

	ALTERNATOR SYNCHRONOUS INVERTER	INDUCTION GENERATOR (20 kW)	
MACHINE COST	$11,033	$11,324	
GEMINI/TRANSFORMER	2,000	517	
INSTALLATION	5,500	5,500	
I.C.	$18,533	$17,341	
AOM	$ 220	$200	
AKWH (V = 11.8)	21,979 kW-HR	21,561 (20 MPH)	22,573 (16 MPH)
(V = 14.5)	35,438	34, 232	40,579
COE $/kW-HR (V = 11.8)	0.16	0.15	0.15
(V = 14.5)	0.10	0.10	0.08

R80-0839-041W

- Power Regulation - There are significant differences of power regulation requirements between the two systems. For the Alternator/Synchronous Inverter system blade pitch is held constant in winds up to rated, the rotor rpm varying directly with wind speed. Above rated winds, up to the cut-off speed, the blade pitch is continuously regulated to hold the rpm and power output to a fixed value. Wind gusts may cause transient increases of rpm and power output. Above the cut-out speed the blades are feathered. For the Induction Generator system, where the rotor rpm is held approximately constant by the generator, the blade is held to a fixed angle throughout the range of winds which produce power. There is no appreciable change of rpm in gusts due to the nearly constant speed control of the generator. Above the cut-out wind speed the blades are feathered.

The Alternator/Synchronous Inverter system was seen to have the following disadvantages as compared to the Induction Generator system:

1. A more complicated power regulation system, where the blade angle would be continously changing in varying winds above rated

2. Higher thrust and gyroscopic forces on the rotor, resulting from higher rotor rpm (125 vs. 73)

3. Reduction of reliability of the power regulation system, resulting from more extensive use.

Based upon the preceding comparisons, the decision was made to utilize the Induction Generator system.

5.6 TRADE-OFF BETWEEN SINGLE AND 3 PHASE INDUCTION GENERATOR POWER SYSTEMS

During the design stage a study was made of a single phase induction generator to determine if it was competitive with the 3 phase generator. However, due to the following disadvantages found in the single phase unit, the 3 phase generator was selected.

- In the present "state-of-art" of Induction Generators, a 3 phase machine would have to be modified to single phase operation. The cost of the single phase machine was $688 higher.

- A transformer is needed to give 120-240 V output, which is not needed in the 3 phase generator.

- Although efficiencies were much the same for the two units, the Power Factor was significantly lower for the single phase system.

- Heat distribution within the single phase machine was unbalanced.

- Although the same frame could be used for each generator, the single phase machine was heavier.

5.7 TRADE-OFF STUDY FOR TRANSMISSION SYSTEM

An assessment of the relative merits of a step-up gearbox and a belt and pulley transmission was made in terms of:

- Satisfying a 25 year life
- Satisfying loading requirements
- Withstanding the wide temperature extremes specified in the work statement
- Resisting the effects of ice
- Resisting the effects of salt
- Producibility
- Reliability
- Maintainability
- Complexity
- Cost of initial purchase
- Development cost
- Development risk.

As a result of this assessment, the step-up gearbox was chosen (see Table VIII).

Table VIII Transmission

CRITERIA	GEAR BOX LO SPD – HI SPD	BELTS & PULLEY	LOW SPEED ALTERNATOR
25 YEAR LIFE	10	9	10
PRODUCIBILITY	9	9	3
RELIABILITY	10	9	10
MAINTAINABILITY	10	7	10
COMPLEXITY	10	7	10
COST	9	10	3
DEVELOPMENT COST	10	10	5
DEVELOPMENT RISK	10	10	8
ENVIRONMENTAL: TEMP	10	10	10
: ICE	10	10	10
: SALT	10	10	10
TOTALS	108	101	89

NOTE: THE ABOVE TABLE IS A CONSENSUS OF THE CRITERIA MERITS RANGING FROM 0 (UNACCEPTABLE) TO 10 (FULLY SATISFACTORY)

R80-0839-042W

5.8 TRADE-OFF STUDY FOR PRIMARY ROTOR SPEED CONTROL SYSTEM

The following primary rotor speed control systems were investigated during the study phase of the design:

- Electro-Mechanical, such as electrically powered screw actuator
- Hydraulic, such as hydraulically powered linear actuator
- Passive (Self-regulating), such as spring biased centrifugal weight
- Fixed blade with Azimuth Control, such as yaw drive motor.

The rotor speed control is vital to the safety and performance of the WTG. A final selection of the Electro-Mechanical system was arrived at after reviewing systems operations (see Table IX).

Table IX Primary Rotor Speed Control

CRITERIA	ELECTRO-MECHANICAL	HYDRAULIC	PASSIVE SELF-REGULATING	FIXED PITCH WITH AZIMUTH CONTROL
25 YEAR LIFE	9	7	10	8
PERFORMANCE (ABOVE RATED SPEED)	10	10	6	3
PRODUCIBILITY	8	9	10	6
RELIABILITY	9	8	10	6
MAINTAINABILITY	9	7	10	7
COMPLEXITY	9	7	10	6
COST	9	9	10	7
DEVELOPMENT COST	10	9	2	7
DEVELOPMENT RISK	10	9	4	2
ENVIRONMENTAL: TEMP	10	9	10	8
: SALT	10	10	10	8
: ICE	10	10	5	9
TOTALS	113	104	97	77

NOTE: SEE NOTE TO TABLE VIII
R80-0839-043W

- Over 25 year life
- Operation in high winds
- Producibility
- Reliability
- Maintainability
- Complexity
- Cost of initial production

- Development cost
- Development risk
- Withstanding the environment risks of temperature extremes, salt atmosphere, and ice formation.

5.9 TRADE-OFF STUDY FOR BACK-UP SPEED CONTROL SYSTEM

To enhance the reliability of the rotor speed control system, it was decided to add a back-up system. Functioning of the back-up system was to be completely independent from the primary system. Necessarily, the back-up systems were reviewed for the same criteria as the primary system (see Section 5.8). The systems reviewed were:

1. A stand-by actuator. This actuator has a separate power source (permanent magnet generator) independent of the primary actuator but using the same linkage to the blades.

2. Centrifugal latch. In the event of the rotor starting to overspeed the centrifugal forces developing on a mass would depress a spring, allowing release of a latch which would position the blade angle. Aerodynamic moments on the blade would then cause the blade to rotate into feather.

3. Blade tip speed brake as used on the WS-25. In routine operation this brake would be held closed by spring operation. In the event of rotor overspeed, centrigual forces on the brake would overcome the spring forces, deploying the brakes and thus subjecting them to aerodynamic forces. These brake forces would produce a torque at the hub of opposite sense to those produced by the blades and thus limit rotor rpm.

4. Solenoid release latch. Failure of electrical power would release a solenoid spring which would release a latch which would force a spring to operate the pitch control system.

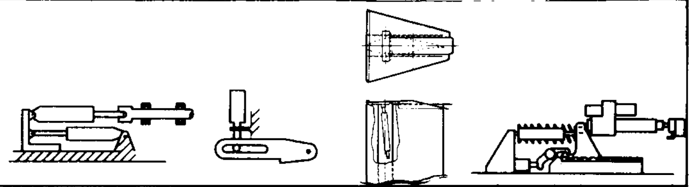
Results of the review are shown in Table X, and the stand-by actuator was chosen upon this basis.

5.10 TRADE-OFF STUDY FOR PARKING SYSTEM

Three schemes for parking the blades were compared during the design stage (see Table XI).

1. Electric Brake. Two versions of this brake were considered: one where power is applied to hold off a spring during WTG operation and power is removed for brake engagement; the other version applied power, and thus braking, when signaled. The first version had the disadvantage that if the power supply failed during WTG operation, brake operation could introduce

Table X Back-Up Over Speed Control



CRITERIA	STAND BY ACTUATOR	CENTRIFUGAL LATCH	W.S. 25 TIP SPEED BRAKE	SELENOID RELEASE LATCH
25 YEAR LIFE	6	8	8	6
PERFORMANCE	10	7	1	7
RELIABILITY	10	8	7	7
MAINTAINABILITY	9	7	10	8
COMPLEXITY	9	6	10	7
COST	8	10	7	9
DEVELOPMENT COST	10	5	2	9
DEVELOPMENT RISK	10	5	2	9
ENVIRONMENTAL: TEMP	10	10	10	10
: SALT	9	9	8	9
: ICE	10	3	1	10
TOTALS	101	78	66	91

NOTE: SEE NOTE TO TABLE VIII
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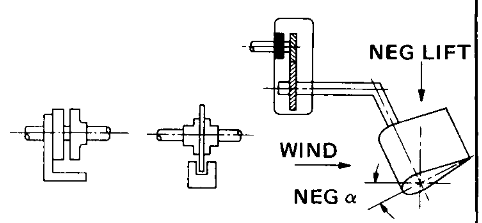
Table XI Brakes-Parking



CRITERIA	ELECTRIC BRAKE	CALIPER DISC BRAKE	GEAR BOX BACK STOP
25 YEAR LIFE	7	8	10
PERFORMANCE	10	10	5
COMPLEXITY	7	8	10
RELIABILITY	8	9	10
MAINTAINABILITY	7	8	10
COST	7	8	10
DEVELOPMENT COST	7	9	10
DEVELOPMENT RISK	10	10	3
ENVIRONMENTAL: TEMP, SALT, ICE	9	9	10
TOTAL	72	79	78

NOTE: SEE NOTE TO TABLE VIII
R80-0839-045W

considerable dynamic effects on the rotor; also, brake sizing had to be increased to handle this load. For these reasons, only the second version was considered.

2. Caliper Brake. Operation of this brake was a function of the blade position. During the high speed run where the swivel link is in the forward position, a cable from the swivel link disengages spring-loaded pucks from the brake disk. In travelling towards the feathered blade position, movement of the swivel link will allow the spring to release the pucks onto the brake. This occurs at 21 1/2° from the nominal blade feather position (see Fig. 14).

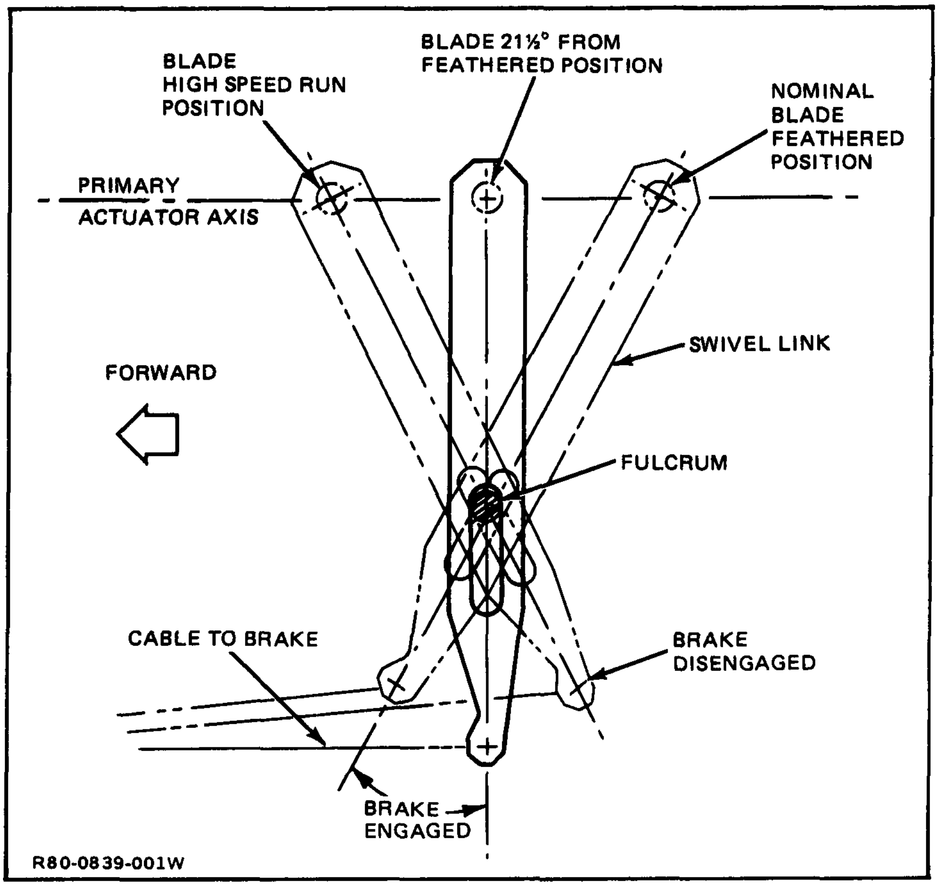


Fig. 14 Relationship of Blade Position to Brake Operation

3. Gearbox Back Stop. In this design the limit in blade travel is past 90° from flat pitch by about 5°. This limit is supplied by the actuator travel. In this position the aerodynamic loading of the rotor will cause it to run backwards. However, this is prevented by the back-stop within the gearbox, and the rotor will be parked. As seen in Table XI, there is little difference between the second and third designs. The design selected for the prototype unit was the caliper brake version. However, the gearbox will be supplied with a back-stop and each design will be evaluated on the prototype unit.

5.11 TRADE-OFF STUDY FOR ROTOR CONING SYSTEM

The initial design of the rotor showed the rotor plane normal to the drive shaft axis. Subsequently, the effect of "coning" the blades was studied to determine its effectiveness in reducing cost and providing structural improvement.

There are two major terms causing blade bending: gyroscopic moments and blade thrust forces. The gyroscopic moments are bi-directional (either bending the blade tip forward or aft), depending upon the yaw direction, and whether the blade is at 12 o'clock or 6 o'clock. Thrust forces are uni-directional and only cause the tip to move aft. However, there is a range of thrust acting on the blade which is dependent upon whether the blade is in the free stream or in the tower shadow.

The immediate advantage of coning was in causing bending loads to develop on the blades from centrifugal forces, which were the opposite of those developed from aerodynamic thrust forces. This allowed for lighter blade construction. A further advantage was that by retaining the blade at the same three-quarter radius from the tower, the coned rotor allowed the hub to move forward. In this manner, both the nacelle weight and cost were reduced, and the weight balance of the unit about the vertical shaft was improved. There was no significant change in performance. The only disadvantage resulted from centrifugal forces, which increased blade torques about their axes, and increased the loading demands on the pitch control actuators. This increase was not prohibitive and the decision was made to cone the blades to 3° 25'. The 3° 25' coning angle gives rise to bending moments which are half the range to which the blade is subjected from thrust forces, and it forms the maximum relief.

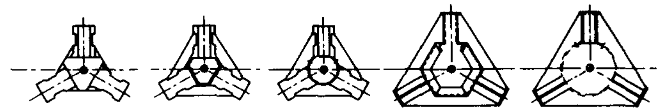
5.12 TRADE-OFF STUDY FOR HUB DESIGN

Five hub design concepts were reviewed during the study phase of the design (see Table XII). Three manufacturing methods were studied: cast, forging, and welding; and two geometries were studied, hexagonal and round hubs. Although the hexagonal or round hub casting had the highest rating, it will only be incorporated in the production design because of the high cost for the casting mold. For the prototype machine the design selected was the welded round hub.

5.13 TRADE-OFF STUDY FOR BLADE CONCEPTS

The four concepts reviewed in arriving at the final blade design are given in Table XIII. Of the four, the constant chord, untwisted aluminum blade was superior to the two tapered and twisted versions. The fabric blade was completely inadequate in conforming to four of the criteria. The untwisted aluminum blade was selected and further reviewed in three different chord sizes of 2 ft., 1 1/2 ft. and 1 ft. This review covered the following topics:

Table XII Rotor Hub Concepts



CRITERIA	OPEN HUB CASTING	HEX HUB FORGING	HEX OR ROUND HUB CASTING	HEX HUB WELDED	ROUND HUB WELDED
25 YR LIFE	10	10	10	8	8
PRODUCIBILITY	10	7	9	5	6
WEIGHT	6	8	7	9	10
COST	6	6	10	8	9
DEVELOPMENT COST	7	5	7	8	8
ENVIRONMENT: SALT	10	10	10	8	8
TOTALS	49	46	53	46	49

NOTE: SEE NOTE TO TABLE VIII

R80-0839-046W

Table XIII Blade Concepts

CRITERIA	ALUMINUM CONSTANT CHORD – UNTWISTED	TAPERED AND TWISTED		FABRIC
		ALUMINUM BUILT-UP	COMPOSITE	
25 YR LIFE	10	10	10	1
PERFORMANCE	8	9	10	5
PRODUCIBILITY	10	5	5	5
MAINTAINABILITY	9	9	9	1
WEIGHT	8	7	10	9
MATERIAL COST	9	7	5	10
DEVELOPMENT COST	10	7	5	8
DEVELOPMENT RISK	10	6	5	6
ENVIRONMENTAL: TEMP	10	10	10	8
: SALT	8	8	8	1
: HAIL	9	9	8	1
TOTAL	101	87	85	55

NOTE: SEE NOTE TO TABLE VIII

R80-0839-047W

5.13.1 Performance

Each of the three blade types were analyzed for their power and thrust coefficients over a range of blade angular positions. Generally, it was found that the lower chord blades had lower power and thrust coefficients at their corresponding optimum blade angles. For instance, the drop in power coefficients was 9.6% between the 2 ft. and 1 1/2 ft. chord (see Fig. 15). Power could be returned by increasing the rotor diameter.

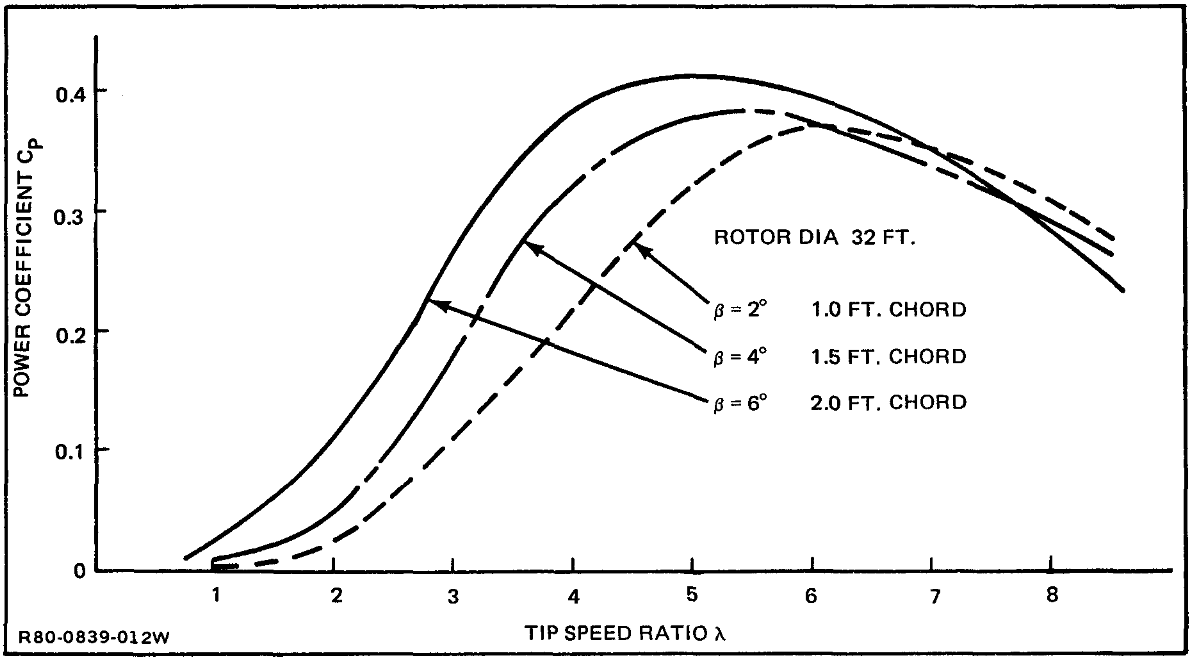


Fig. 15 Comparative Optimum Performance of Three Blades

5.13.2 <u>Influence on Loads to WTG and Tower</u>

The two major analytical terms defining loads to the WTG and the tower are rotor thrust forces and the gyroscopic forces. Rotor thrust forces were partially balanced by coning loads, locally alleviating loads to the blades though still influencing the rest of the structure. The gyroscopic force is the product of the rotor mass moment of inertia and the rates of rotation about the horizontal and the vertical axes. Only the mass moment of inertia varied with change of blade construction. As an approximation, the blade mass varied linearly with change of chord (the wall thickness remaining constant) and the square of the rotor diameter. This resulted in an 18% reduction of gyroscopic moment in changing from a 2 ft. chord, 32 ft. diameter rotor, to a 1 1/2 ft. chord, 33.5 ft. diameter rotor.

5.13.3 <u>Blade Natural Frequency</u>

Both reduction of the chord dimension, from 2 ft. to 1 1/2 ft. (where stiffness under its own mass decreases), and increase of rotor diameter, from 32 ft. to 33.5 ft., reduced the natural frequency of the blade. This resulted in a 31% reduction of the blade's first bending frequency to 3.97 Hz. Although this is close to 3 times the nominal operating frequency of 1.25 Hz (3 pulses per revolution), it is not anticipated to be a problem (see Section 6.4).

5.13.4 <u>Blade Materials</u>

A further advantage in incorporating the smaller chord aluminum extruded blade arose from material limitations. Increasing the chord

Table XIV Cost of Aluminum Extrusion (1½ Ft. Chord)

COMPANY / ITEMS	ALCOA (8/2/78)	TABER METALS (2/20/79)
DIE COST	$ 19,000	$19,500
SET UP CHARGE	$ 2,500	$4,000 (0-60 UNITS) $ 0 (300 UNITS)
EXTRUSION COST	$ 1.577/LB	$1.560/LB
DELIVERY	LATER THAN SCHEDULE	ON SCHEDULE

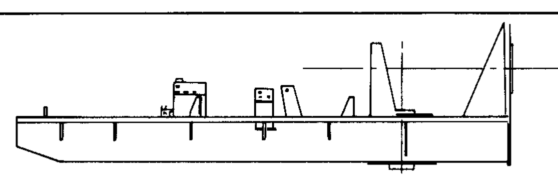
R80-0839-013W

- To allow a ready attachment to the top of the tower
- To provide free yaw capability about the tower axis.

To satisfy these requirements, a basic design of a steel box with an upper surface deck was chosen. Frames were mounted to the upper surface to support doors; bearings in the top and bottom surfaces of the box would permit free yaw rotation about the vertical shaft which mounts to the tower.

Several designs were studied, covering both materials and the method of construction (see Table XV). Although this indicates that fabricated aluminum is the optimum construction, cost consideration and cnanges of design and materials finally led to a welded steel construction. Originally, the Strongback was intended to be constructed of A36 steel, which required surface protection and maintenance of the protection over its 25 year life. Subsequently, COR-TEN steel was chosen

Table XV Strong Back



CRITERIA	CASTING			WELDMENT		BUILD UP		COMPOSITE
	AL	IRON	STEEL	AL	STEEL	AL	STEEL	
25 YR LIFE	10	10	10	10	10	10	10	10
MAINTABILITY	9	10	8	9	8	9	8	9
PRODUCIBILITY	10	9	9	8	8	8	8	2
WEIGHT	8	5	6	9	7	10	7	10
COST	7	10	9	6	8	6	8	2
DEVELOPMENT COST	5	5	5	9	9	10	10	1
TOTAL	49	49	47	51	50	53	51	34

NOTE: SEE NOTE TO TABLE VIII
R80-0839-048W

meant only 6063 specification could be extruded, whereas decreasing the chord to 1 1/2 ft. or less allowed 6061 or 6205 specification to be used. In the T6 heat treat condition, the yield strength of these materials increased from 25 ksi to 35 ksi, a 40% increase.

5.13.5 Blade Strengths

Comparison of two blades of equal wall thickness showed that their bending strengths were proportional to the square of their chord dimensions. Even allowing for the increase in material properties given in the last paragraph, it was found that a 1 ft. chord had insufficient strength at its root. Both the 1 1/2 ft. and 2 ft. chord blades had satisfactory strengths.

5.13.6 Blade Wall Thicknesses

Contact with the aluminum extrusion manufacturers yielded the following information:

- Alcoa. Uniformly reported a minimum wall thickness of 0.188 in. on each of the three blades.

- Taber Metals. Only the 1 1/2 ft. and 1 ft. chord were investigated. Taber Metals report they can obtain a minimum wall thickness of 0.16 inches.

5.13.7 Cost of Extrusion

As shown in Table XIV, in a letter dated August 2, 1978, Alcoa quoted die costs of $19,000; a set-up charge of $2,500 and an extrusion cost of $1.577/lb. for the 1 1/2 ft. chord blade. Taber Metals, in a letter of February 20, 1979, quoted die costs of $19,500; a set up charge of $4,000 in quantities up to 60 and zero for quantities of 300, and extrusion costs of $1.56/lb. for the equivalent blade. Both manufacturers emphasized the long lead times for the production of the extrusions, however, Taber Metals could meet a delivery schedule compatible with our production while other extruders could not.

5.13.8 Conclusions

Based on the preceding review, the 1 1/2 ft. chord extruded blade was selected for the 8 kW WTG, with Taber Metals as the manufacturer.

5.14 TRADE-OFF STUDY FOR STRONGBACK STRUCTURES

The design requirements for the Strongback are:

- To supply a rigid structural platform to mount the WTG system equipment

- To provide support for the nacelle structure, and to allow ready accessibility to the equipment mounted on it for the Strongback material, which had the advantages of not needing surface protection, and also having higher yield strength (50 ksi versus 36 ksi tension). One further advantage of using steel for the Strongback was in obtaining a better weight balance about the vertical shaft. Design of the Strongback was simplified after PDR by maintaining a flat top surface and increasing the width. With this change, the nacelle was kept as low as possible, though the original design was based on keeping it as narrow as possible.

5.15 TRADE-OFF STUDY FOR YAW DAMPER

A major loading term which affects the whole WTG structure is the gyroscopic forces developed from the rotating rotor moving in yaw. In this study, five methods of reducing yaw rate were reviewed to determine if they would produce a cost savings on the WTG. These methods are listed in Table XVI. Of the five methods reviewed, the slip clutch showed the most promise. For this study, only commercially available components were considered; no special designs to match the 8 kW WTG were made.

First, an estimate of the reduction of cost and weight of the structure was determined, then compared against yaw rates. This gave cost and weight targets for the damper to meet. Then the cost and weight of a slip clutch damper was found to achieve these targets (Fig. 16). It was found that the cost of the purchased damping components only (no labor charges or cost additions for mounting the damper between the tower and the nacelle) was significantly above the target cost savings. For this reason, no damper system was added to the WTG.

Although we were not successful in obtaining a suitable yaw damper using available hardware and concepts for this design, it was recognized that there is a need to limit gyroscopic forces, and that further study is worthwhile.

Table XVI Yaw Damper Approach

CRITERIA	EDDY CURRENT	VISCOUS DAMPER	INERTIAL	SLIP CLUTCH	PM GENERATOR
FAIL SAFE	0	6	10	8	7
ENVIRONMENTAL	7	6	10	8	7
DEVELOPMENT COST	10	6	4	8	8
DEVELOPMENT RISK	10	7	6	8	9
PRODUCTION COST	5	9	6	10	7
TOTAL	32	34	36	42	38

NOTE: SEE NOTE TO TABLE VIII
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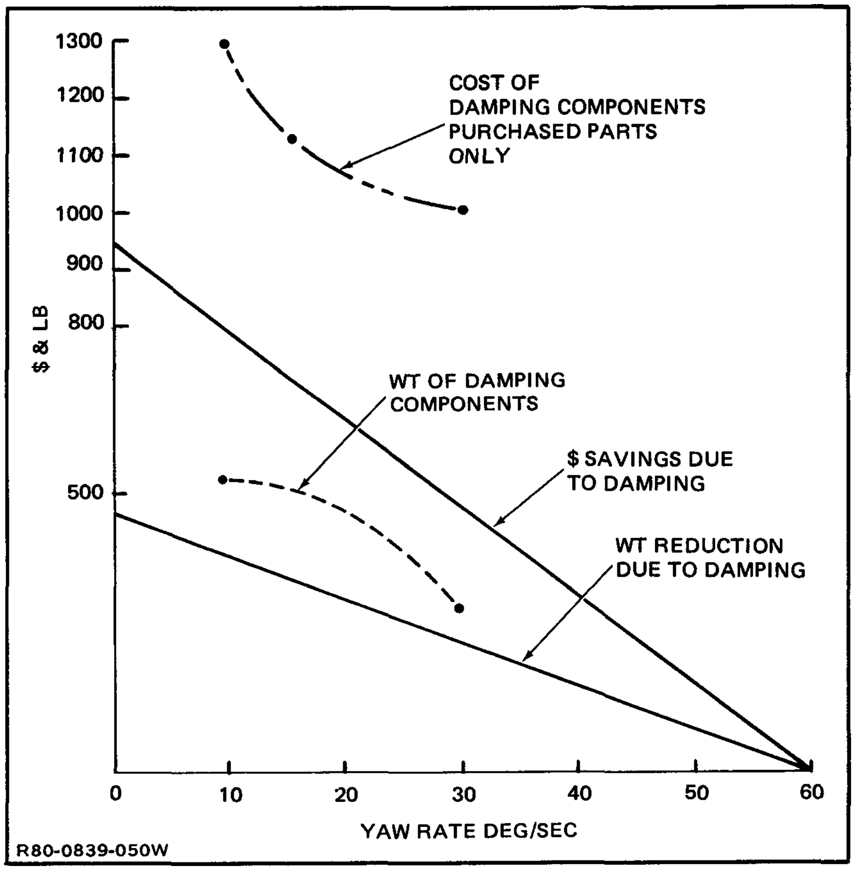


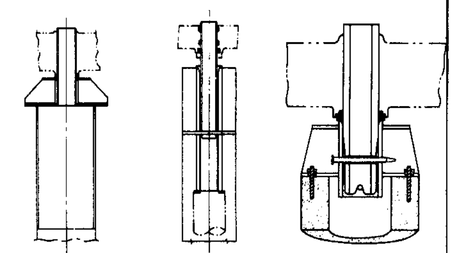
Fig. 16 Yaw Damper Study

5.16 TRADE-OFF STUDY FOR VERTICAL SHAFT/TOWER INTERFACE

The interface of the nacelle vertical shaft with the tower was critical, both for its ease of installation and for its structural transfer of loads from the nacelle to the tower. Optimization of this interface was strongly dependent upon the type of tower structure chosen. Initially a concrete tower was proposed; however, after a trade-off study with a steel trussed tower and steel tube tower, the latter was selected (see Section 5.17).

Table XVII illustrates the three designs considered for a trade-off study for the concrete tower. The external socket design consisted of a steel sleeve which was welded to a pedestal, the sleeve being integrated with the concrete during manufacture of the tower. The internal socket design avoided the need for a pedestal by integrating a long internal steel tube into the tower, but required a long vertical shaft to accept the loads. The third design retained the short shaft of the first, but directly connected with the pre-tensioning cables of the tower. Illustrative of the influence of the tower construction on this interface was the fact that concrete has very low shear and tension strengths, thus requiring the long engagements of the first two designs. In the third design, by directly engaging with the tower pretension members, long engagements were avoided.

Table XVII Vertical Shaft and Tower Interface



CRITERIA	EXTERNAL SOCKET	INTERNAL SOCKET	CABLE-FIXED PEDESTAL
25 YR LIFE	9	8	10
LOAD CAPABILITY	4	7	10
EASE OF INSTALLATION	10	8	10
PRODUCIBILITY	9	9	8
WEIGHT	8	9	10
COST	9	8	9
DEVELOPMENT COST	9	10	8
DEVELOPMENT RISK	6	9	10
ENVIRONMENTAL: TEMP	8	8	10
TOTAL	72	76	85

NOTE: SEE NOTE TO TABLE VIII
R80-0839-051W

Installation of the WTG on the tower would be accomplished by hoisting the full nacelle assembly and inserting the vertical shaft into the pedestal or tower sleeve. The long engagement of the second design required accurate positioning of the suspended nacelle, which was inferior to the installation of the other two designs. In conjuction with the other criteria listed in Table XVII, these considerations led to the selection of the cable-fixed pedestal design for the concrete tower interface.

The tower finally selected for the WTG was a guyed, tubular COR-TEN steel design (see Section 5.17). Two different interface designs were considered; one where the interface between the nacelle and tower was a bolted face connection; the other was similar to the cable-fixed pedestal of Table XVII, except that the method of attaching the pedestal to the tower was modified. The first design appeared to be easy to install, but had disadvantages of being heavier and more costly than the second design. Therefore, the second design was chosen and detailed.

5.17 TRADE-OFF STUDY FOR TOWER CONSTRUCTION

The WTG tower is a major structural and cost item to the Wind Energy program, and a large effort was devoted towards its optimization.

Three forms of construction were considered:

1. A centrifugally spun, pretensioned concrete pole
2. A steel pipe
3. A steel membered, trussed tower.

The first construction was unique to its manufacturer, whereas there were many manufacturers of the second and third constructions.

The following criteria were assessed in selecting a tower.

5.17.1 Strength

The tower has to withstand all factored operating loads, maximum loads occurring during the shut-down condition, gravity loads (including ice-build up), installation, and transportation loads. Repetitive loading experienced over the 25 year design life has to be withstood.

5.17.2 Dynamics

Cyclic loads to the tower occur from:

- Shadowing of blades from wind forces during their passage behind the tower
- Unbalanced centrifugal rotor forces from uneven ice formed on the rotating blades.
- Shedding of vortices in the tower wake (Von Karman Vortex street)
- Galloping of the guys from aerodynamic excitation.

Preferably, these cyclic loads will not be tuned to the natural frequency of the tower. If the loads are tuned, then their magnification must not be sufficient to cause dangerous stresses to the tower. As a precaution, a vibration sensor will be provided to shut down the WTG on the approach of excessive vibration

5.17.3 Cost

The cost factor has a large influence in tower selection. Besides the manufacturing cost, there are costs influenced by the design, such as the size of foundations for the tower, guy "deadmen" installation costs (machinery involved, number of personnel, time, etc.), and transportation costs. With respect to the latter criterion, it is an advantage to have a tower design which can be manufactured locally.

5.17.4 Weight

Tower weight is important in its influence on transportation and ease of erection.

5.17.5 Fixed vs. Pivot Base

When a guyed tower is selected, there is no requirement to supply a fixed base. Therefore, both forms of tower support were considered.

5.17.6 Weather Protection

Three influences of weather on the tower structure were considered in the selection. These were:

- Corrosion from exposure to a moisture laden, salt environment
- Thermal changes causing differential expansions or contractions
- Structural deterioration on exposure to temperatures as low as -50 °C (-58°F).

5.17.7 Minimizing the Ground Area of the Tower Installation

It is desirable to limit the ground area devoted to the wind generator in terms of guy spread, supportive shacks, cable runs, etc.

5.17.8 Lightning Protection

Protection against lightning strike shall be afforded. Since each form of tower construction included substantial steel parts, no impact on tower selection resulted from lightning protection requirements.

5.17.9 Appearance

Where no considerable cost impact was made, a pleasing tower appearance was included in the criteria for selection.

To assess the different tower designs, the following tower manufacturers were contacted:

1. Bayshore Concrete Products, manufacturer of pretensioned, centrifugally spun concrete towers
2. Unarco-Rohn, manufacturer of steel membered, trussed towers
3. Kline Towers, manufacturer of steel membered, trussed towers
4. Tubesales, Ryerson Steel, and Thypin Steel Company, three companies which produce steel pipes.

To simplify comparison, each of the first three manufacturers were sent a combined loading condition to obtain a "first-cut" tower design. To satisfy dynamic requirements, the manufacturers were also asked to satisfy a minimum tower stiffness. In-house analysis was conducted on the steel pipe tower construction.

It was found that the two steel trussed towers were not cost competitive with either the concrete or steel pipe designs, so they were eliminated. Further design features were then compared as incorporated in the steel and concrete towers, as shown in Table XVIII. These are:

● Guy wire spread. In an effort to reduce the ground area devoted to the tower, the diametral distance between guy anchor points was reviewed for 30 and 45 ft., and compared with the original spread of 80 ft. Generally, the reduction in structural efficiency had a significant cost impact, and the original guy spread of 80 ft. was retained.

● The flexible cable guy with povot base was compared with the more rigid strap guy with fixed base configuration. Again, the first configuration introduced a cost penalty, and the second configuration was chosen.

● Guy redundancy. To provide a fail-safe mode in the tower design, duplication of guys was considered. Although there was a cost impact, some form of redundancy was strongly recommended. It was subsequently decided to design the tower for free-standing capability.

The previous trade-offs were separately conducted for a machine with and without unbalanced ice on the rotor.

Table XVIII Tower Cost and Weight Summary

LOADING CASE	MATERIAL	DIA	GUY WIRE SPREAD	CABLE GUYS & PIVOT BASE	STRAP GUYS & FIXED BASE	GUY REDUNDANCY	COST	WEIGHT
WITH ICE IMB	CONCRETE	16.8 IN.	30	YES	–	NO	3466	14951
				–	YES	YES	2020	13165
						NO	1893	13112
			45	YES	–	NO	3052	14491
			80	–	YES	YES	2003	13087
						NO	1882	13060
	GALVANIZED STEEL PIPE *	18 IN.	30	YES	–	NO	3637	7887
				–	YES	YES	2464	5497
						NO	2356	5444
			45	YES	–	NO	3229	7006
			80	–	YES	YES	2272	5319
						NO	2206	5292
NO ICE IMB	CONCRETE	15.2 IN.	30	–	YES	YES	1920	11615
						NO	1793	11562
			80	–	YES	YES	1902	11537
						NO	1782	11510
	GALVANIZED STEEL PIPE	16 IN.	30	–	YES	YES	2040	4389
						NO	2003	4336
			80	–	YES	YES	1940	4211
						NO	1874	4184

R80-0839-052W

From Table XVIII it can be seen that the concrete tower was 10-20% more cost effective than the galvanized steel pipe. However, the galvanizing contributed significantly to the cost of the steel pipe ($350-500). By using COR-TEN steel, which had good strength properties and needed no surface protection, the two forms of tower construction were equal in cost. However, there were these disadvantages to the concrete tower:

1. A unique manufacturer, located in W. Virginia

2. A heavy tower would impose penalties in terms of the transportation costs and the size of ground handling equipment required

3. Present experience indicates thermal cracking can develop, as seen on a WS-25 tower.

For these reasons a COR-TEN steel tower was selected.

6.0 SUPPORTING ANALYSES

6.1 ROTOR PERFORMANCE, POWER AND ENERGY PRODUCTION

The first task in evaluating rotor performance was to satisfy the requirements of producing 8 kW power output in a 20 mph wind at sea level. The blade section selected for this was that which was also selected, after an extensive survey, for the Grumman Windstream 25. It was found that for a three bladed rotor of blade chord 2 ft. and rotor diameter of 28 ft., this requirement was satisfied. System efficiencies were included in this determination.

Two improvements to power production were made. First, retaining the 2 ft. chord, extruded 6063-T6 aluminum blade, the rotor diameter was increased to 32 ft., based upon the limiting strength at the root of the blade. Further improvement in power production was made by going to a 1½ ft. chord, extruded 6061-T6 aluminum blade, with a rotor diameter of 33.5 ft., again limited by blade root strength. The latter rotor would produce 10.8 kW in a 20 mph wind.

Prediction of power coefficients versus tip speed ratios for a range of blade settings are shown in Fig. 12. From these curves an optimum blade setting of 4° was seen to give a maximum power coefficient of 0.38 at a tip speed ratio of 5.25. Power conversion on the WTG was through an Induction Generator, running at approximately constant speed (2% slip at 100% load). Therefore, when a gearbox was selected, there was only one wind speed at which the maximum power coefficient developed (the reference wind speed). At other wind speeds, the tip speed ratio was inversely proportional to this reference wind speed. Therefore, with increasing wind speed, the tip speed ratio dropped and the corresponding drop in power coefficient limited the WTG from extracting excessive power. Although a gearbox could be selected which would give maximum power for a 20 mph wind, another reference wind speed would give optimum annual energy for a given wind regime. This is discussed further in Section 5.3.

Of interest in determining the blade and rotor forces were the thrust coefficient plots. Having selected a 4° blade setting, a plot of thrust coefficient versus tip speed ratio was derived (see Fig. 17).

From the plots of Cp and Ct vs. λ, the variation of power and rotor thrust versus wind speed were derived for the WTG set to a reference wind of 17 mph (Figs. 18 and 19).

In assessing the economic viability of the WTG, two approaches were available: that of determining a given power in a given wind speed, and

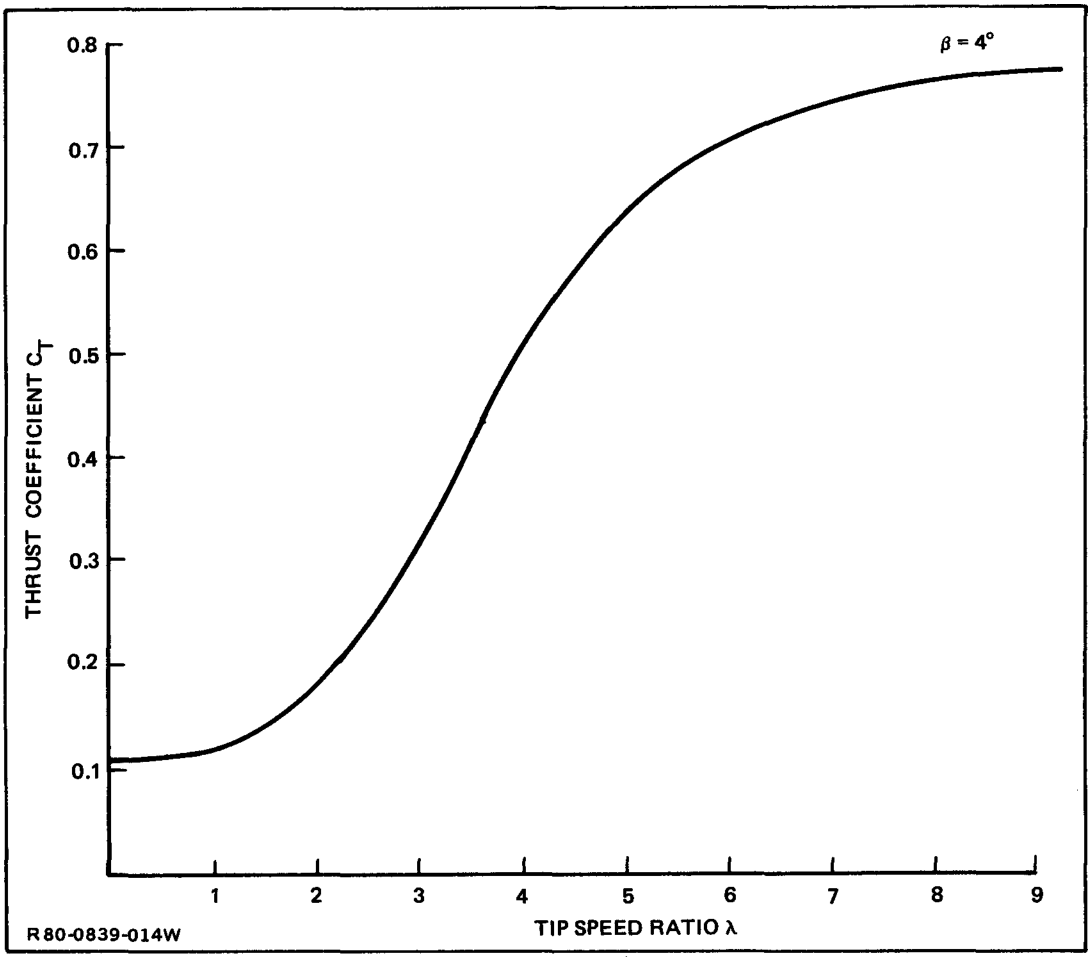


Fig. 17 Rotor Performance Thrust Coefficient vs Tip Speed Ratio for Blade Angle Setting of 4°

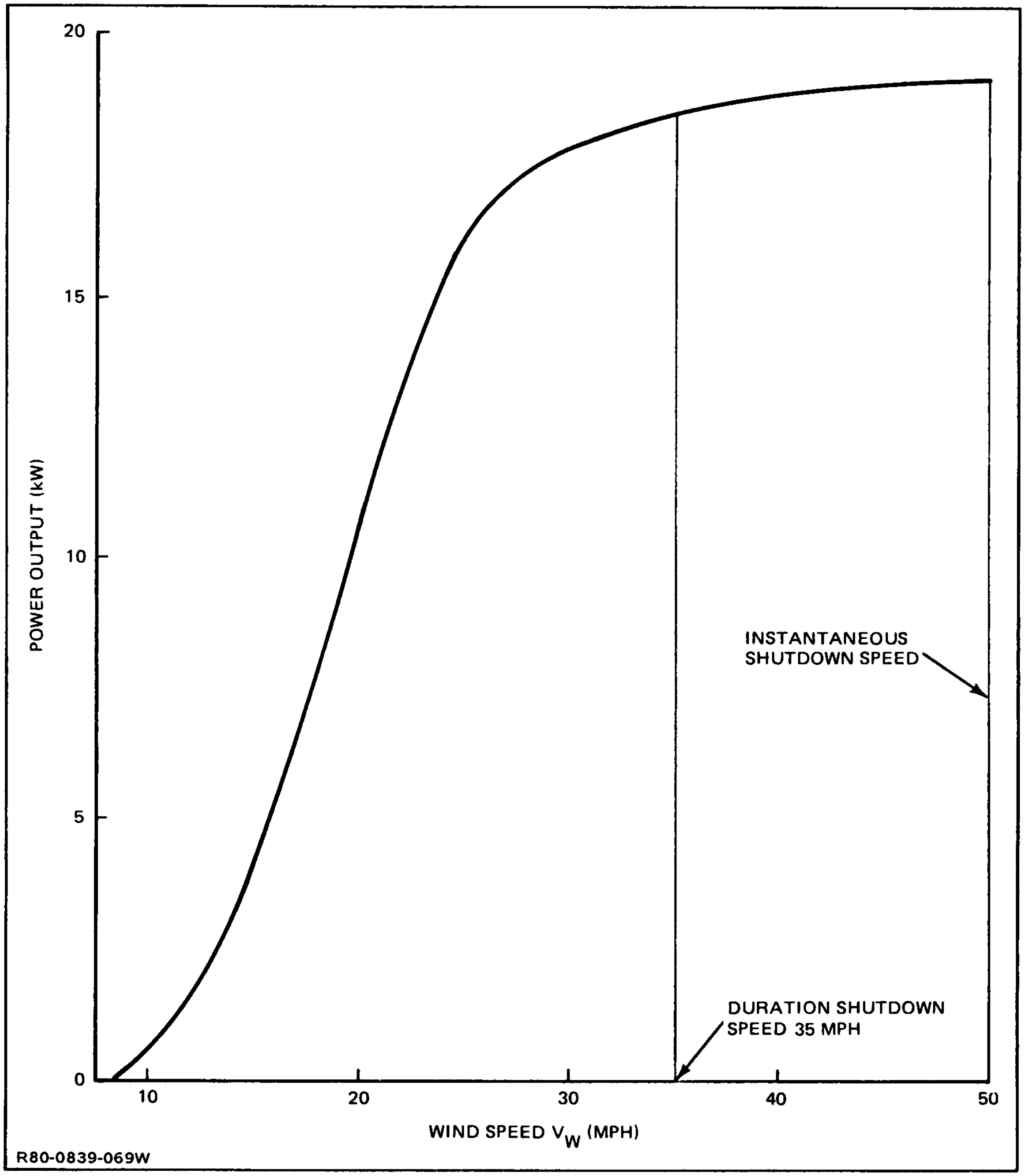


Fig. 18 Induction Generator Output vs Wind Speed (V_{REF} = 16.63 MPH)

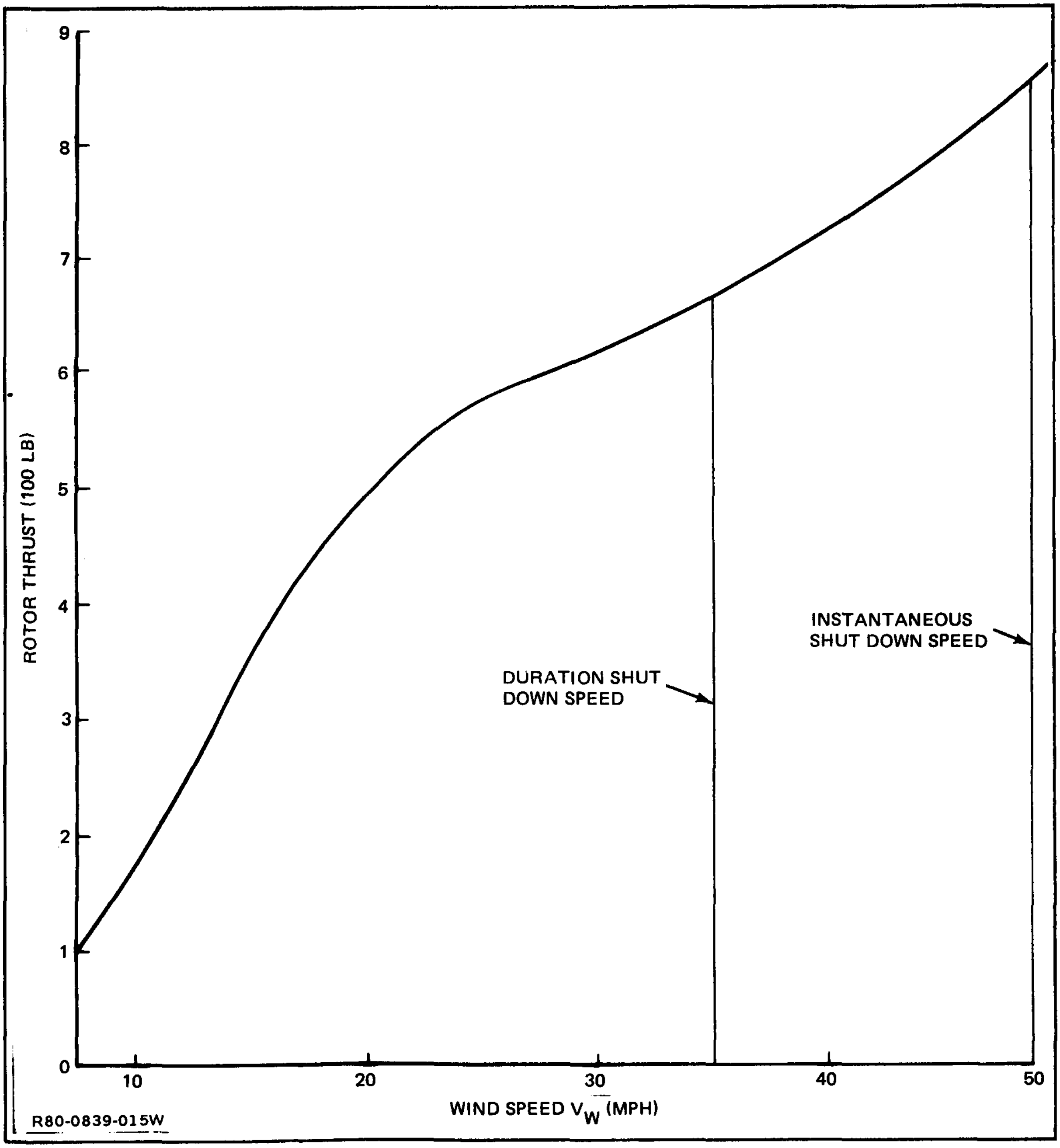


Fig. 19 Rotor Thrust vs. Wind Speed (V_{REF} = 17 MPH)

that of obtaining a given energy output from a given annual average wind speed. In each approach, the power or the energy was optimized. Where the electrical output is through an alternator, each of the two approaches is equally valid. This was because the unit was working at constant maximum power coefficient up to the rated wind speed, after which constant power is extracted to the cut-out wind speed. In other words, having selected maximum power for a design wind speed, the power versus wind speed is fixed over the total wind speed range. The power versus wind speed characteristic of an Induction Generator system would be different. At only one wind speed (the reference wind speed) was the power coefficient maximum; at all other wind speeds it was less than maximum. For this reason it was very important to select the gearbox which would give a reference wind speed at which annual energy production became a maximum. This did not necessarily produce the maximum power at a stated wind speed; in fact, it would only do so if the stated wind speed was also the reference wind speed. As shown in Fig. 11, a reference wind speed of approximately 17 mph would give optimum annual energy for a wind range of annual average wind speeds.

In satisfying the initial requirement of producing at least 8 kW in a 20 mph wind, the Grumman WTG was predicted to produce 11.2 kW for a 20 mph reference wind speed, and 10.8 kW in a 17 mph reference wind speed. However, in a 13 mph annual average wind speed, the energy production for the 17 mph reference wind was 126.2% that of the 20 mph reference wind, with larger improvements for higher annual average wind speeds (see Fig. 11). Estimates of the annual energy output versus annual average wind speed are shown in Fig. 20. The Grumman WTG was optimized on the basis of annual energy output.

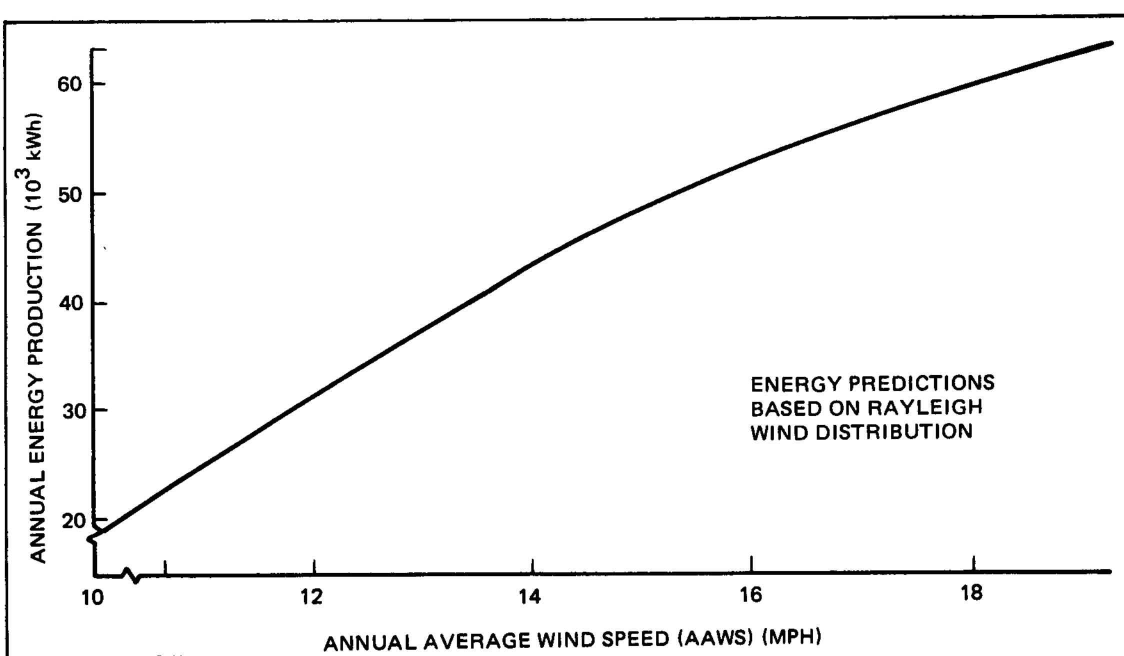


Fig. 20 Annual Energy Production vs. AAWS (V_{REF} = 16.63 MPH)

6.2 PRODUCTION COSTING

The costing of the 8 kW Wind Turbine Generator was based on a production rate of 1,000 units per year. For the purposes of estimating tool and die costs, it was assumed that this production rate would continue for a minimum of five years. The major cost categories considered in determining the cost per unit are as follows:

1. Purchased parts
2. Raw materials
3. Tool and die
4. Fabrication (labor)
5. Final assembly (labor).

6.2.1 Purchased Parts Costs

Purchased parts costs were primarily obtained from vendor supplied estimates and quotes. In a few instances, where a quote was not available for a specific size part, a GESI estimate was made by pro-rating (by weight) the quote for a similar part.

6.2.2 Raw Material Costs

Raw material costs for mill stock were obtained either from vendor quotes, or GESI material cost data. The cost of castings was estimated by prorating available vendor estimates for castings of representative complexity.

6.2.3 Tool and Die Costs

Tool and die costs associated with castings and stampings were also estimated by pro-rating a selected sample of vendor estimates representing varying degrees of complexity. Since these costs had very little impact on the per unit cost, the estimates were made on the conservative side.

6.2.4 Fabrication Costs

The cost of fabrication for major components was obtained from vendor quotes. In the absence of specific quotes, estimates were made by prorating available cost data for similar parts. The majority of the detail parts costs were determined by first identifying the fabrication operations involved in the manufacturing of each part, and the intensity of metal working (volume removed, area ground length cut or welded) associated with each operation. Subsequently, a cost was assigned to each operation based on prices published by commercial machining shops for similar services. This estimating method was compared to hard vendor quotes for a number of selected parts, and was found to agree very closely. Gross estimating of system or sub-system manufacturing costs was avoided in every instance because of the lack of sufficient statistical history for this type of machine.

6.2.5 Final Assembly (Labor) Costs

The final assembly labor costs were determined by estimating the time required for each assembly operation. The time estimates were based on previous GESI experience with similar assembly operations. The labor rate assigned to each operation time sequence was selected on the basis of the skill required and the industry rate standard for that skill.

6.2.6 Fixed Costs

The fixed costs were established by estimating the known indirect costs of a 60,000 sq. ft factory and applying them to the labor rate as overhead charges. The indirect costs were allocated as follows:

Factory ($600,000 ÷ 25 years)	$ 24,000
Capital Equipment ($900,000 ÷ 8 years).	113,000
Indirect Labor: 3 @ $30,000	90,000
5 @ 20,000	100,000
Non Work Fringe Benefits.	632,000
Freight In	100,000
Indirect Supplies and Miscellaneous Cost	155,000
Utilities, Tax, Security	120,000

Utilizing this cost center approach, the costing would be as follows:

Direct Labor	1,000 units x 205 hrs x $5.52	=	$1,131,600
Purchased Parts	1,000 units x 4280	=	4,280,000
Raw Material	1,000 units x 1958	=	1,958,000
Indirect	(Equals 117.9% of Direct Labor)	=	1,334,000
	1,000 Units Total Cost	=	$8,703,600

Table XIX gives the breakdown of the cost of the 8 kW WTG at a production rate of 1,000 units per year.

6.3 STRUCTURAL ANALYSIS

To ensure structural integrity, analyses were conducted of each structural component, purchased parts, the tower, and its attachment to the foundation.

The analysis was conducted along the following lines:

- Describe all the loading conditions to which the WTG will be subjected

Table XIX Production Cost Summary (1978 Dollars)

ASSEMBLIES COST APPROACH				
BASE LINE EQUIPMENT	PURCHASED PARTS	RAW MATERIAL AND/OR CASTINGS	FAB & ASSY LABOR	SUB TOTALS
1 LOW SPEED DRIVE	1063	242	174	1479
2 HIGH SPEED DRIVE	288	–	6	289
3 ROTOR HUB ASSY	355	339	530	1224
4 BLADES	884	12	98	994
5 VERTICAL SHAFT/PEDESTAL	36	154	186	376
6 STRONG BACK	–	137	400	537
7 COWLING ASSY	–	86	143	229
8 PITCH CONTROL MECH	8	58	120	186
9 ELECTRICAL EQUIP	1601	–	136	1737
10 MISC HARDWARE	50	–	50	100
11 PAINT	–	50	60	110
12 FINAL ASSY	–	–	184	184
13 TOWER	–	880	379	1259
	4280	1958	2466	8704
G & A & PROFIT (26.5%)				2306
SELLING PRICE				11,010

R80-0839-037W

- Combine the loading conditions where applicable to obtain loading bases
- Determine maximum values of load applied to the WTG system for each loading case
- Resolve these loads at each major station on the WTG
- Select suitable safety factors to be applied to the loads
- Select all critical areas within the WTG and determine stress levels
- Assure positive margins of safety (M.S.) where

$$M.S. = \frac{\text{yield strength of material}}{\text{section stress level}} - 1$$

- Where M.S. is significantly above zero, investigate possible material reduction as a means to cost savings.

The ten major loading conditions to which the WTG is subjected are listed in Table V and more fully described in Section 4.3. Also shown in Table V are the eight loading cases analysed, each case being the combination of conditions marked with an X.

For ready reference, and to maintain a constant description, the load and geometry convention shown in Fig. 21 was established. Where necessary, local conventions are used (such as the blades). Loads developed from the conditions listed in Table V are presented in Table XX. Illustrative of the combination of design conditions to form design cases is Table XXI. This shows the maximum loading on the blade (from case 4) at four critical stations, and allows interpolation of loads at intermediate stations.

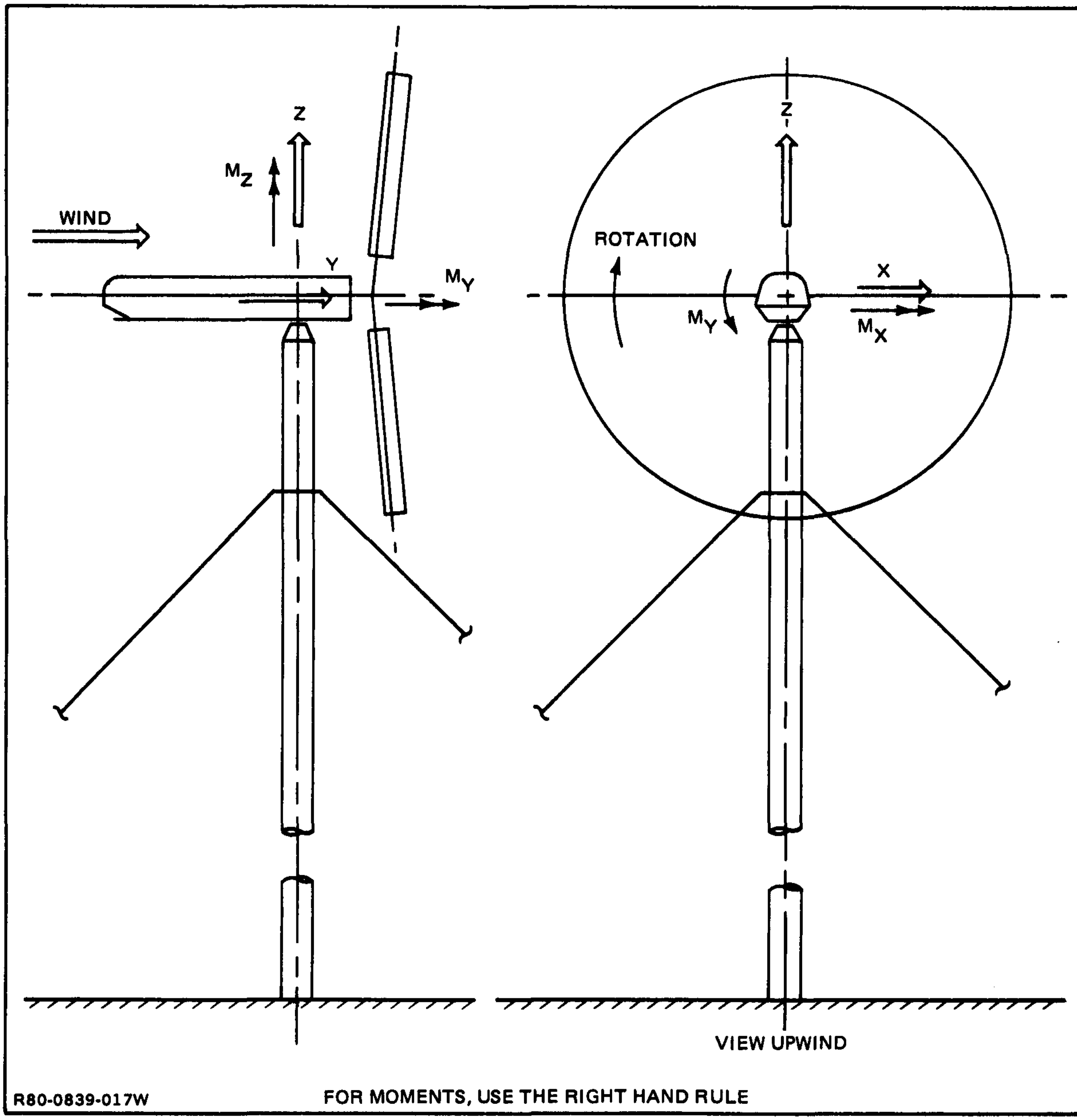


Fig. 21 Load and Geometry Conventions

Table XX Design Condition Loads

CONDITION	DESIGN LOADS
CUT-OUT WIND WITH GUST	BLADE THRUST = 383 LBS (F_y) BLADE TORQUE (AT HUB) = -4,804 IN. LBS (M_y) TOWER LOAD = 6 LBS/FT (F_y)
TOWER SHADOW	BLADE IN SHADOW THRUST = 0 (F_y)
GYROSCOPIC	ROTOR MOMENT AT HUB = ± 176,068 IN. LB (M_x)
GRAVITY	BLADE = 188 LBS (F_z) NACELLE WITH ROTOR = 2,589 LBS (F_z) TOWER = 3,400 LBS (F_z)
CENTRIFUGAL FORCES	AXIAL FORCE = 3,330 LBS (F_x, F_z) CONING MOMENT = 22,986 IN. (M_x, M_z) AT 73 RPM
UNBALANCED ICE LOADS (10 LBS AT TIP)	AXIAL FORCE = 640 LBS (F_x, F_z) CONING MOMENT = 6,934 IN. LBS (M_x, M_z) AT 73 RPM (INCLUDES MAGNIFICATION FACTOR = 2)
MAX. WIND SPEED OF 165 MPH	TOWER LOAD = 65 LBS/FT (F_y)
UNIFORM 2½ IN. LAYER OF ICE	WEIGHT = 11.67 LBS/FT^2 (F_z)

NOTE: THE DESIGN LOADS DO NOT INCLUDE SAFETY FACTOR.

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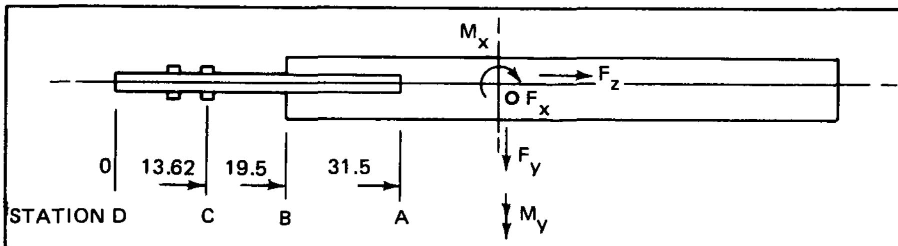
Table XXI Rotor Blade Loads – Case 4 (See Table V)



STA	CONDITION	F_x	F_y	F_z	M_x	M_y
A	CUT-OUT WIND WITH GUST	0	32.1	–	3,791	0
	GYROSCOPIC FORCES	867.0	–	–	–	91,147
	GRAVITY FORCES	–	–	175.8	–	891
	CENTRIFUGAL FORCES	–	–	3,279.0	–	20,612
	UNBALANCED ICE LOADS	–	–	640.4	–	6,472
	TOTAL	867	32	4,095	3,791	119,122
B	CUT-OUT WIND WITH GUST	0	32.1	–	4,176	0
	GYROSCOPIC FORCES	911.0	–	–	–	101,814
	GRAVITY FORCES	–	–	216.4	–	1,031
	CENTRIGUGAL FORCES	–	–	3445.0	–	22,986
	UNBALANCED ICE LOADS	–	–	640.4	–	6,934
	TOTAL	911	32	4,302	4,176	132,765
C	CUT-OUT WIND WITH GUST	0	32.1	–	4,365	0
	GYROSCOPIC FORCES	920.7	–	–	–	107,240
	GRAVITY FORCES	–	–	229.1	–	1,109
	CENTRIFUGAL FORCES	–	–	3,482.0	–	24,243
	UNBALANCED ICE LOADS	–	–	640.4	–	7,160
	TOTAL	921	32	4,352	4,365	139,752
D	CUT-OUT WIND WITH GUST	0	32.1	–	4,804	0
	GYROSCOPIC FORCES	930	–	–	–	119,790
	GRAVITY FORCES	–	–	257.2	–	1,309
	CENTRIFUGAL FORCES	–	–	3,517.0	–	27,053
	UNBALANCED ICE LOADS	–	–	640.4	–	7,685
	TOTAL	930	32	4,415	4,804	155,837

R80-0839-019W

Presently, there is no industrywide agreement on safety factors to be used in analysing WTG structures. For the design of the 8 kW WTG, Grumman used a factor of 2.0 times maximum applied loads, and compared the developed stresses with the yield strength of a given structure. This is judged to give confidence in the structural integrity of the WTG without involving too high material costs.

Table XXII is a summary of the critical sections on the WTG systems, with the drawing number, design case, stress level, material, and margin of safety for each section.

The Grumman approach to dynamic loading is to avoid dynamic excitations. Section 6.4 gives the details of the dynamics analysis. Table XXIII lists the natural frequencies up to and above 3 times the operating speed of the rotor, i.e., 3 x 1.25 Hz. As explained in Section 6.4, only mode 7 appeared to give a critical frequency (3.899 Hz against 3 x 1.25 Hz nominal

Table XXII Critical Sections with Stresses and Margins of Safety

SECTION DESCRIPTION	DRAWING	CASE	YIELD STRESS (PSI)	MATERIAL	M.S.
OUTBOARD END OF BLADE ENGAGED WITH STUB SHAFT	C11B06021	4	32,986	6061-T6	0.06
INBOARD BOLT HOLE STATION OF STUB SHAFT IN ENGAGEMENT WITH BLADE	C11B06128	4	95,763	15-5 PH (H1025)	0.51
HUB I - - BEAM	C11B06008	4	49,506	4130 NORMALIZED	0.41
HUB: WELDING OF I BEAM TO BARREL HOUSING	C11B06008	4	29,445		0.01
BENDING OF LOW SPEED SHAFT AT REAR BEARING	C11B06016	4	112,200	15-5 PH (H1025)	0.29
BENDING OF STRONGBACK AT VERTICAL SHAFT STATION	C11B06006	4	39,670	COR-TEN	0.26
BENDING OF FORWARD BEARING SUPPORT RIB	C11B06006	4	36,813	COR-TEN	0.35
VERTICAL SHAFT, AT PIN-HOLE (SCF = 3.47)	C11B06031	4	142,200	(H1025)	0.02
"RING" LOADING OF TOWER PEDESTAL CENTRAL TUBE	C11B06136	4	46,290	COR-TEN	0.08
BENDING ON TOWER AT BASE WITH GUYS INEFFECTIVE (SF = 1.5)	C11B06137	5	46,707	COR-TEN	0.05
GUY ROD	C11B06137	5	41,855	COR-TEN	0.19
ANCHOR ROD WELD, (SCF = 2.7)	C11B06032	5	27,200	AWS A5:5 E80XX-G	0
BLADE STUB CRANK	C11B06119	1-4	49,491	4130 (NORMALIZED)	0.41
BLADE STUB LINK (LUG TENSION)	C11B06022	1-4	15,593	4130 (NORMALIZED)	0.21
BENDING OF PITCH CONTROL ACTUATOR CRANK	C11B06026	1-4	54,240	1020	0.17
BENDING OF AUXILIARY ACTUATOR CRANK BRACKET	C11B06013	1-4	49,084	COR-TEN	0.01
BENDING OF AUXILIARY ACTUATOR CRANK PIVOT PIN	C11B06013	1-4	173,840	NAS 464-1 2	0.03
BENDING OF PITCH CONTROL TORQUE ARM	C11B06020	1-4	65,894	4130 NORMALIZED	0.06

NOTE: CALCULATION OF STRESS INCLUDES THE SAFETY FACTOR OF 2.0 ON FIELD

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operating). Further work indicated that this excitation (using conservative assumptions) produced low stress excursions in the blade and tower, and its influence was small.

Table XXIII WTG System Natural Frequencies

MODE NO.	DESCRIPTION	BLADE COND 1 f (Hz)	BLADE COND 2 f (Hz)	BLADE COND 3 f (Hz)
1	WTG RIGID YAW	0.100	0.100	0.100
2	ROTOR/NACELLE ROLL	1.537	1.537	1.537
3	TOWER FORE-AFT BENDING	2.300	2.300	2.300
4	TOWER LATERAL BENDING	2.770	2.770	2.770
5	TOWER FORE-AFT BENDING (NACELLE PITCH)	2.801	2.801	2.801
6	NACELLE YAW-BLADE NORMAL BENDING	3.820	3.821	3.820
7	BLADE NORMAL BENDING TOWER FORE-AFT	3.899	3.900	3.899
8	NACELLE/ROTOR PITCH	6.150	6.152	6.150

NOTE: THE FREQUENCIES SHOWN DO NOT INCLUDE CENTRIFUGAL STIFFENING EFFECTS

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Two ground handling conditions were considered: hoisting of the nacelle with rotor (but no tower), and hoisting of tower. The hoist beam was analysed for a factored load of 15,000 lbs. (5 x 3000 lbs.); the associated loads to the nacelle were analysed with a safety factor of 4.0. The tower was similarly analysed for a factored load of 17,500 lbs. (5 x 3,500 lbs.) for support either at its center or at each end.

During the design phase of the WTG system, careful attention was paid to avoiding features which would limit the life of the system. These include rapid change of section, small radii, holes positioned at highly stressed stations, threaded sections subjected to high stress excursions, and coarse machined surfaces in areas of high stress. In three locations, it was not found practical to avoid these stress raisers. The locations were: a) the low speed shaft key-way, b) the vertical shaft at the location pin, and c) the guy rod eye-ends. To avoid extensive fatigue analysis, a simplified method of determining fatigue strength was taken. This was to retain the safety factor of 2.0 applied to the maximum stress (f max); determine the stress concentration factor (SCF) for the section from the literature, and to verify that $2.0 \times f_{max} \times SCF < F_y$, the yield strength of the material.

6.4 DYNAMIC ANALYSIS

The Wind Turbine Generator and its tower will be subjected to a variety of cyclic loads throughout its life. An analysis of the whole structure was conducted to ensure that no excessive vibrational loads developed either from the rotating machinery or from wind loadings.

Cyclic loads could occur in the following conditions:

- Shadowing of the blade aerodynamic loading during its passage behind the tower
- Unbalanced centrifugal rotor forces from uneven ice formation on the rotating blades
- Shedding of vortices in the tower wake (Von Karman Vortex Street)
- Galloping of the guys from aerodynamic excitation
- Rotation of the high speed shaft at its critical speed.

Separate analyses were conducted on the total WTG installed on the guyed tower, on the blades, on the guys, and on the high speed shaft.

In analyzing the total WTG installed on the guyed tower, use was made of the NASTRAN (NASA Structural Analysis) program. The tower was divided into 18 grid points, joining 17 bar elements (see Fig. 22). Each of the four guys was represented by a rod element capable of withstanding tension or compression. The vertical shaft and the rotor shaft were each represented by elements whose flexibilities had already been determined. The vertical shaft yaw flexibility was given a high value to represent its freedom in yaw. An aribitrary value was chosen which would result in a yaw frequency of 0.1 Hz. Each of the three blades was represented by a member having 9 grid points connecting 8 bar elements. Coning of the blades was included in the model. In the analysis, the number 1 blade was separately investigated in the 12 o'clock, 3 o'clock, and 6 o'clock positions. No significant differences in natural frequencies were observed in any of the three positions (see Table XXIII). Both mode shapes and vectors at the grid points were determined for the tower and WTG. The mode shape for mode 7 is shown in Fig. 23. Responses (accelerations, velocities and displacements) at all grid points (except those which were fixed) were obtained. Structural loads (shears, bending moments and torques) on all bar elements, and axial loads on all rods were obtained for unit force inputs in the x, y, and z axes, and for unit moments about the x and z axes at the rotor hub.

The operational speed of the Induction Generator powered WTG system is limited to a range very close to 75 rpm at the rotor. Three of the modes shown in Table XXII are close to this speed or a multiple of three times this speed: modes 2, 6, and 7. In mode 2 (rotor/nacelle roll), the natural frequency of 1.537 Hz is 23% higher than the operating speed of 1.25 Hz. Only one form of excitation is visualized in this mode: that of an unbalanced ice load on the rotor producing a sinusoidal torque about the hub from its own gravity. However, this is a small variation (±10 lb. x 33.5/2 ft.), and in conjunction with the 23% separation with the natural frequency, is not considered serious.

Mode 6, which is normal blade bending occurring from nacelle yaw oscillation, has a natural frequency of 3.820 Hz. Although this is close to

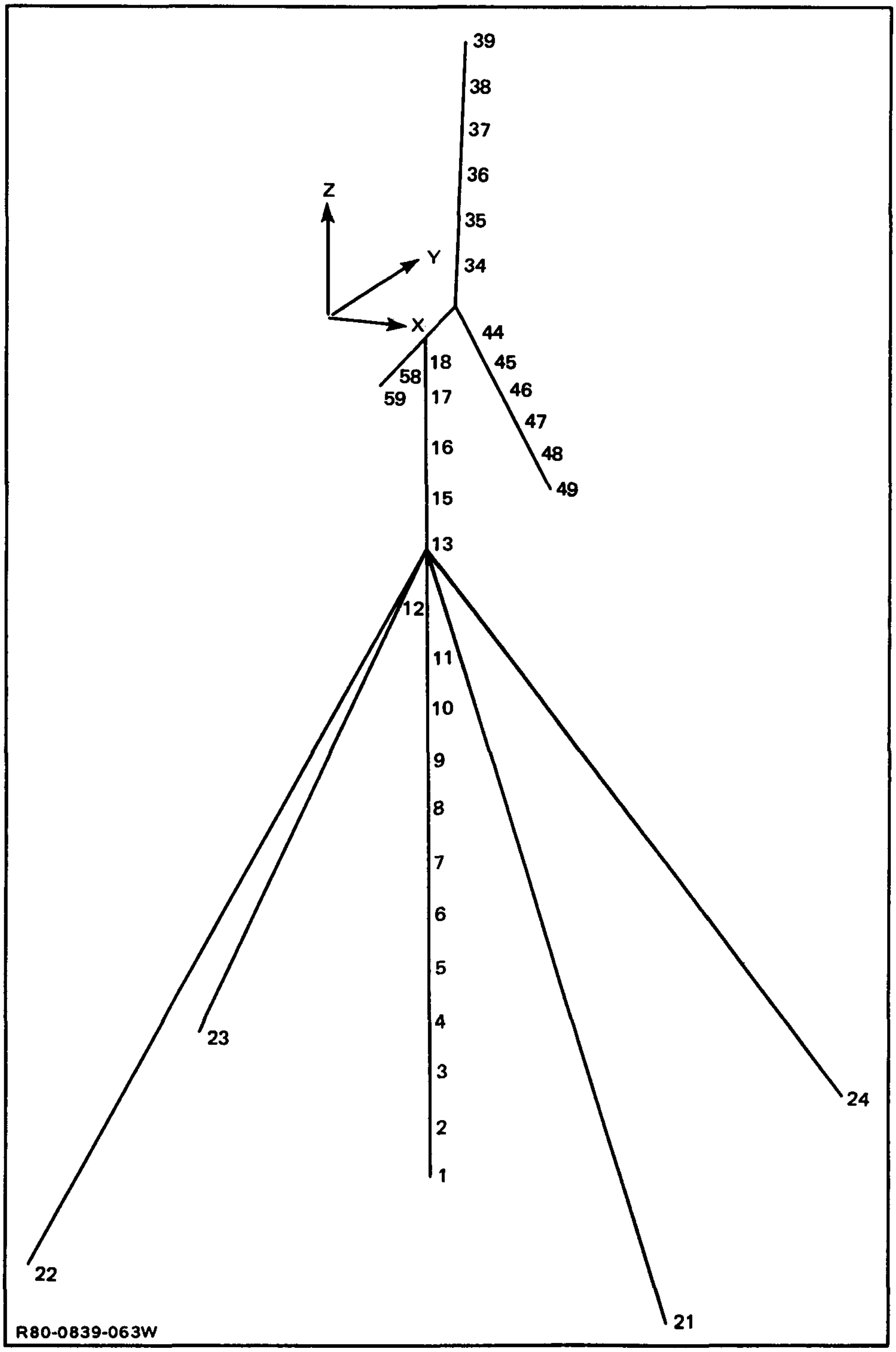


Fig. 22 Vibration Analysis of 8 kW WTG on Guyed 55 Ft. COR-TEN Tower Blade Condition 1 – No. 1 Blade at 12 O'Clock Undeformed Shape

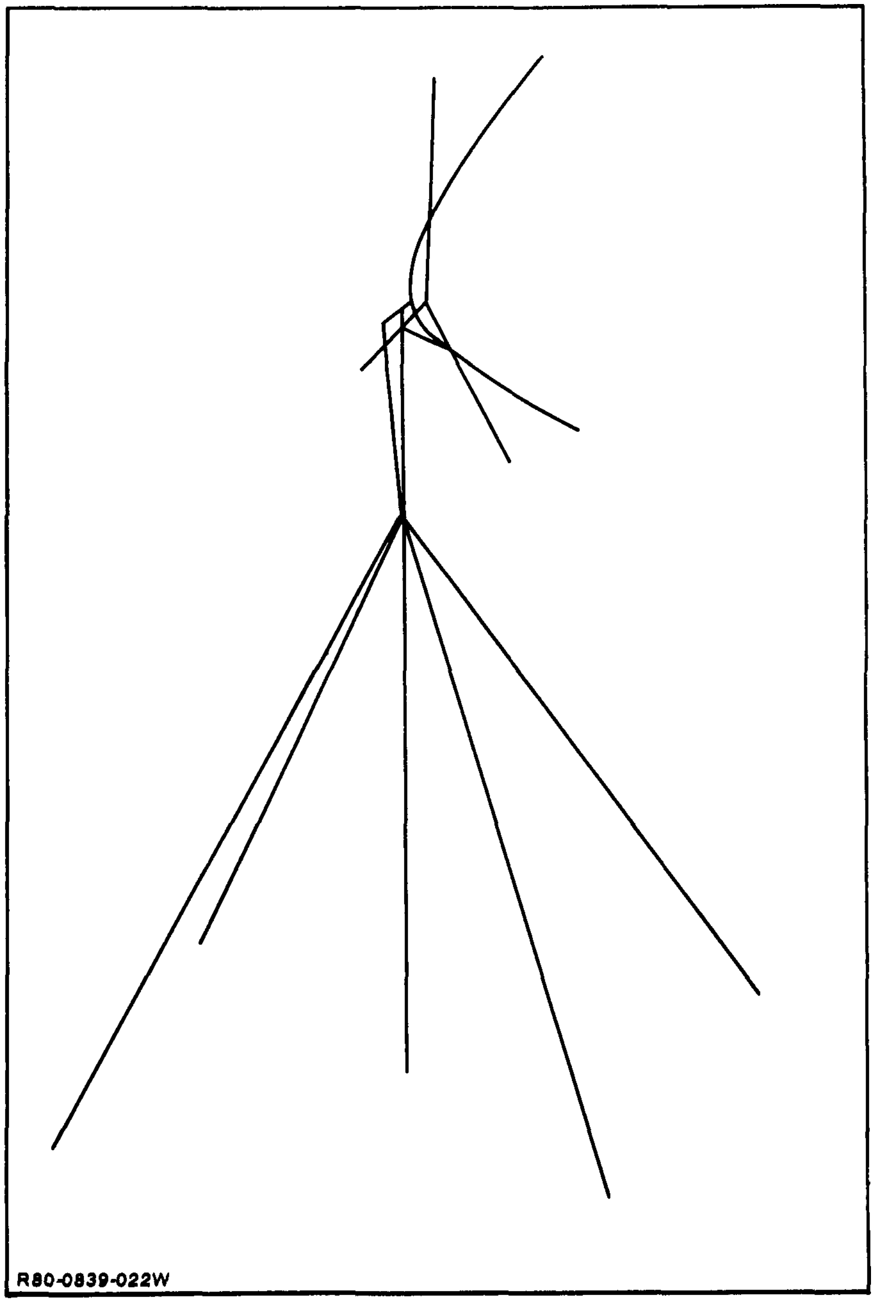


Fig. 23 Mode Shape for Mode 7 – Number 1 Blade at 12 O'Clock, f_n = 3.899 Hz

three times the operating speed of 1.25 Hz, no cyclic loads are foreseen which can excite this mode. Again, this is not considered serious.

Mode 7, which is normal blade bending associated with tower fore-aft bending, has a natural frequency of 3.899 Hz. This is close to three times the operating speed of 1.25 Hz (within 4%). Excitation of this mode could be expected from passage of each blade through the tower wind shadow in each revolution. As a given blade passes through the shadow its aerodynamic thrust force is reduced, and as the succeeding two blades pass through the tower shadow, the first blade responds to changes of force on the succeeding two blades. On a given blade there will be three load changes per rotation of the rotor. A more detailed analysis of this condition was performed. The following, conservative assumptions were made just for this analysis:

1. The operating speed of the rotor exactly matched the natural frequency of 78 rpm

2. The aerodynamic thrust force on a blade reduced to zero in the shadow

3. The same blade fully responded to the following two blades in that its thrust force reduced to zero on the passage of each of the two blades through the tower shadow

4. The width of the tower shadow is 30° of rotation, following a (1-COSθ) function of thrust loss (see Fig. 24).

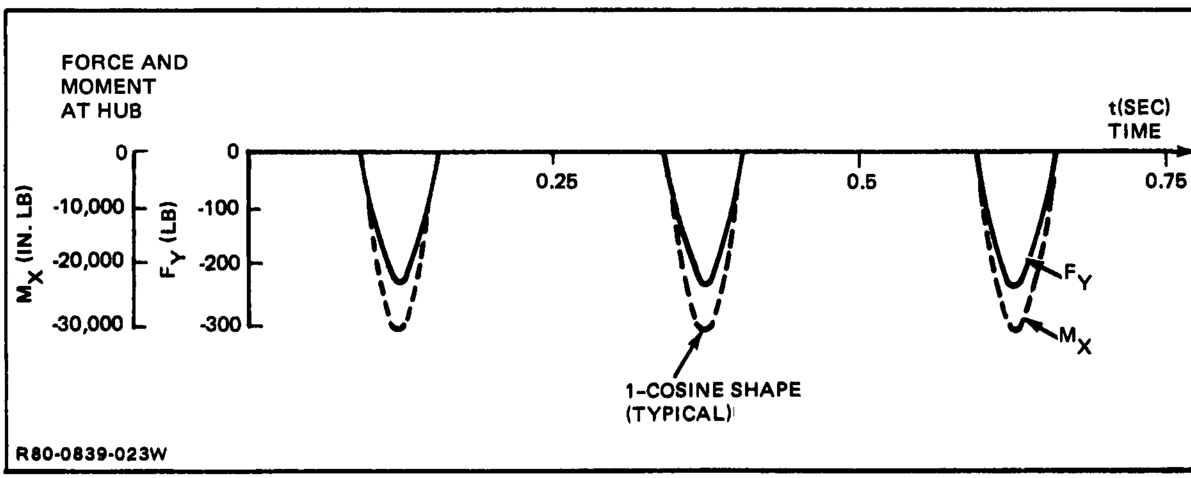


Fig. 24 Tower Shadow Forcing Function at 78 RPM

The preceding assumptions produced forcing functions of 0 to -227 to 0 lb. thrust, and 0 to -30,400 to 0.in. lb. moment at the hub, occurring every 0.256 sec. The forcing cycle was divided into 36 increments, which repeated every 0.256 sec., as shown in Fig. 24. Thirty-two of these cycles were included in the analysis, covering a period of 8 sec. Figure 25 is an output of this analysis, and shows response to the blades first bending mode frequency of 3.899 Hz. As seen in Fig. 25, the maximum blade tip acceleration to loads at the sec^2, or 2.6 g. Reduction of this acceleration to loads at the blade stub results in a stress cycle of 6.6% of its yield strength. Similar reduction on the tower results in a stress cycle of 3.2% of its yield strength. Similar reduction on the tower results in a stress cycle of 3.2% of its yield strength. Because of the four conservative assumptions listed above, even these low stress cycles are higher than expected, and tower shadowing is not considered serious.

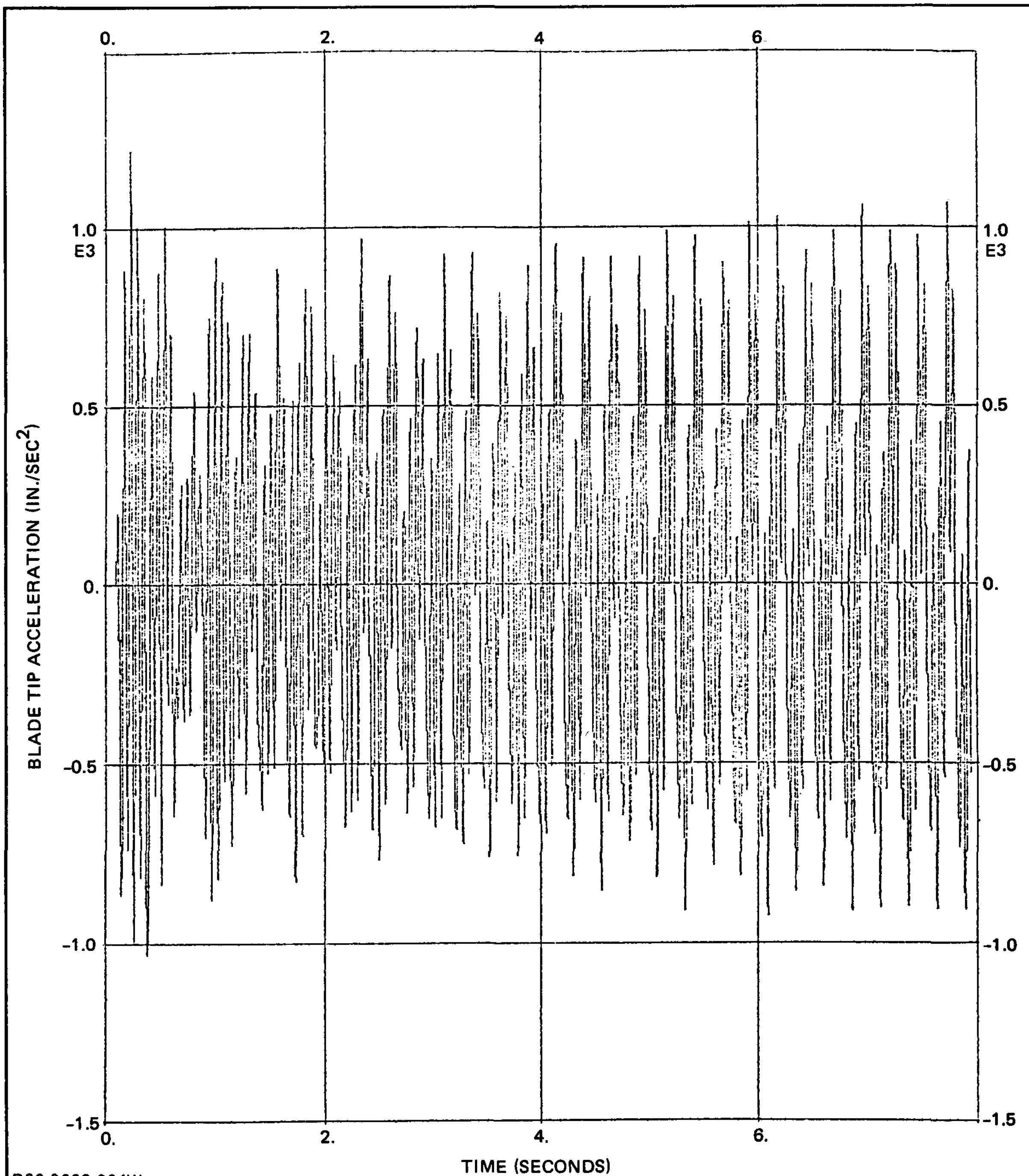


Fig. 25 Blade Normal Tip Acceleration -- Transient Response of Blade to Tower Shadow Loads at 78 RPM

Further analysis was conducted on the blade to assess the possibility of flutter. Flutter could exist where a bending and torsional frequency were closely matched, allowing an exchange of their excitation energies. A simplified model of the blade, without centrifugal stiffening and without the tower, was used to give a good approximation. Bending support of the blade was considered by taking full fixity at the root. Torsional support of the blade included the flexibility of the pitch control system through to the support structure of the actuator, and was represented by a flexibility term at the root of the blade. The first three coupled modes and frequencies are shown in Fig. 26, where h is the bending deflection normalized to 1.0 near the tip, and α is the twist angle of the blade. Mode 1 is the first bending frequency of 3.39 Hz, mode 2 is the first torsional frequency of 14.89 Hz, and mode 3 is the second bending frequency of 23.26 Hz. These frequencies are widely separated and no flutter is predicted. It is seen that the first frequency of 3.39 Hz is only an approximation of the 3.899 Hz of mode 6 of Table XXIII, which reflects the simplified model. However, it is perfectly adequate for flutter determination for the 8 kW WTG range of interest.

In calculating the natural frequencies of the WTG system as given in Table XXIII, no account was taken of the stiffening effects of the centrifugal forces acting on the rotor. These were subsequently considered, and the results are shown in the Campbell diagram, Fig. 27. Generally, it is seen that the centrifugal stiffening of the 8 kW WTG rotor is not a very influential term, changing frequencies by less than 5%.

Although the rotor will be balanced upon installation, subsequent ice formation could introduce an imbalance to the rotor, thus producing cyclic loading on the hub from centrifugal forces. Analysis of this condition showed stress cycles which were low compared to the yield strengths of the structure; 4.6% for the blade stub shaft and 4.7% for the tower. As further protection against excessive vibration, the tower could be provided with a vibration sensor which would command the blades to go to feather upon exceeding an acceleration threshold.

Passage of the wind past the tower would produce stable vortex shedding on alternate sides of the tower (Karman Vortex Streets). Over the range of Reynolds number of interest to us, the frequency of vortex shedding is considered linear with wind speed (refer to pages 12-18 of the Standard Handbook for Mechanical Engineers, Marks). It was found that a 13.9 mph wind was required to produce a vortex shedding frequency of 2.77 Hz, which is the tower lateral bending frequency (Fig. 28). However, there is little energy in the vortices at this speed, and since the frequency is also well separated from the WTG operating speed, no excitation is expected from Karman Vortex Streets. Since the wind velocity causing a given excitation is proportional to the diameter of the member, the effect of Karman Vortex Streets is even less on the guy rods.

One dynamic effect for which there was no exact solution was galloping of the guy rods. Den Hartog discussed this phenomenon in pages 299-305 of his "Mechanical Vibrations". In discussing the galloping of transmission

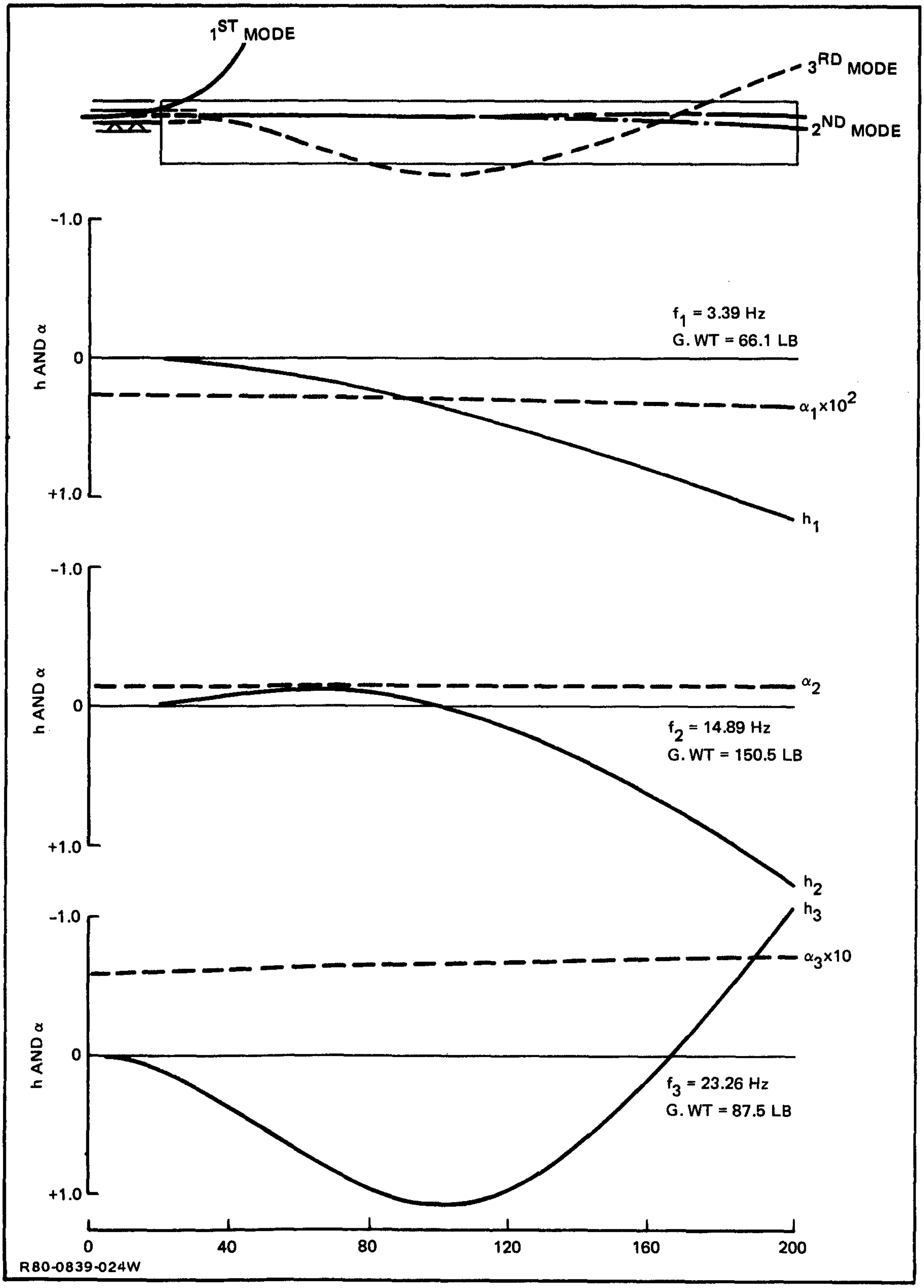


Fig. 26 8 kW Rotor Blade Modes 1.5 Ft. Chord – Pitch Control Flex Included

lines, he stated "It has never been observed in countries with a warm climate, but it occurs about once every winter in the Northern states and in Canada, when the temperature hovers around 32°F and when a rather strong transverse wind is blowing. In most cases sleet is found on the wire." The conclusion was that unstable aerodynamic forces opposing the elastic restoring forces in the wire caused it to vibrate in its first natural mode. For the 3/4 in. diameter steel rod guys pretensioned to 4,000 lbs., the frequency of this mode was calculated at 2.59 Hz, again well separated from the operating speed of the WTG. Compared with transmission lines, the guy rod would be much stiffer and its excursion lower, so that less change could be expected in the angle of attack of a sleeted guy, and therefore much less prone to galloping.

The high speed shaft operates within a narrow speed range around 1800 rpm. It consists of a solid 1 7/8 in. diameter steel shaft of 42 in. length, with two 30 lb. weight couplings 6 in. from the ends. For a perfectly aligned shaft, the critical speed is 3,020 rpm. It is shown that an eccentricity of 0.010 in. would reduce the critical speed to 1800 rpm. However, the tolerances on the shaft are much closer than this, and the critical speed is not expected to be approached. During the check-out phase of the WTG the machine will be run at operating speed and any approach to critical speed found.

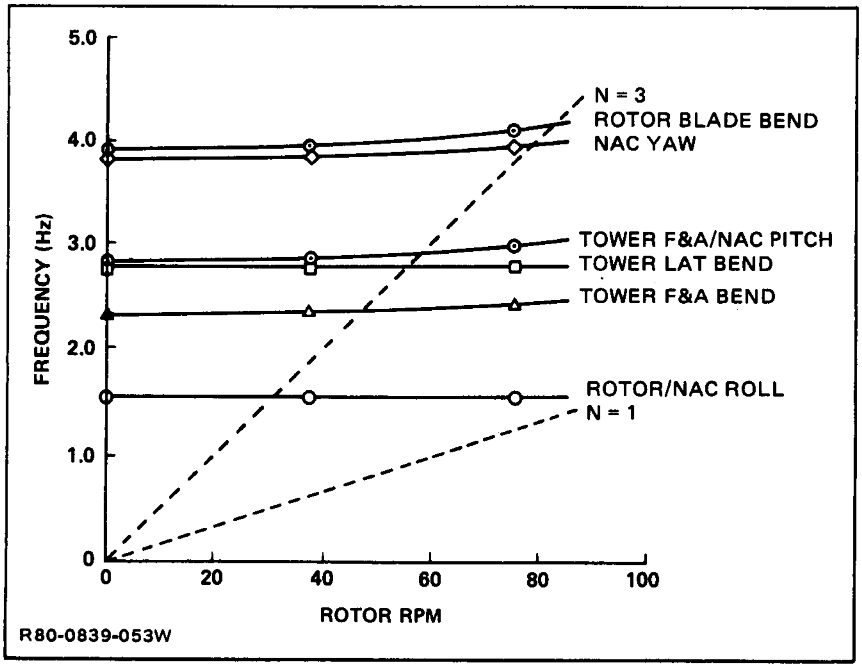


Fig. 27 Vibration Frequency Variation with Rotor RPM (Campbell Diagram)

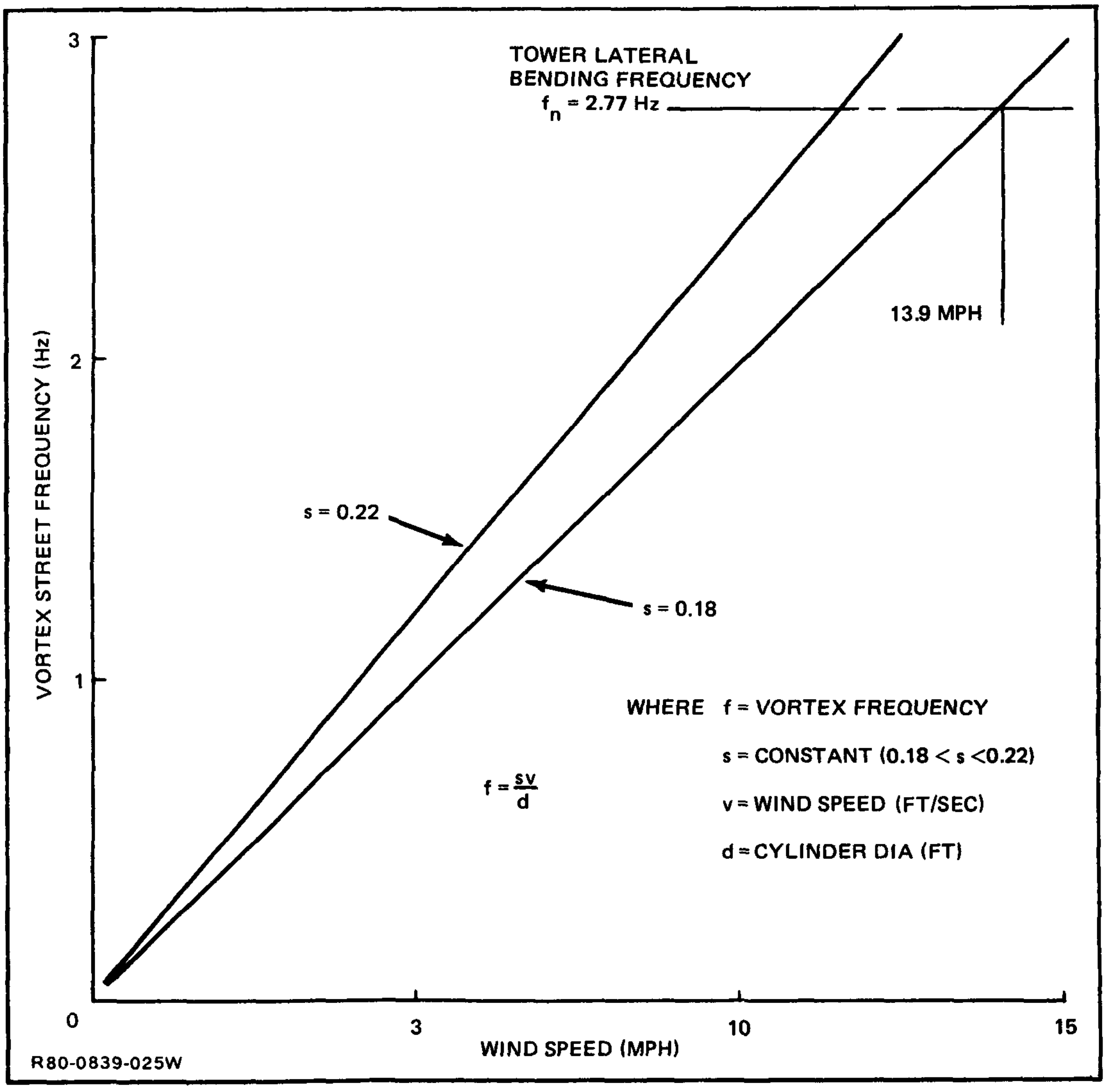


Fig. 28 Karman Vortex Street Frequency vs. Wind Speed for 16 In. Dia. Tower

6.5 ELECTRICAL SUBSYSTEM

6.5.1 Summary

The electrical power and control systems have been designed to provide a simple, efficient, fully automatic and fail-safe Wind Turbine Generator (WTG) operation. The primary features of the design are:

- 20 kW high efficiency induction generator
- Direct utility grid interface - no paralleling or synchronization equipment required
- Inherent voltage, frequency, and rotor speed control established by the utility grid

- Microprocessor controlled system providing fail-safe operation and a software programming capability

- Simplified system concept and design using environment matched components with most electrical components located on the ground.

Component selection and configuration design were the results of in-depth trade studies which developed a cost effective approach to maximize annual energy production, and provided a highly reliable system.

Figure 29 is a schematic showing the electrical components located in the nacelle. (Not shown in the schematic is the nacelle J box and the slip ring connector.)

The following sections provide a detailed description of the electrical/control system.

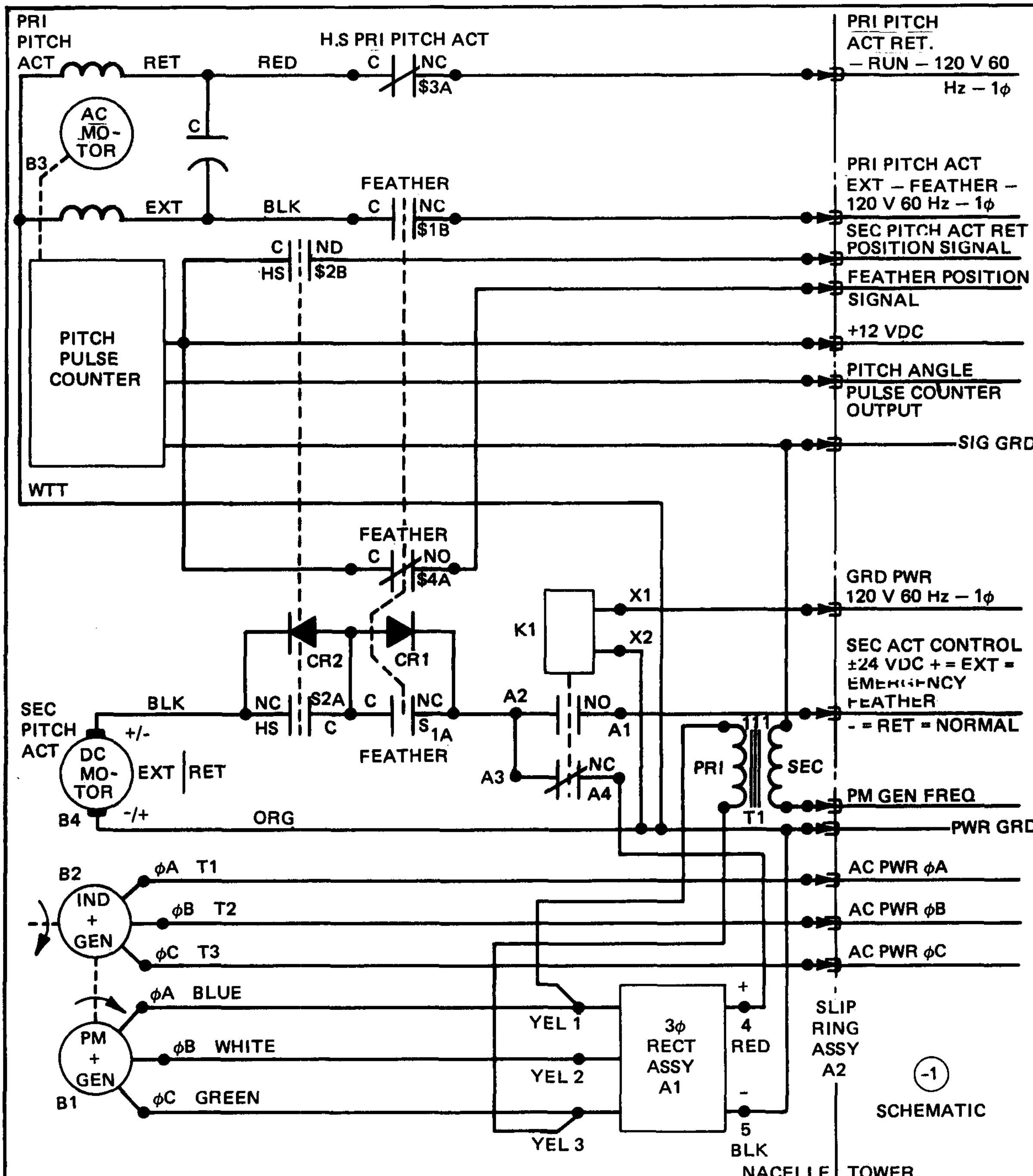
6.5.2 Generator/Power System

Trade studies have indicated that an Induction Generator is more cost effective than a separately excited synchronous alternator and a line commutated inverter when total energy production cost are compared.

The induction generator utilized in the 8 kW WTG system is basically a 30 HP, 4 pole, 3 phase, 60 Hz Induction motor on a 284 TZ frame. It produces a 20 kW (nameplate 19.7 kW) output at maximum wind speed, with a rotor speed of 1835 rpm. It is a Westinghouse Model - TUDP, Serial 7905, Style No. 79A58791 unit. The stator winding design has been modified to reduce the air gap flux density, which greatly improves the efficiency and power factor, (see Fig. 30), particularly at the lower power output levels. Although this modification reduces the peak pullout torque capability of the generator, the design still provides a 50% margin on torque with line voltage at 85% of nominal and wind speed in excess of 40 mph. The slip at full load will be approximately 2%. This modification reduces the amplitude and time constant of the initial current transient when the generator is initially connected to the line, when at or near synchronous speed. The initial current peak of 140 amps decays with a time constant of 0.02 sec., and is down to 67% of peak in 1/2 cycle and to 44% in one cycle. This generator, operating under these starting conditions, presents a considerably less severe current transient to the utility grid than the starting transient of a conventional 3 phase, 30 HP induction motor when it is initially connected to the line.

6.5.3 Rotor Blade Pitch Control System

Two linear actuators are utilized to provide a highly reliable rotor pitch control capability. Under normal conditions, the 115 V, 60 Hz primary actuator will be capable of changing the blade pitch angle from the high speed run position to the feather position in 4-6 sec. High quality limit switches provide the power control to limit actuator travel. This



NOTE: SWITCH POSITIONS SHOWN IN "FEATHER". RELAY K1 SHOWN IN THE "LOSS OF GROUND POWER" POSITION. K1 = SEC PITCH ACT CONTROL RELAY.

S_{1A} & S_{1B} = FEATHER POSITION SW – POWER

S_{2A} & S_{2B} = SEC PITCH ACT (RET POSITION) SW – POWER – INSTRUMENTATION

S_{3A} = HS PRI PITCH ACT

S_{4A} = FEATHER POSITION SW – INSTRUMENTATION

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Fig. 29 Schematic of Electrical Components in the Nacelle

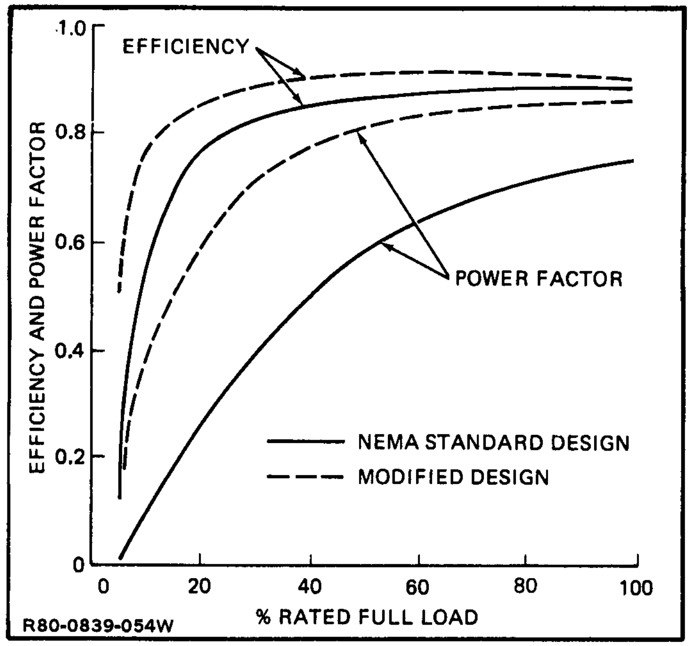

Fig. 30 Generator Performance Comparisons

actuator will also provide the control to position the blade to the start-up position of approximately 27°. Implementation of the start-up blade angle capability is provided by the use of the programmable controller, within the MCU (Main Control Unit) and the pulse counter located within the primary actuator housing. This mechanization will permit angular position control of the blades to any angle between high speed run and feather, with a resolution of approximately 1.5°.

The secondary actuator mechanically in series with the primary actuator operates from 28 V dc, and provides a generically different back-up to the primary actuator. In the event of a utility grid power failure, the automatic, fail-safe operation of this actuator is initiated by the release of the holding relay, K1, as shown in Fig. 29. The permanent magnet generator, mechanically connected to the rotor, will provide the energy to permit this actuator to reposition the rotor blades from high-speed run position to the feather position. The backup feather action will be accomplished in less than 15 sec during high winds. This rapid operation will minimize rotor overspeed after a grid disconnect, and will be completed before the rotor stops its rotation.

The utility company should not be concerned with the connecting transient, since it will be considerably less than that of a motor starting on the line. The Induction Generator is inherently safe. It cannot sustain a terminal voltage when the utility grid excitation is lost. An exception to this might exist if a large number of capacitors remain on the line connected to the generator when the grid goes down. However, an unstable voltage and frequency will be generated, and the MCU logic will safely shut the system down.

The 3 phase power from the generator passes through the three 40 amp rated slip rings, down the tower through the line contactor box in the equipment shed, to the customer supplied safety/disconnect box, which is tied into the utility grid. A matching transformer will be required for other than 480 V system.

6.5.4 Main Control Unit (MCU)

The MCU consists of three subassemblies; the controller assembly, the power contactor/monitor assembly, and the transformer panel as shown in Fig. 31. Also shown is the customer supplied safety switch and fuse panel.

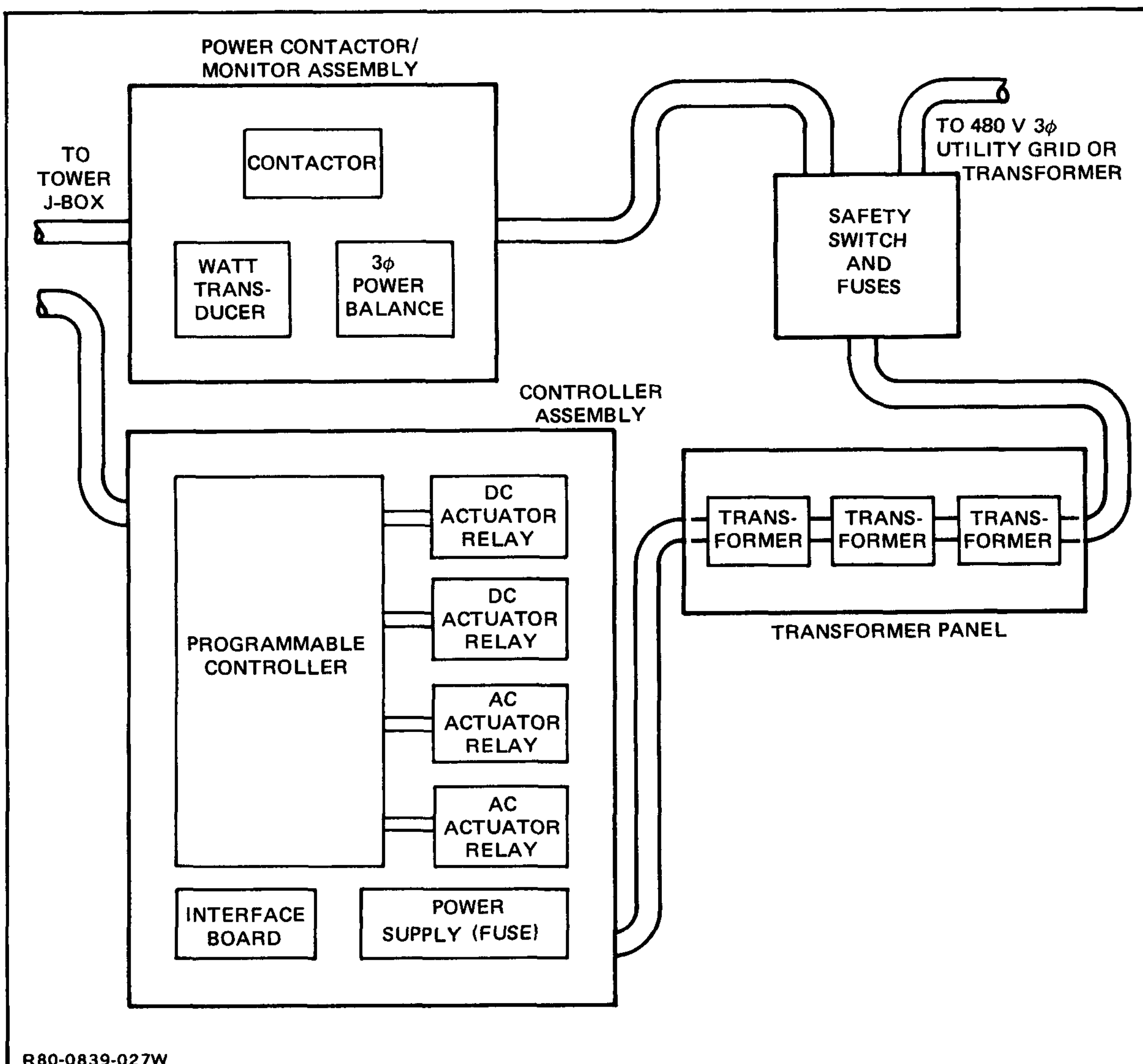

Fig. 31 Typical 8 kW Prototype WTG Installation

The transformer panel contains three single phase transformers which are used to reduce the 480 V grid potential to 120 V with the required phase relationships for the controller logic circuitry.

The power contactor/monitor assembly contains three components; the contactor, the watt transducer, and the 3 phase power monitor. The main power contactor provides for the connection of the Induction Generator to the utility grid in response to a command from the controller. The contact ratings are 45 amps. The watt transducer provides an output signal proportional to real power, and is used by the controller to monitor generator power output as well as negative power flow indication. The 3 phase power monitor provides a logic signal to the controller in the event that the 3 phase power grid becomes unbalanced.

The controller assembly consists of three components; an interface board, a power supply, and a programmable controller. The interface board provides the appropriate signal conditioning to the permanent magnet generator (PMG) output to indicate rotor rpm, and to the power monitor output to indicate excessive power generation or negative power generation. The power supply provides the positive and negative 28 V dc used by the secondary actuator, as well as the ac and dc voltages required by the interface board and the programmable controller.

The programmable controller will be selected from one of the many commercially available units, such as the Struthers-Dunn Director 1001, the Square-D SY/MAX, the Texas Instruments Program Master, the Reliance Auto Mate, the Computer Products RTP, or others.

Table XXIV lists the fourteen inputs provided to the controller. All inputs are bi-level in nature originating from switch closures or electronic circuitry. The pulses from the contact anemometer are processed within the controller to provide internal logic at four (or more, as may be required) discrete wind speeds. Similarly, the signal from the pulse counter within the primary actuator is processed to permit multiple intermediate blade angle positions.

Table XXV lists the fifteen outputs from the controller. Five outputs are used to control the power contactor and the two rotor blade pitch control actuators. Five outputs energize fault indicators which retain their indication, even if power is lost. The remaining five outputs energize the indicating lamps.

6.5.5 System Logic

Overall system logic is presented in flow-chart form, in Fig. 32. The relationship between all inputs, outputs, timers, and counters is shown. All logic and the timer and counter quantities are software programmable. It is planned to provide 1 K of non-volatile, yet reusable memory (LEROM-UV light erasable).

Table XXIV Controller Inputs

1. ANEMOMETER
   a. < 7 MPH OR > 7 MPH
   b. < 35 MPH OR > 35 MPH
   c. < 50 MPH OR > 50 MPH
   d. < 10 MPH OR > 10 MPH
2. PRIMARY FEATHER COMMAND (FROM SWITCH)
3. SECONDARY FEATHER COMMAND (FROM SWITCH)
   GENERATOR FREQUENCY (FROM INTERFACE BOARD)
4. $< XH_z$ OR $> XH_z$ $(X \approx 15H_z)$
5. $< YH_z$ OR $> YH_z$ $(Y \approx 60\ H_z)$
6. $< ZH_z$ OR $> ZH_z$ $(Z \approx 61.H_z)$
7. CURRENT IMBALANCES SENSOR < A AMPS OR > A AMPS
8. POWER LIMIT SENSOR < W WATTS OR > W WATTS
9. NEGATIVE POWER < P WATTS OR > P WATTS
10. ACTUATOR FEATHER POSITION (LIMIT SWITCH)
11. HIGH SPEED POSITION OF SECONDARY ACT (LIMIT SWITCH)
12. RESET (SWITCH POSITION)
13. POWER ON (SWITCH POSITION)
14. ROTOR BLADE ANGLE POSITION COUNTER
   a. < C OR > C (COUNTS FOR INTERMEDIATE BLADE ANGLE)
   b., c. ALTERNATE COUNTS (FUTURE USE)

R80-0839-029W

Table XXV Controller Outputs

1. LINE CONTACTOR, ON/OFF
2. PRIMARY ACTUATOR, DRIVE TOWARDS HIGH SPEED RUN POSITION
3. PRIMARY ACTUATOR, DRIVE TOWARDS FEATHER POSITION
4. SECONDARY ACTUATOR, DRIVE TOWARDS HIGH SPEED RUN POSITION
5. SECONDARY ACTUATOR, DRIVE TOWARDS FEATHER POSITION
6. FAULT INDICATOR PMG FAILURE
7. FAULT INDICATOR ANEMOMETER FAILURE
8. FAULT INDICATOR OVERSPEED
9. FAULT INDICATOR PRIMARY ACTUATOR
10. FAULT INDICATOR IMBALANCE 3ϕ POWER
11. LAMP SYSTEM READY
12. LAMP PMG FEATHER
13. LAMP PRIMARY FEATHER
14. LAMP SECONDARY FEATHER
15. LAMP RESET

R80-0839-028W

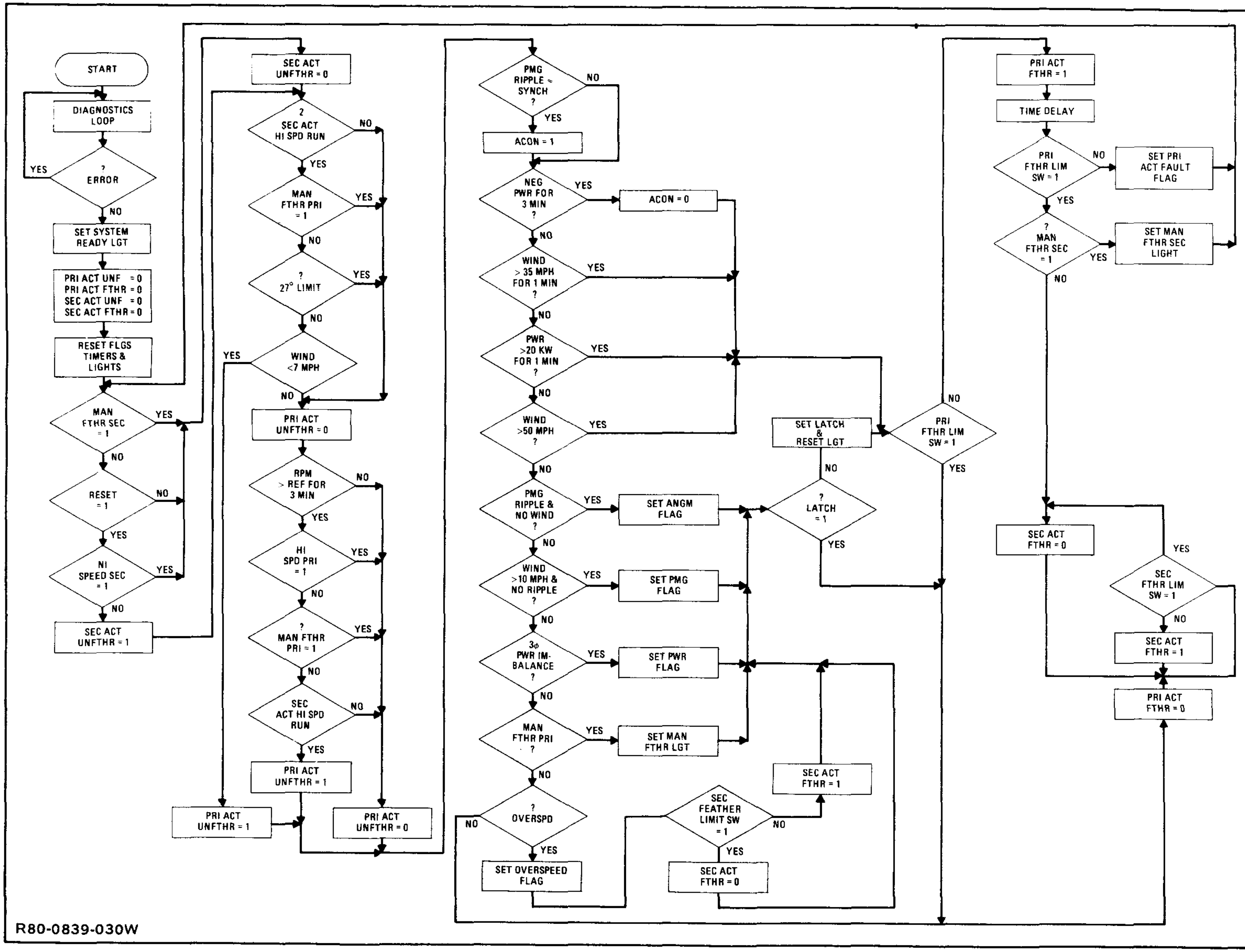

Fig. 32. Control Logic Diagram

Changing system logic may be readily accomplished by reprogramming a memory chip and inserting it into the controller in place of the existing chip.

Support equipment will be available to permit the monitoring of the value of all timers and counters. Additionally, the status of all inputs and outputs can be monitored on-line as well as the internal storage locations. The support equipment also provides a capability to change the presets of selected timers and counters. LEDs are provided on the controller to further indicate input/output status, and may be used to isolate system faults.

Figure 33 shows the panel layout on the controller assembly portion of the MCU.

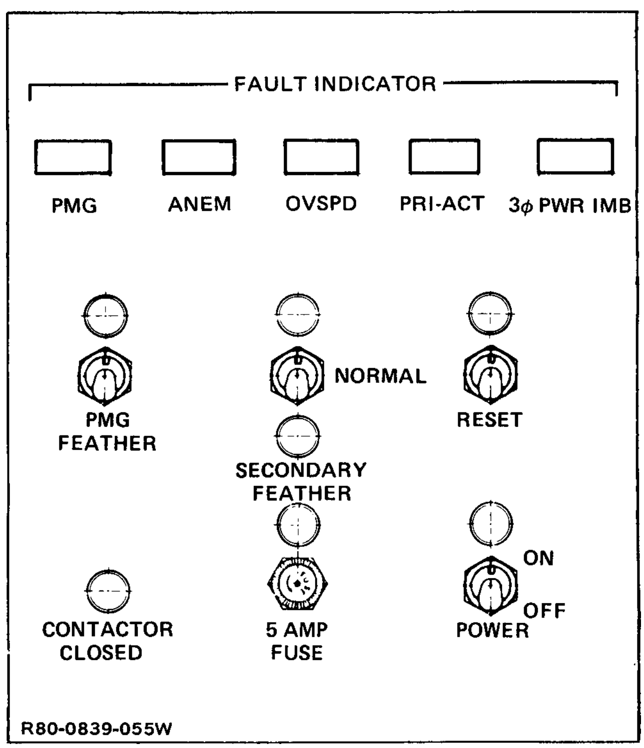

Fig. 33 8 kW Wind Turbine Generator Control Panel

7.0 CONCLUSIONS AND RECOMMENDATIONS

The Grumman Energy Systems, Inc. (GESI) design of a WTG is fully responsive to the U.S. Department of Energy (DOE) Statement of Work for an 8 kW WTG. A major effort was made to design a reliable machine, capable of operating for 25 years with a minimum maintenance requirement. In only one area did the WTG fall short of the original statement, that of achieving the target capital cost goal of $750/kW (1977 dollars). It is expected that future testing of the machine will allow design improvements which will reduce the cost per kW.

It is recommended that Grumman, in conjunction with DOE, proceed with the manufacture and testing of the Wind Turbine Generator to demonstrate the capability of wind energy production.

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