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UC-60

ENERTECH 15-KW WIND-SYSTEM DEVELOPMENT

Phase 1
Design and Analysis

Technical Report

September 1981

Prepared by
ENERTECH CORPORATION
Post Office Box 420
Norwich, VT 05055

For
Rockwell International Corporation
Energy Systems Group
Rocky Flats Plant
Wind Systems Program
Post Office Box 464
Golden, Colorado 80402

Subcontract No. PF-07711T

As a Part of
THE UNITED STATES DEPARTMENT OF ENERGY
WIND ENERGY TECHNOLOGY DIVISION
FEDERAL WIND ENERGY PROGRAM

Contract No. DE-AC04-76DP03533

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ABSTRACT

A utility interfaced wind machine rated for 15 kW at 9 m/s (20.1 mph) has been designed to be cost effective in 5.4 m/s (12 mph) average wind sites. The unit is designed to meet or exceed environmental conditions as specified in Contract PF07711T.

Approximately 18 months into the research and development program a completed design meeting contract specifications was submitted to the buyer. The design is for a horizontal axis, down wind machine which features three fixed pitch wood-epoxy blades and free yaw. Rotor diameter is 44 feet (13.4 meters). Unit shutdown is provided by an electro-hydraulic brake. Blade tip brakes provide back-up rotor overspeed protection. Design merits have been verified through dynamic truck testing of a prototype unit.

This report was edited by D.M. Dodge at the Rocky Flats Plant

TABLE OF CONTENTS

Page

LIST OF FIGURES

Frontispiece - Enertech 15 kW Wind System Prototype (1981 photo).

Page

LIST OF TABLES

Page

NOMENCLATURE

A or amp - ampere
ac - alternating current
AISI - American Iron and Steel Institute
AKWH - Total Annual kWh Produced
AOM - Uniform Annual Operation and Maintenance Costs
AWG - American Wire Gage

C - Celcius
CD - drag coefficient
CDR - Critical Design Review
C.G. - center of gravity
C_l - lift coefficient
COE - Cost of Energy

D - diameter
dc - direct current

FCR - fixed charge rate
FDR - Final Design Review
FMEA - failure mode and effects analysis
FOB - free on board
ft - foot (feet)
ft-lb - foot pound

G & A - general and administrative cost

IC - installed system cost
I.D. - inside diameter
in - inch(es)
in-lb - inch pound

K - stress

K_t - torsional stress

kg - Kilogram

KSI - kilograms per square inch

kW - kilowatt

kW(e) - net power in kilowatts

kWh - kilowatt hours

kv - dynamic velocity factor

lb - pounds (weight)

lbf - pound force

λ - failure rate

λCOMP - component failure rate

λGEN - generic failure rate

M - moment

m - meter

max. - maximum

min. - minimum

MOV - metal oxide varister

mph - miles per hour

m/s - meters per second

MTBF - mean time between failures

MTTR - mean time to repair

M_W - weight moment

NEC - National Electric Code

Pa - annual availability (site specific)

Pa_i - intrinsic annual availability

P_b - bolt preload

PDR - Preliminary Design Review

O.D. - outside diameter

OEM - original equipment manufacturer

Re - Reynolds number

REV - revolution(s)

rpm - revolutions per minute

SF - safety factor

SLD - safe life design (design for infinite life)

T_A - aerodynamic torque

T_B - braking torque

τ - gust length

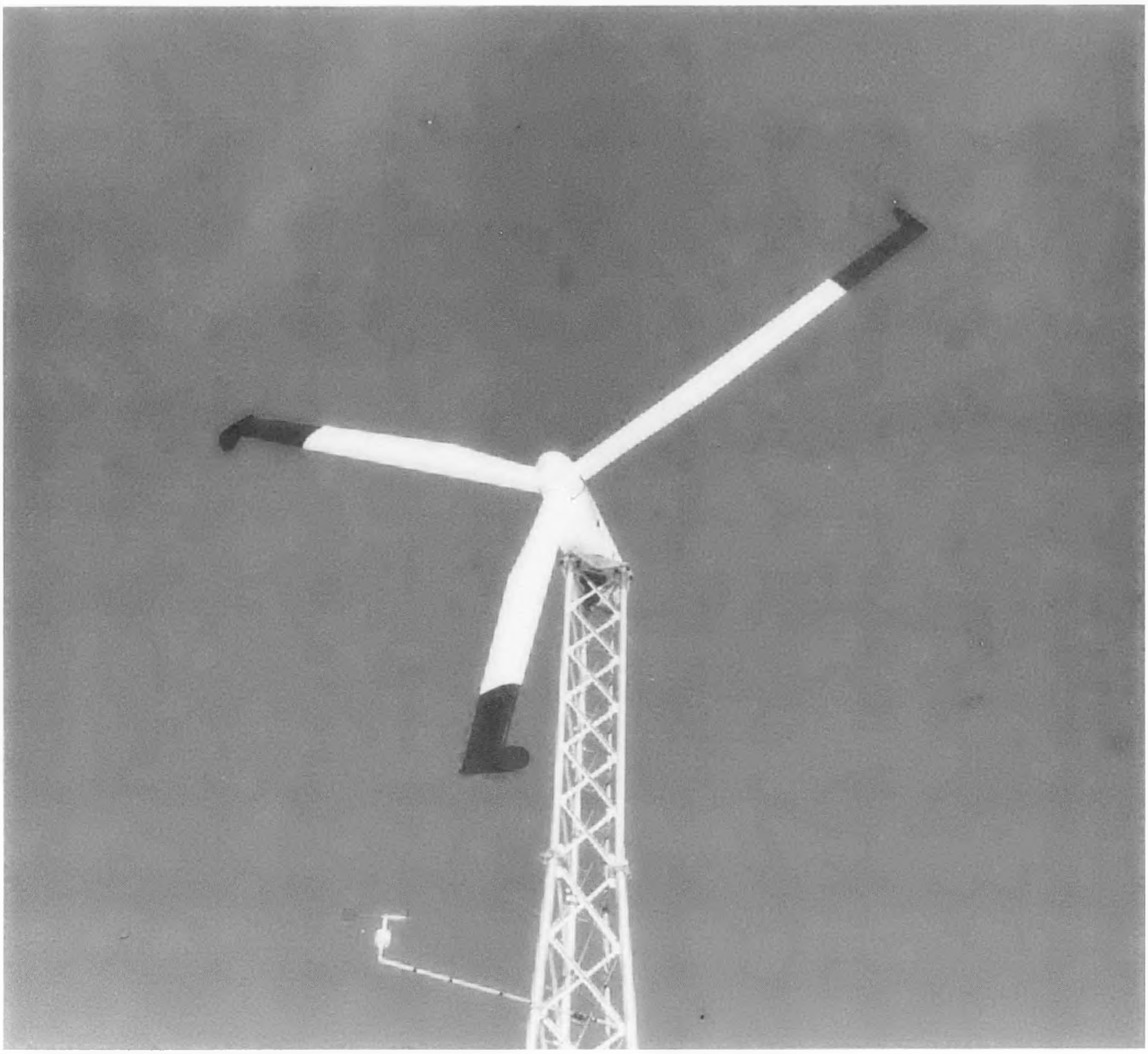
Vac - volts-alternating current

Vdc - volts-direct current

V - velocity

V - mean wind speed

° - degrees (angle or temperature)



Enertech 15 kW Wind System Prototype (1981 Photo)

1.0 INTRODUCTION AND SUMMARY OF PHASE I ACTIVITIES

1.1 Introduction

Enertech Corporation is currently engaged in a development project to fabricate and test a 15 kilowatt (kW) utility-interfaced SWECS. The machine is to be capable of producing its rated output in a 9 m/s (20.1 mph) wind and be capable of surviving wind gusts of 56 m/s (125 mph).

The work described in this report is sponsored by the United States Department of Energy, Wind Energy Technology Division, and is administered by Rockwell International, Rocky Flats Plant, Golden, Colorado.

Enertech is performing this development work under contract to Rockwell as part of the DOE program to advance the technology and accelerate the utilization of reliable and economically viable wind energy systems. The applications envisioned for the machine include small businesses, farms and residences.

Figure 1 shows the basic project organization. Enertech is the primary contractor to Rockwell. The project includes one major sub-contractor and three consultants. Gougeon Brothers of Bay City, Michigan has developed the 22 ft. wood-epoxy blades for use on the machine. Their work includes development of molds and tooling required to fabricate two sets of blades complete with stud attachments (a total of six blades). Michael D. Zuteck acts as an aerodynamic consultant and blade construction consultant. Dr. Norman T. Ham is the aeroelastic consultant. Vail Church is consulted in matters of utility interface.

The contract was awarded on August 14, 1979. Throughout the design effort was guided by design criteria and system specifications developed by Rockwell International (see Tables I and II). Note that two designs were called for: 1) a "standard" design and 2) a "special" design intended for severe environments. Completion of Phase I tasks has accounted for about thirteen months of concentrated design effort. Phase I activities were considered completed as of Final Design Review (FDR), September 12, 1980.

Figure 1

Contractor Organization

15 kW
Wind System Development Project

CONTRACTING AGENCY
DOE

PRIME CONTRACTOR
ROCKWELL INTERNATIONAL

SUB-CONTRACTOR
Enertech Corporation
Norwich, Vermont

SUB-CONTRACTORS TO ENERTECH

Electronics - Analog Systems, Boulder, CO
Nacelle - HBA Cast Products, Springfield, MA
Blades - Gougeon Bros., Bay City, MI
Hub - Joy Mfg., Claremont, NH
Glenco, Inc., Newport, NH
Gearbox - Winsmith Inc., Springfield, NY

CONSULTANTS TO ENERTECH

Aeroelasticity - Norman Ham
Utility - Vail Church
Blade Construction -
Micheal Zuteck

Table I

15 kW SYSTEM DESIGN CRITERIA

A-1 Structural

1. Survival loading condition:
   - (i) A. Wind machine operating
     B. Wind machine shutdown
   - (ii) Safety Factor of 1.5 minimum for ductile materials on yield strength, 3.0 minimum for brittle materials on ultimate strength.

2. Fatigue loading condition:
   - (i) Worst loading condition under continuous operation
   - (ii) Minimum of 10^7 cycles
   - (iii) Safety Factor of 1.5 minimum

A-2 Electrical

- (i) Protect customer electrical service from lightning strikes.
- (ii) Protect SWECS from nearby lightning strikes.
- (iii) Motor/generator to operate continuously at 17 kW in a real-life environment.
- (iv) Wiring to handle maximum expected current.

B Operation

- (i) Shutdown at high windspeeds.
- (ii) Low wind start-up/shut-down
  - (a) Start-up energy consumption
  - (b) Minimum generation windspeed (cut-in)
  - (c) Output characteristics at low winds
  - (d) Wind characteristics data
  - (e) Control system characteristics
- (iii) Shutdown when utility line fails
- (iv) Rotor overspeed control
- (v) Shutdown at excessive vibration

Table I (continued)

C-1 Installation

(i) No crane for installation
(ii) Easy installation, transportation and minimum personnel

C-2 Safety

(i) SWECS - high winds, power loss, overspeeding and environmental damage
(ii) Personnel - Utility personnel, maintenance personnel, and installation personnel

C-3 Reliability

95% Availability

C-4 Maintenance

(i) Routine maintenance: 6 months
(ii) Major maintenance: 8 years

D Utility Acceptance

(i) Disconnect during utility power failure
(ii) Frequency and voltage of generator
(iii) Utility service rating adequate to handle maximum output

E Cost vs Performance vs Technical and Schedule Risk

(i) Cost/kWh energy
(ii) Quality and Performance
(iii) No undue technical and schedule risks

Table II

15 kW SYSTEM SPECIFICATIONS

	"Standard" Design	"Special" Design
Output:	. 45,000 to 55,000 kWh/yr in a wind regime having a mean annual wind speed of 5.4 m/s (12 mph) measured at 9.1 m (30 ft) above grade level.	Same
	. 240 ± 5% VAC Single Phase, 60Hz for intertie with a utility or with an interconnection with an auxiliary generator (e.g., diesel).	Same
	. Design envelope: 13-18 kW at a windspeed between 7.1 and 9.0 m/s (16 and 20 mph).	Same
Operating Wind Range:	. Cut-in: minimize / Cut-out: maximize — with respect to minimizing energy costs	Same
	. survival (peak gust) - 56 m/s (125 mph)	74 m/s (165 mph)
Operation Environment:	. -30°C to +60°C (-22°F to +140°F)	-50°C - +60°C (-58°F - +140°F)
	. ice 1" thick*	ice 2 1/2 thick*
	. rain, dust, lightning	Same
		Salt water spray
Operation Availability:	. 95% availability factor	Same
Controls:	. Automatic startup and shutdown. Brake for locking rotor during maintenance. Rotor overspeed production.	Same
System Life	. 25 years minimum	Same
Energy Cost Goal	. 3c/kWh for 10,000th production system	Same

*Machine need not operate with ice coatings. It shall be designed to shut down or otherwise protect itself from damage should ice build-up occur during operation.

1.2 Phase I Activities Summarized

Figure 2 shows the generalized Phase I schedule and summarizes the major milestones completed in the Phase I portion of the contract. Figure 3 shows the Phase I schedule by task. Preliminary Design Review (PDR) was held in November of 1979. At that time a three-bladed fixed-pitch machine was proposed which featured a plate hub (see Figure 4). Preparation for Critical Design Review (CDR) revealed that the proposed hub would not meet the loadings adequately and that the proposed method of blade hub attachment was unsatisfactory. In addition, it was found that the production of three-phase power as originally proposed would not prove useful in the majority of unit applications (rural, small businesses). Therefore, the scope of the program was changed to encompass single-phase power and the services of Gougeon Brothers of Bay City, Michigan were retained to accomplish blade manufacture. CDR was held in June, 1980 and the only significant change arising after the review involved the substitution of a hydraulic brake in place of the originally proposed electric brake (see Figure 5). FDR was concluded in September, 1980 and authorization to proceed with Phase II was received October 3, 1980.

Table III shows the evolution of major system design characteristics, from the proposed (baseline) design, through PDR, CDR, and FDR.

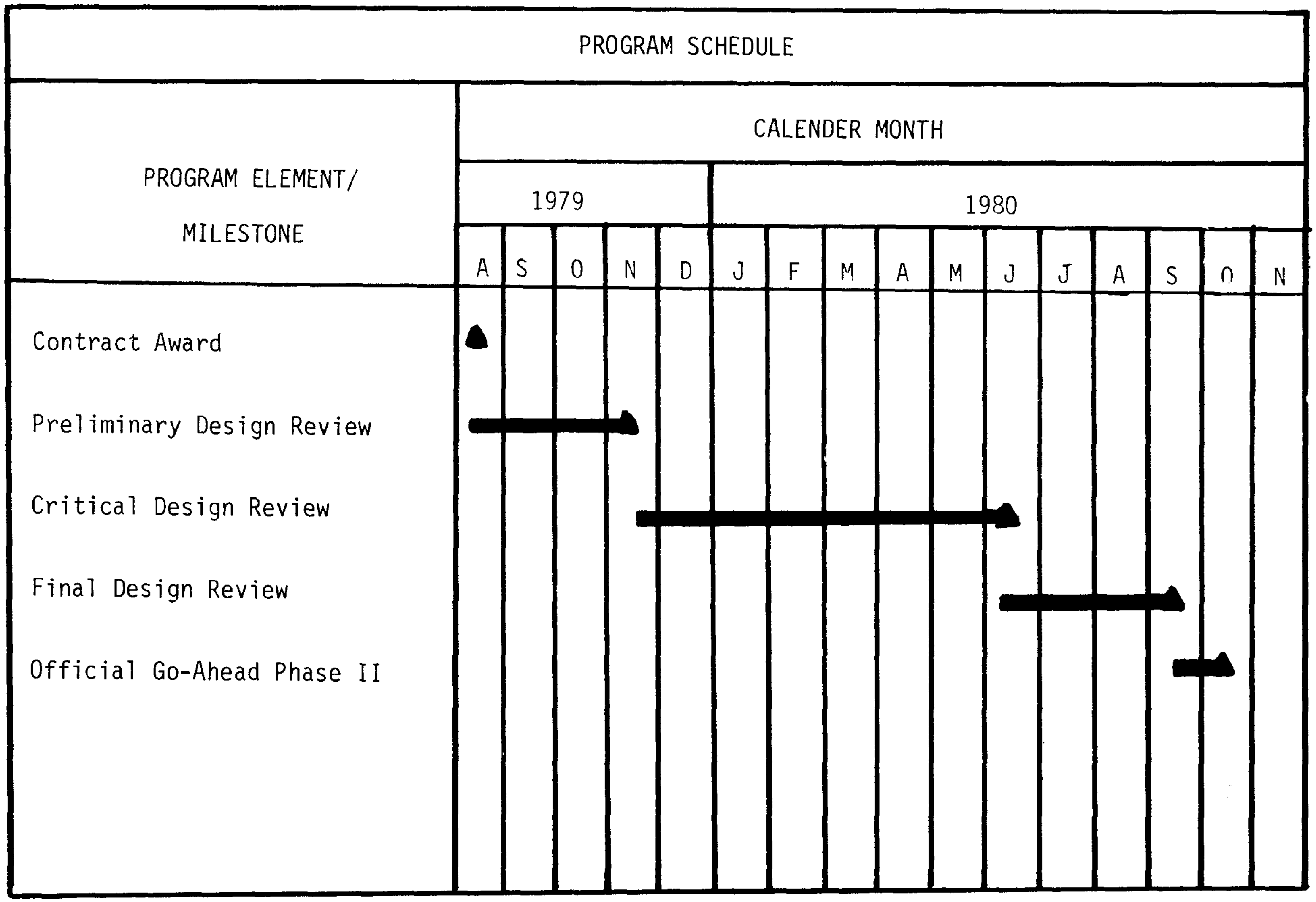


Figure 2

Program Schedule

Figure 3

Program Plan and Work Schedule

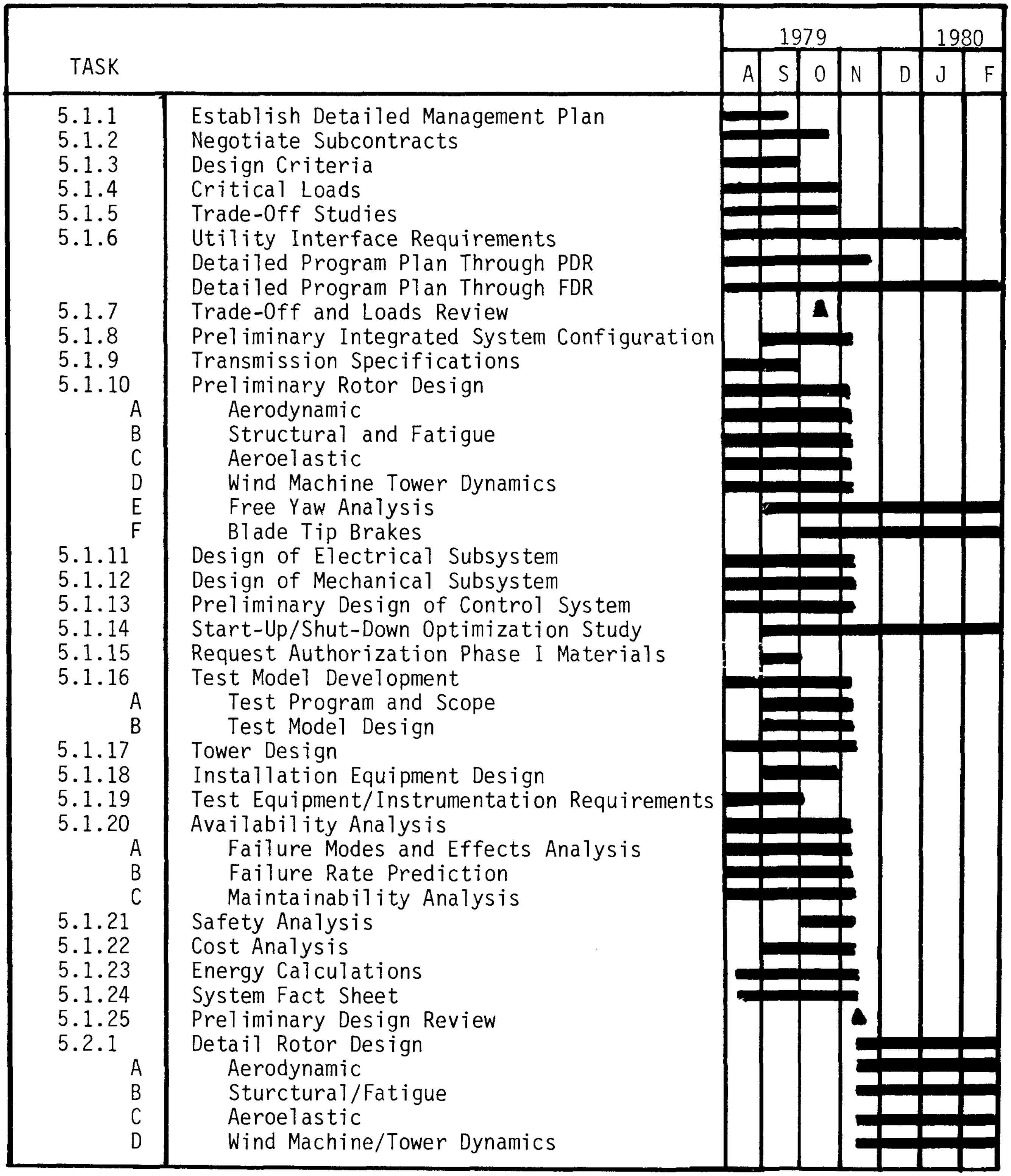
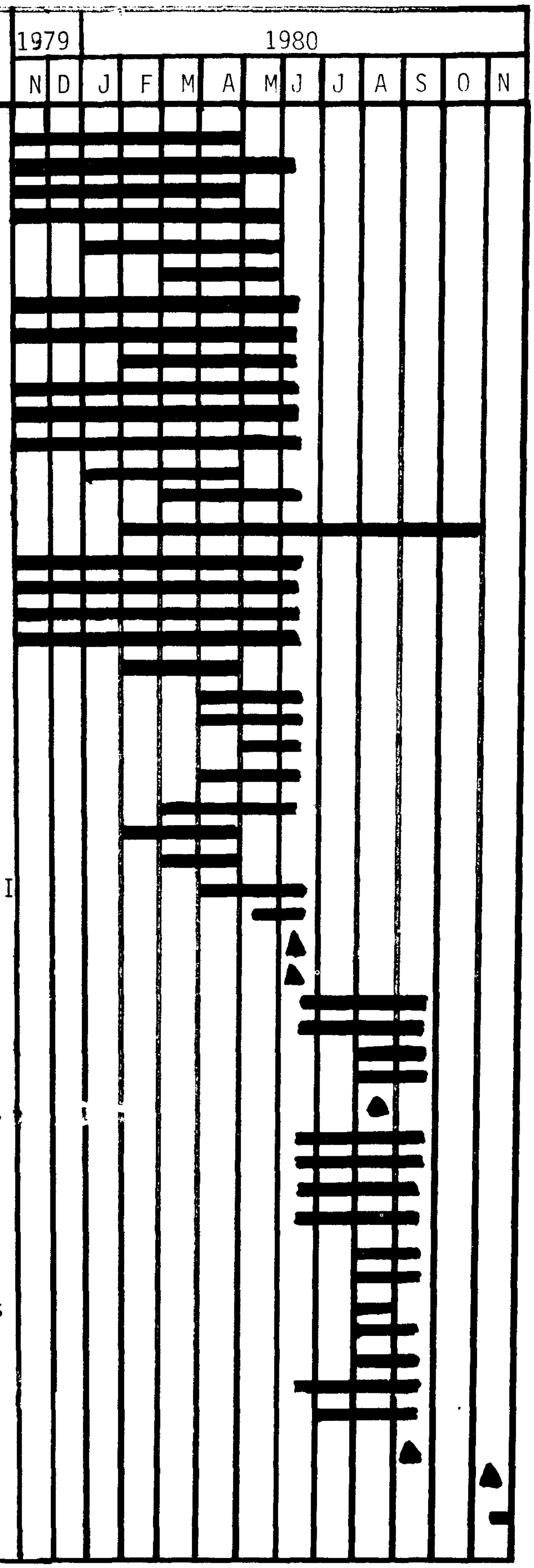


Figure 3 (cont'd)

Task	
5.2.2	Detail Design of Electrical Subsystem
5.2.3	Detail Design of Mechanical Subsystem
5.2.4	Detail Design of Control Subsystem
5.2.5	Test Model Program
A	Modification and Fabrication
B	Testing Program
5.2.6	Detail Tower Design
A	Tower Configuration
B	Foundation Configuration
C	Detail Tower Structural Analysis
D	Tower Dynamics
E	Tower/Wind Machine Interface
5.2.7	Detailed SWECS Installation Equipment
5.2.8	Update System Integration
5.2.9	Induction Generator Testing
5.2.10	Update Availability Analysis
A	Failure Modes and Effects Analysis
B	Failure Rate Prediction
C	Maintainability Analysis
5.2.11	Update Safety Analysis
5.2.12	Update Cost Analysis
5.2.13	Update Annual Energy Output Calculations
5.2.14	Update Systems Fact Sheet
5.2.15	Test Plan Presentation
5.2.16	Mounting for Test Instrumentation
5.2.17	Recommend Special Test Instrumentation
5.2.18	Long-Lead Items List
5.2.19	Update Program Plan for Balance of Phase I
5.2.20	Prepare Updated Phase II Program Plan
5.2.21	Critical Design Review
(Task 6.17)	Indicate Spare Parts to be Fabricated
5.3.1	Final Design Modifications
5.3.2	Preparation of Final Drawings
5.3.3	Final System Integration
5.3.4	Model Test Results Documentation
5.3.5	Request Authorization for Long-Lead Items
5.3.6	Final Availability Analysis
A	Failure Modes and Effects Analysis
B	Failure Rate Prediction
C	Maintainability Analysis
5.3.7	Final Safety Analysis
5.3.8	Final Cost Analysis
5.3.9	Final Energy Output and Cost Calculations
5.3.10	Systems Facts Sheet
5.3.11	Update Phase II Program Plan
5.3.12	Phase II Test Plan Recommendations
5.3.13	Software Report
5.3.14	Final Design Review
5.3.15	Submit Draft of Phase I Report
5.3.16	Submit Final Phase Report



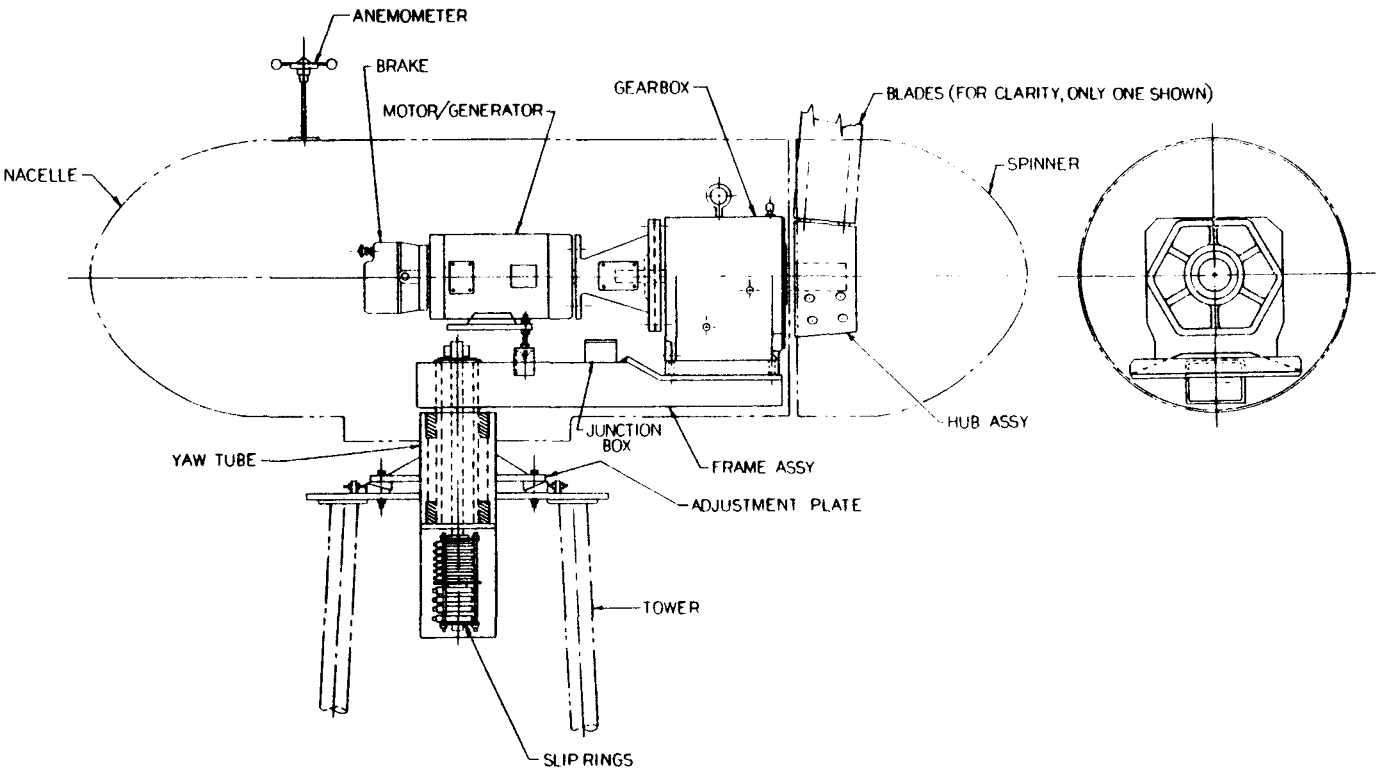


Figure 4

15 kW Integrated System Configuration at PDR

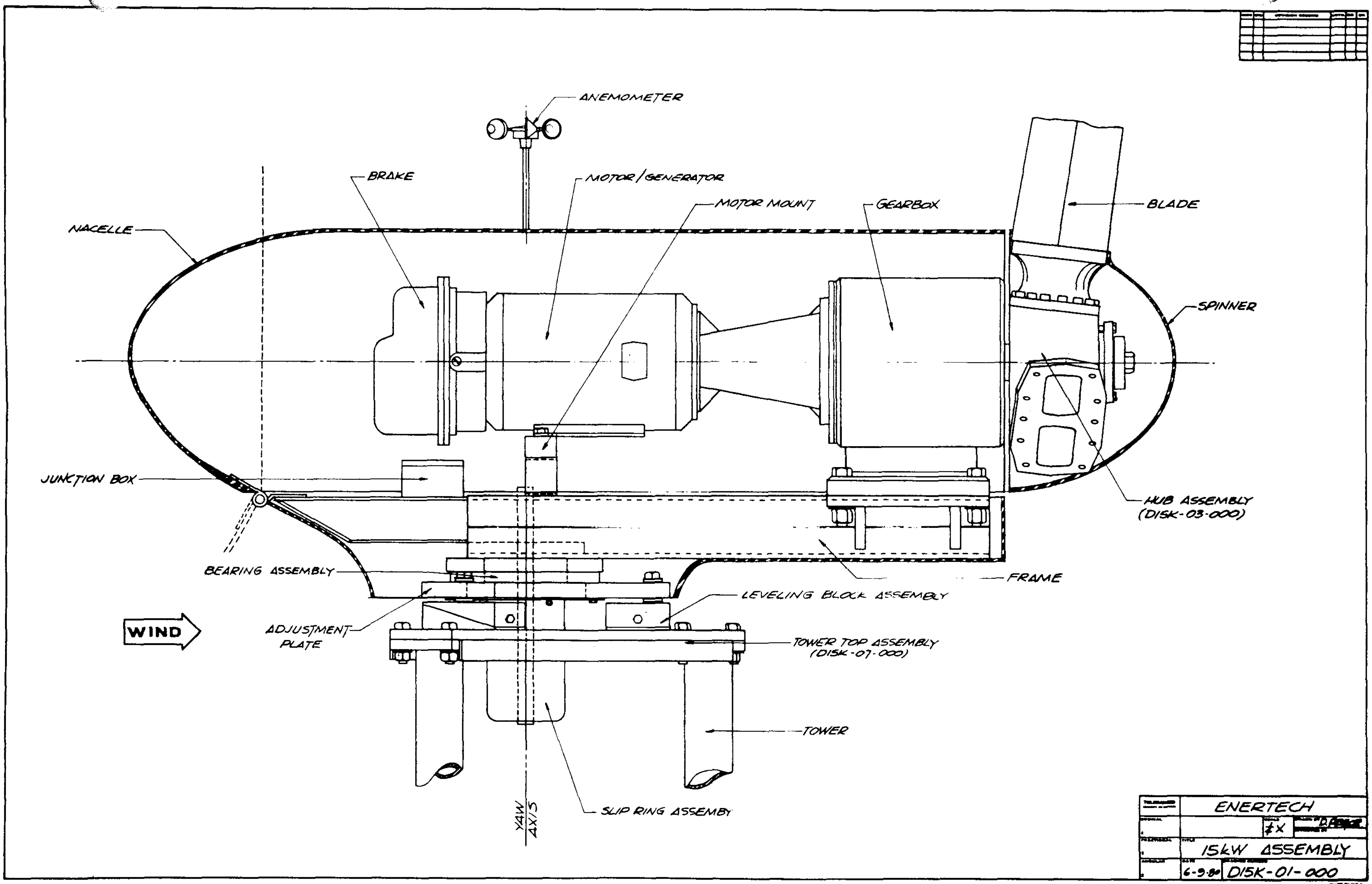


Figure 5

15 kW Integrated System Configuration at CDR

Table III

Design Characteristics
Enertech 15 kW Wind System

STANDARD DESIGN	Proposed (Baseline)	PDR	CDR	FDR
GENERAL CONFIGURATION				
Output Power (kW) @ Rated Wind V	15 kW @ 20.1 mph	15 kW @ 20.1 mph	15 kW @ 20.1 mph	15 kW @ 20.1 mph
Axis - Vertical/Horizontal:	Horizontal	Horizontal	Horizontal	Horizontal
Rotor Location, Upwind/Downwind:	Downwind	Downwind	Downwind	Downwind
Rotor Diameter (ft): (Width & Height - Vert. axis	44	44	44	44
Number of Blades:	3	3	3	3
Centerline Hub Height (ft):	62	62	62	62
Method of Rotor Overspeed Control:	Blade Tip Brakes	Blade Tip Brakes	Blade Tip Brakes	Blade Tip Brakes
Type of Output Voltage & Ø:	240/480, 3 Ø	240, 3 Ø	240 Single Phase	240 V Single Phase
System Weight/Tower Weight (lb):	2400/4000	2875/2985	2850/3224	2950/3314
System cost (FOB) - 1st, 100th, 1000th & 10,000th	32723; 13092 -- ; 6574	34388; 18318 14178; 11835	24457; 18597 15900; 11981	22869; 19382 16353; 11934
PERFORMANCE PARAMETERS				
System Cp @ Rated Wind Velocity:	0.25 @ 20.1 mph	0.236 @ 20.1 mph	0.236 @ 20.1 mph	0.236 @ 20.1 mph
Cut-in Wind Velocity (mph):	8	8	8	8
Cut-out Wind Velocity (mph):	40	40	40	40
Survival Wind Velocity (mph):	125 peak gust	125 peak gust	125 peak gust	125 peak gust
ANNUAL OUTPUT (kWh)/Cost of Energy (¢/kWh)				
(For avg. wind veloc. meas @ 30 ft based upon NASA Wind Dist) Avg wind veloc. @ 30 ft - NASA Dist.	Based on NASA Lewis Distribution & 90% Availability	Based on Raleigh Distribution & 90% Availability		Based on Raleigh Distribution & 90% Availability
@ 12 mph:	51500/3.0	51908/3.35	51908/3.20	54608/3.26
@ 14 mph:	-	63265/2.75	63265/2.63	68295/2.61
@ 15 mph:	70600/2.2	67057/2.59	67057/2.48	73666/2.42
@ 16 mph:	-	70498/2.46	70498/2.36	78011/2.28
@ 18 mph:	80000/2.0	74218/2.34	74218/2.24	83610/2.13

SPECIAL DESIGN	Proposed (Baseline)	PDR	CDR	FDR
GENERAL CONFIGURATION				
Output Power (kW) @ Rated Wind V	15 kW @ 20.1 mph	15 kW @ 20.1 mph	15 kW @ 20.1 mph	15 kW @ 20.1 mph
Axis - Vertical/Horizontal:	Horizontal	Horizontal	Horizontal	Horizontal
Rotor Location, Upwind/Downwind:	Downwind	Downwind	Downwind	Downwind
Rotor Diameter (ft): (Width & Height - Vert. axis	44	44	44	44
Number of Blades:	3	3	3	3
Centerline Hub Height (ft):	62	62	62	62
Method of Rotor Overspeed Control:	Blade Tip Brakes	Blade Tip Brakes	Blade Tip Brakes	Blade Tip Brakes
Type of Output Voltage & Ø:	240/480, 3 Ø	240, 3 Ø	240 Single Phase	240 V Single Phase
System Weight/Tower Weight (lb):		3020/4350	2900/4695	
System cost (FOB) - 1st, 100th, 1000th & 10,000th		41315; 22839 17925; 15071	25093; 21226 18246; 13621	25463; 22011 18671; 14031
PERFORMANCE PARAMETERS				
System Cp @ Rated Wind Velocity:	0.25 @ 20.1 mph	0.236 @ 20.1 mph	0.236 @ 20.1 mph	0.236 @ 20.1 mph
Cut-in Wind Velocity (mph):	8	8	8	8
Cut-out Wind Velocity (mph):	40	40	40	40
Survival Wind Velocity (mph):	165 peak gust	165 peak gust	165 peak gust	165 peak gust
ANNUAL OUTPUT (kWh)/Cost of Energy (¢/kWh)				
(For avg. wind veloc. meas @ 30 ft based upon NASA Wind Dist) Avg wind veloc. @ 30 ft - NASA Dist.	Based on NASA Lewis Distribution & 90% Availability	Based on Raleigh Distribution & 90% Availability		Based on Raleigh Distribution & 90% Availability
@ 12 mph:	51500/-	51908/4.76	51908/3.59	54608/3.71
@ 14 mph:	-	63265/3.9	63265/2.95	68295/2.97
@ 15 mph:	70600/-	67057/3.68	67057/2.78	73666/2.75
@ 16 mph:	-	70498/3.5	70498/2.64	78011/2.60
@ 18 mph:	80000/-	74218/3.33	74218/2.51	83610/2.42

2.0 SYSTEM DESIGN SUMMARY

Figures 6 and 7 show the overall final design assembly of the 15 kW SWECS, while Figure 8 shows predicted power output. The wind machine is a horizontal axis, downwind machine with three epoxy laminated wooden blades spanning a diameter of 13.4 m (44 ft). Speed of the fixed pitch rotor is controlled by generator loading and blade stalling in higher winds. Blade tip brakes deploy in the event of an emergency rotor overspeed condition.The machine incorporates an electro-hydraulic brake which stops the machine in very low windspeeds or windspeeds above 19.7 m/s (45 mph). The machine features a utility-interfaced induction generator.

Throughout the design evolution the basic configuration of a three-bladed fixed pitch machine has been retained. Early studies indicated that the originally proposed method of hub blade attachment would not be satisfactory to withstand the design loads. Also, between PDR and CDR it was decided that wood-epoxy blades would be advantageous; therefore the present blade stud attachment method was devised. A summary of the trade-off studies appears in Table IV.

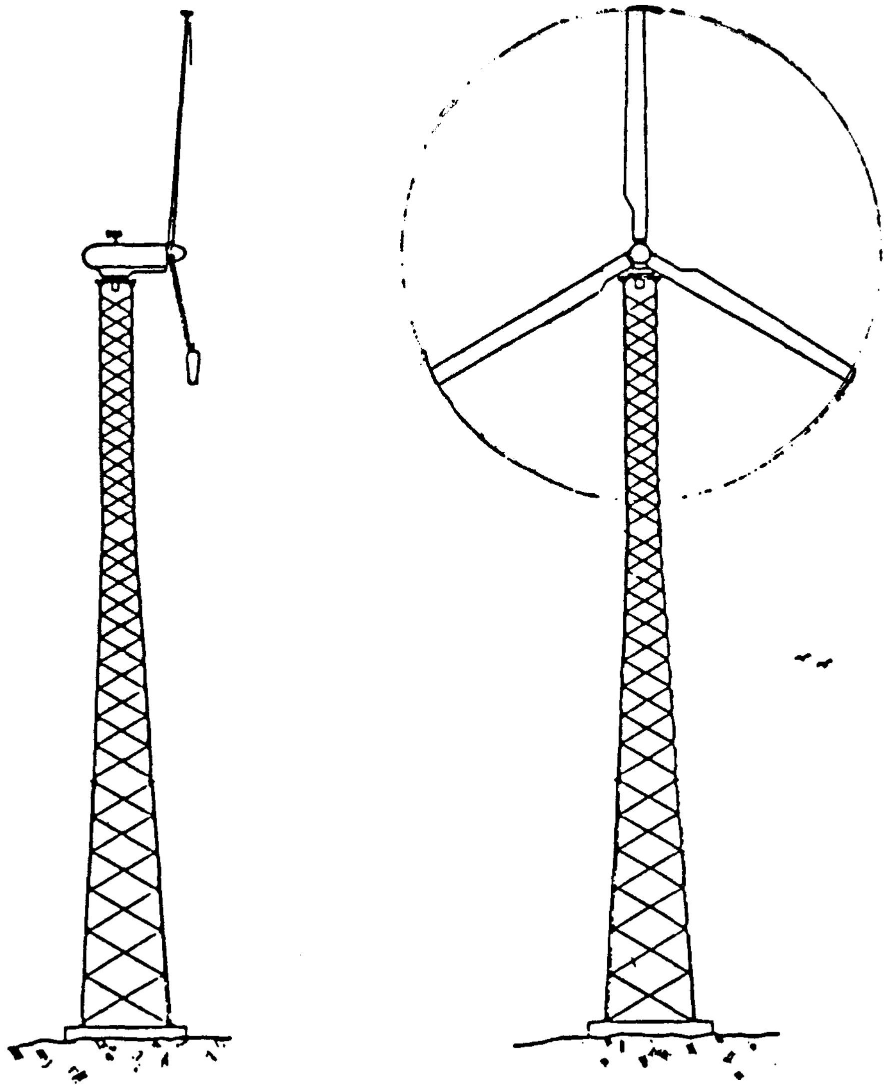


Figure 6
15 kW Final Concept

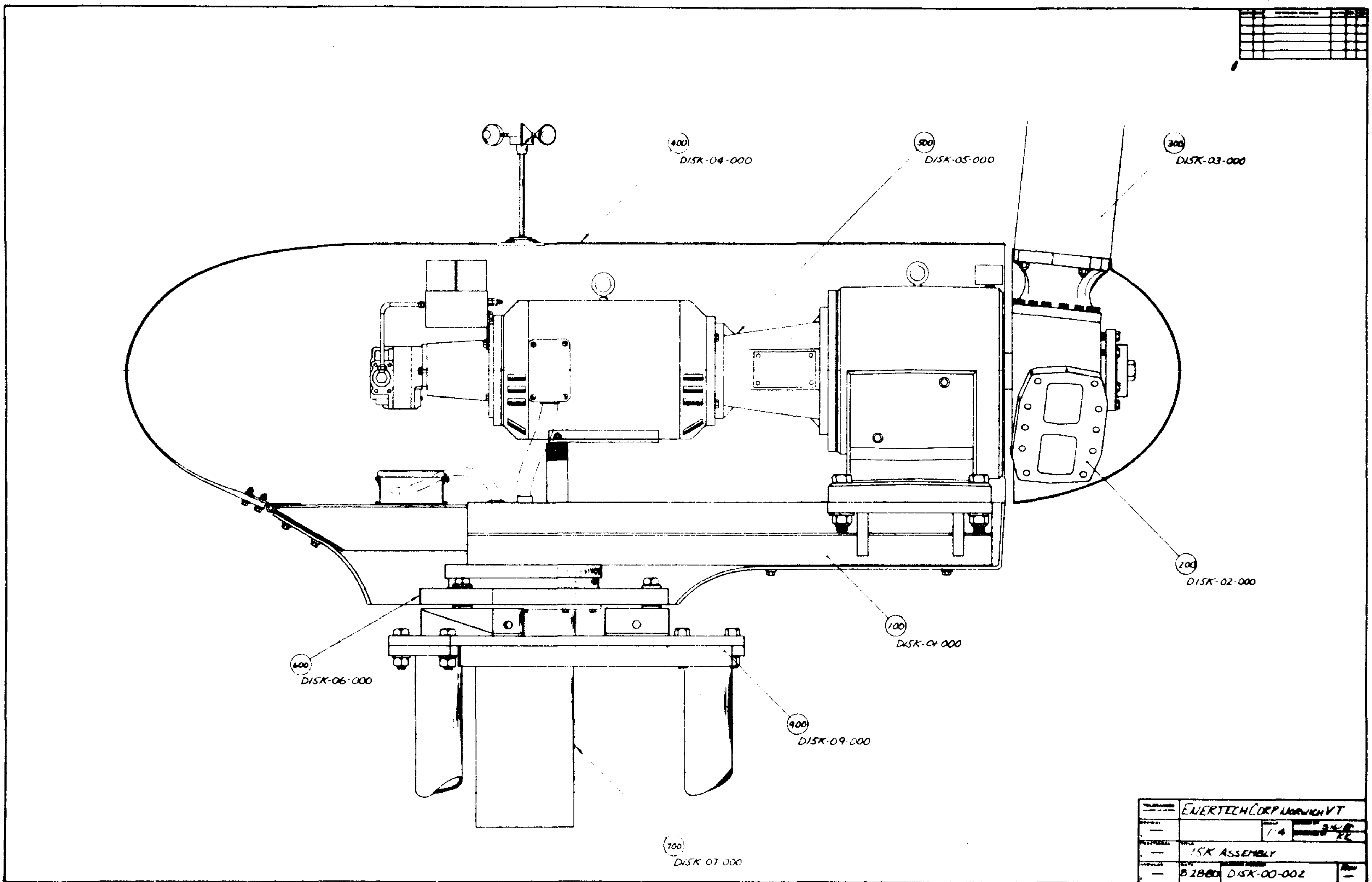


Figure 7

15 kW Integrated System After FDR

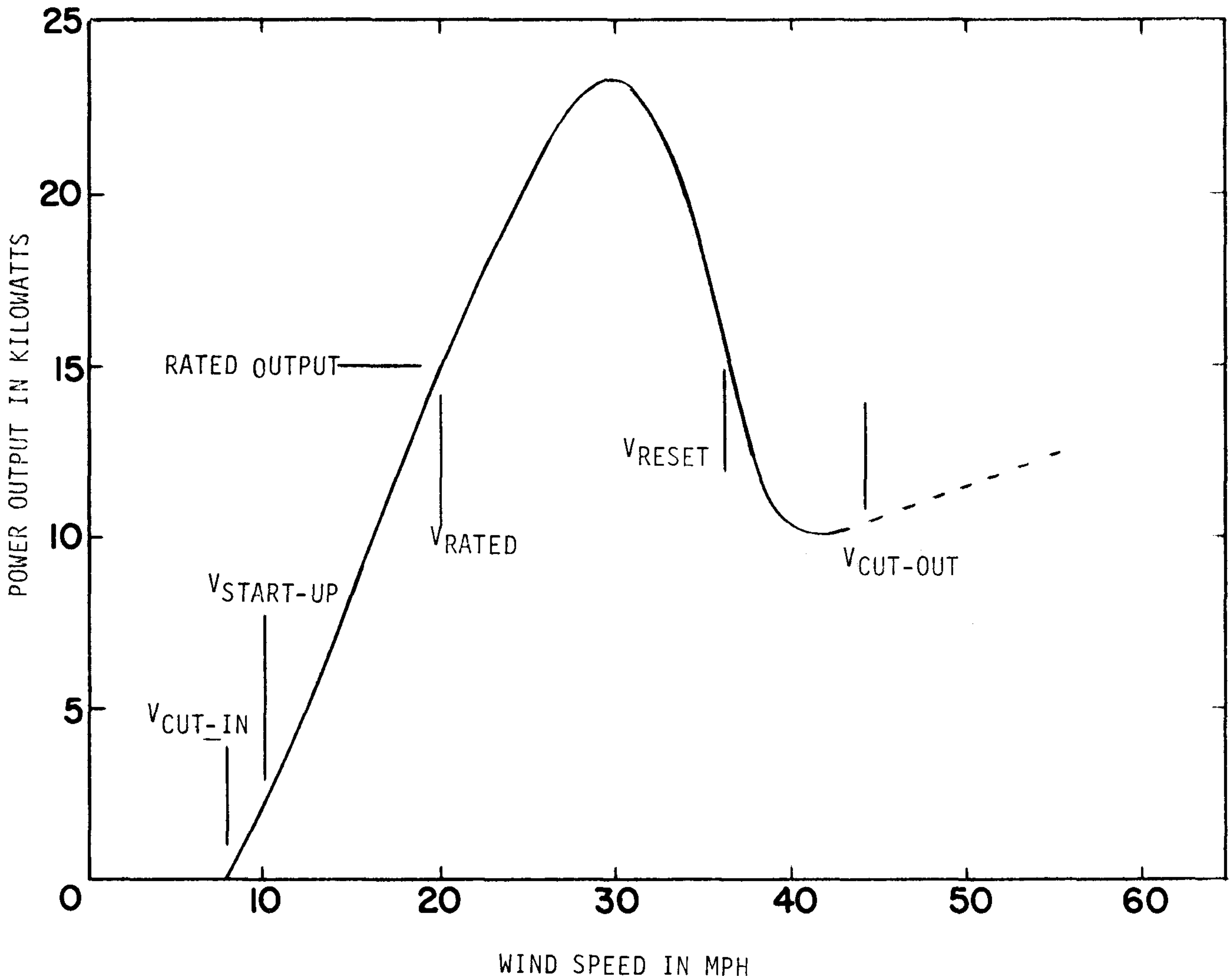


Figure 8
15 kW System Predicted Performance Characteristics

Table IV

SUMMARY OF TRADE-OFF STUDY RESULTS

	TRADE-OFF STUDY	RESULT
A.	Fixed vs Variable Pitch Rotor	Retain Fixed Pitch as proposed
B.	Number of Blades	Retain 3 blades as proposed
C.	Rotor Diameter	Retain 44 feet as proposed
D.	Rotor Solidity and Speed	Retain baseline values as proposed
E.	Free Yaw vs Damping or Control	Retain Free Yaw as proposed
F.	Control System Configuration	Tentatively retain system as proposed; some modifications may result from detailed design analysis & development
G.	Tip Speed Brakes vs Alternative Overspeed Control Methods	Retain tip brake as proposed
H.	Generator Size and Type	Use single phase Gould induction generator
I.	Blade Construction	Retain Composite Wood Blades as proposed
J.	Tower Alternatives	Retain Steel Truss tower as proposed

3.0 DETAIL MACHINE DESIGN

3.1 Rotor Design

3.1.1 Rotor Aerodynamics

Several considerations entered into the finalized design of the 15 kW rotor. Because it was decided to utilize the Gougeon laminated epoxy-wood process, great flexibility in airfoil shape and transition was afforded. Briefly the process involves: 1) forming of 2 half blade wooden molds, 2) laying up of successive epoxy-coated fir veneers into the mold, 3) evacuation of each half blade section area to compress the veneers and cure the epoxy, 4) trimming (cutting) of each half section to assure a good mating surface to the other, 5) gluing of the half sections together, 6) capping of the blade root and tip end, 7) drilling of stud holes in the blade root, and 8) insertion of epoxy-coated steel studs in the root.

Actual airfoil shapes were derived from Enertech 1500* data and Rhode St. Genese airfoil test data. The latter are shown in Figure 9. Every effort was made to maintain the desirable power and stall characteristics of the Enertech 1500 blade. Actual airfoil shapes at various blade sections were worked out with the help of Gougeon Bros. consultant Michael D. Zuteck. Figure 10 shows the airfoil cross section. Figure 11 is a lofting of the blade showing the various airfoil shapes throughout its length. The blade has 5½-degree twist and linear taper.

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* The Enertech 1500 is a three-bladed, fixed pitch machine (1.5 kW output at 9.8 m/s) which shares several features with the larger 15 kW design.

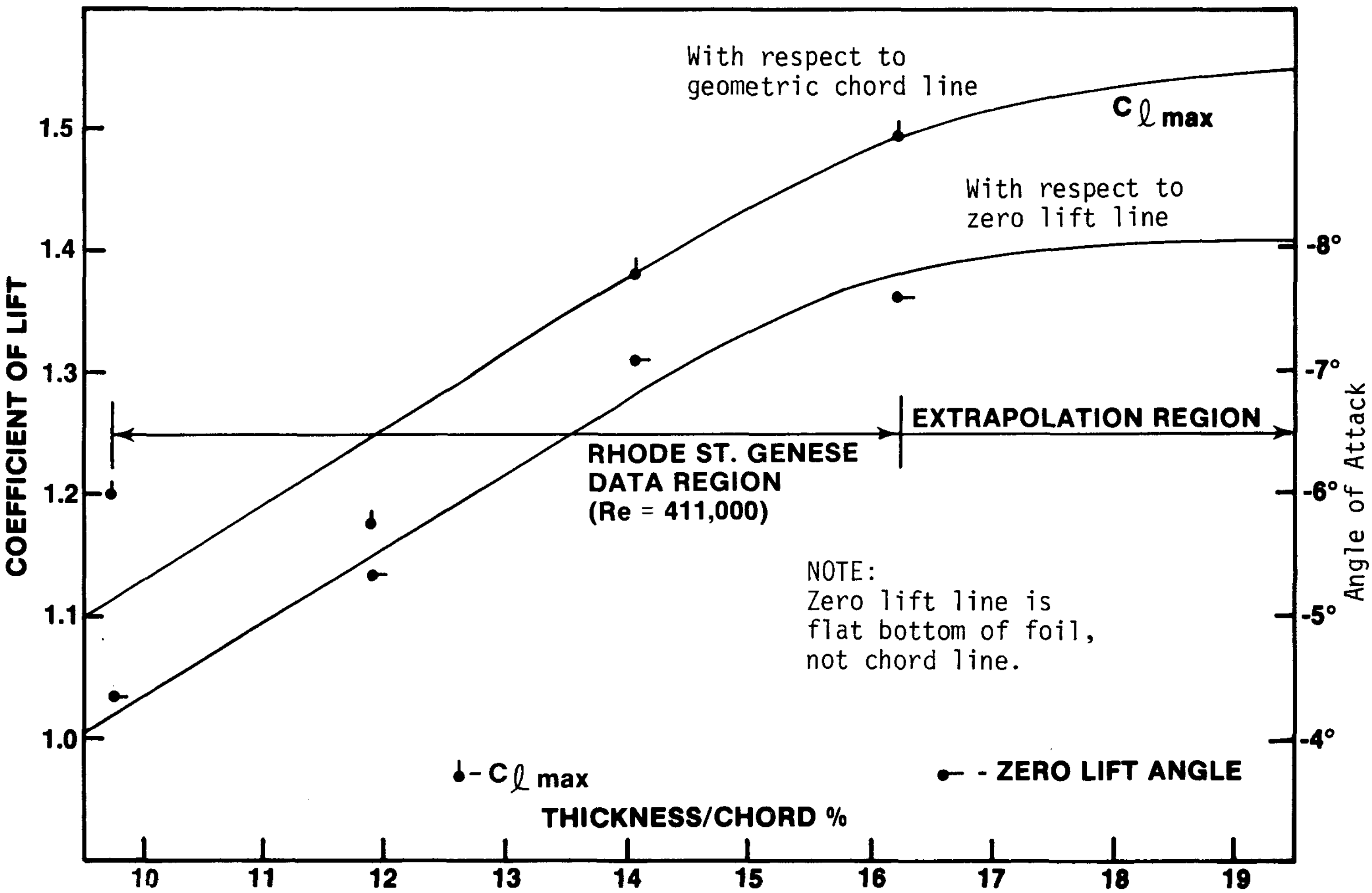


Figure 9

Droop-Nose, Flat Bottom Airfoil Data
Used in Blade Design

AIRFOIL COORDINATES BLADE TIP - CHORD : 20 INCH - THICKNESS : 10%					
STATION		UPPER SURFACE		LOWER SURFACE	
%	INCHES	%	INCHES	%	INCHES
0	0	2.50	0.50	2.50	0.50
1.25	0.25	4.57	0.915	1.15	0.23
2.50	0.50	5.45	1.09	0.82	0.16
5.00	1.00	6.72	1.34	0.35	0.07
7.50	1.50	7.56	1.51	0.15	0.03
10.00	2.00	8.21	1.64	0.03	0.05
15.00	3.00	9.13	1.83	0	0
20.00	4.00	9.71	1.94	0	0
30.00	6.00	10.00	2.00	0	0
40.00	8.00	9.75	1.95	0	0
50.00	10.00	8.99	1.80	0	0
60.00	12.00	7.83	1.565	0	0
70.00	14.00	6.20	1.24	0	0
80.00	16.00	4.32	0.86	0	0
90.00	18.00	2.47	0.47	0	0
95.00	19.00	1.54	0.31	0	0
100.00	20.00	0.62	0.125	0	0
L.E. RADIUS .25 = 1.25%					

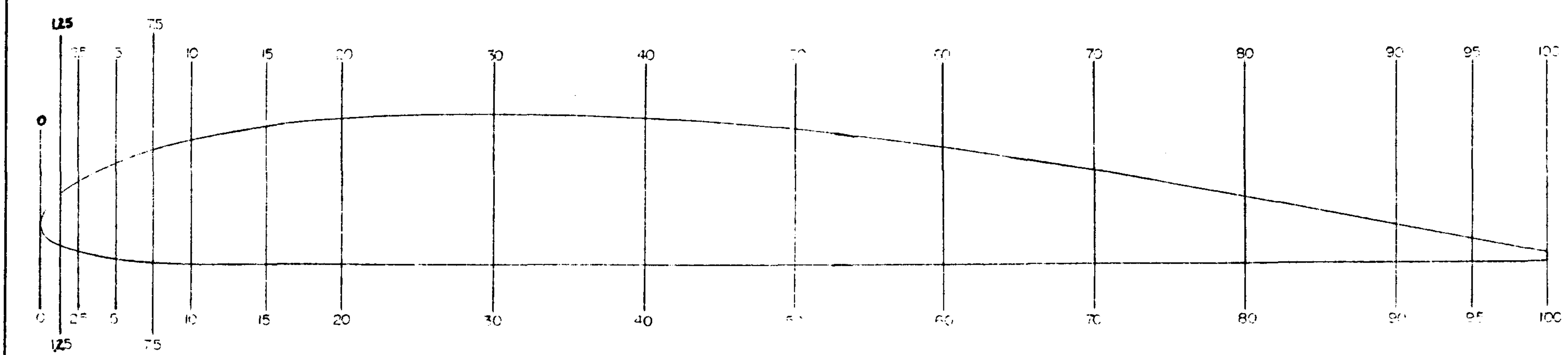


Figure 10

Airfoil Cross Section

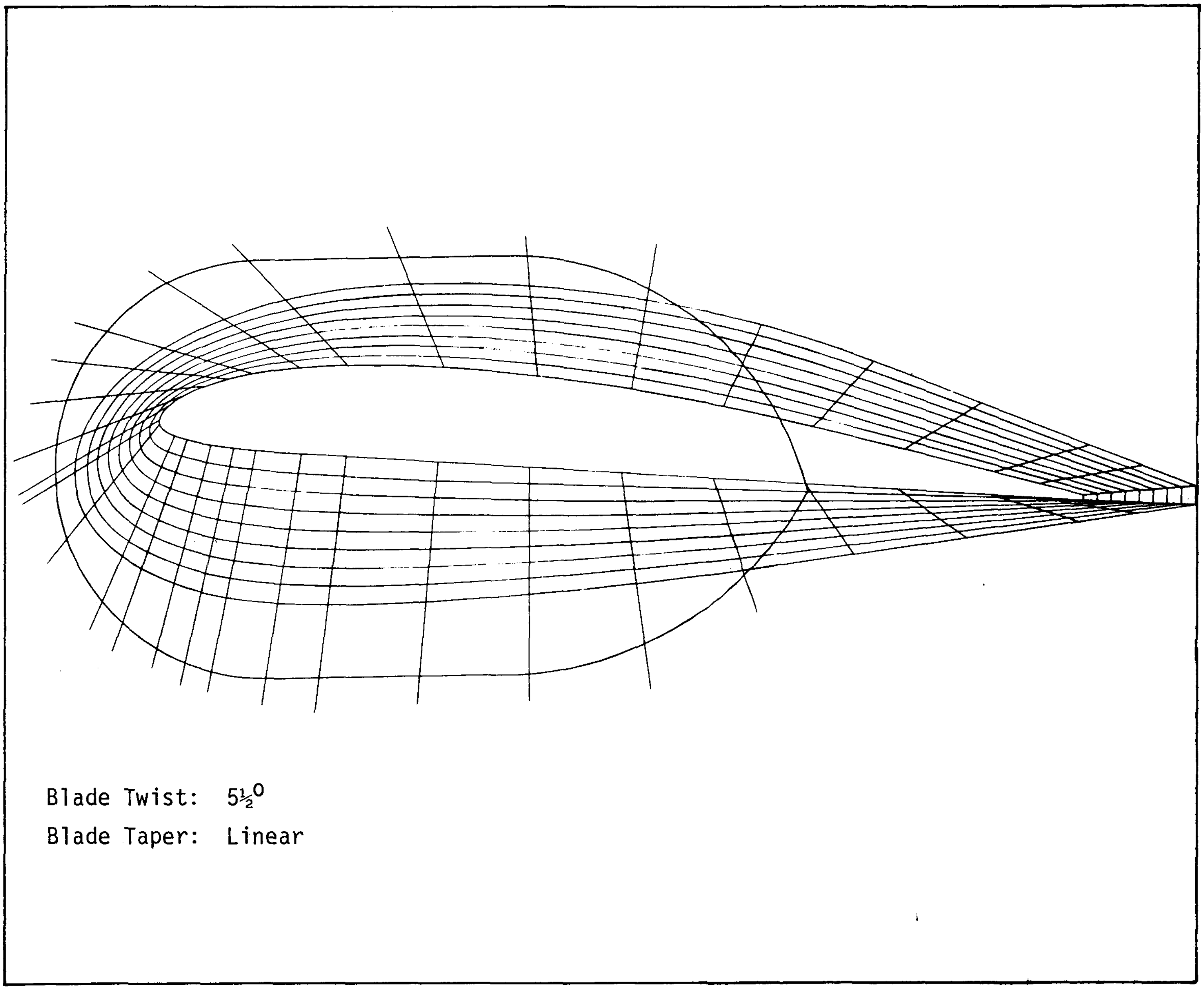


Figure 11
Blade Lofting

3.1.2 Rotor Structure/Fatigue

The blades were designed by Gougeon Brothers, Inc. of Bay City, Michigan, using specifications provided by Enertech. A first approximation of the blade shape was made by Gougeon based on an early estimate of the loads. The revised loads were calculated based on this blade shape. The maximum wind loading condition for the standard version was a gust to 56 m/s (125 mph). The maximum wind loading condition for the special version was a gust to 74 m/s (165 mph). The loading condition for fatigue was an 18 m/s (40 mph) mean wind speed with gusts, wind direction changes, tower shadow, and wind shear. A summary of the resulting loads on the machine is shown in Table V. The method used in calculating these loads is discussed below.

Blade Natural Frequencies Blade natural frequencies and mode shapes were calculated for use in determining the dynamic response of the blades to various loadings. The natural frequencies are discussed in Section 4.3.

Maximum Loads The standard design is required to withstand a maximum gust to 56 m/s. Based on data provided by Frost (Reference 1), a design gust from 31 m/s to 56 m/s was chosen. Similarly, a gust from 41.4 m/s to 74 m/s was chosen for the special design. The gust shape used was that recommended in Reference 1. The gust shape used for the standard design is shown in Figure 12 where τ is the length of the gust.

The gust was assumed to hit the blades flatwise because this results in the highest loading. Various values of the gust length, τ, were tried to determine which produced the highest loads. The first three flatwise bending modes were used in calculating the dynamic response of the blade to the gusts. Aerodynamic damping of the blade flapping was included in the analysis. Tower shadow and wind shear effects were considered in calculating the moments transferred to the gearbox.

Fatigue Loads The fatigue design loading was based on a mean wind speed of 18 m/s with wind direction changes of 30° and gusts. The rotor is spinning at 54 rpm. The effects of wind shear and tower shadow were considered.

Table V

Summary of Critical Loads

1. Maximum Loads - Standard Design
31 - 56 m/s gust, rotor stopped

Blade root bending moment	30,700 ft-lb per blade
Thrust - three blades exposed	7,900 lb
Gearbox Moment - one blade shadowed	18,900 ft-lb
Maximum deflection at tip of blade	1.1 ft

2. Maximum Loads - Special Design
41.5 - 74 m/s gust, rotor stopped

Blade root bending moment	53,400 ft-lb per blade
Thrust - three blades exposed	13,700 lb
Gearbox Moment - one blade shadowed	32,900 ft-lb

3. Fatigue Loads
18 m/s mean wind speed, gusts up to 22 m/s and down to 14.3 m/s, 30° wind direction changes, tower shadow and wind shear, rotor turning at 54 rpm

Blade root moment	4400 ± 5500 ft-lb
Thrust on gearbox shaft	1400 ± 640 lb
Moment transmitted to gearbox	980 ± 4750 ft-lb
Bending moment in gearbox shaft	±5730 ft-lb

4. Operating Loads

Driving torque at hub (18 kW)	3380 ft-lb
Braking torque at hub (125 ft-lb brake)	4450 ft-lb
Centrifugal pull on one blade (54 rpm)	2200 lb

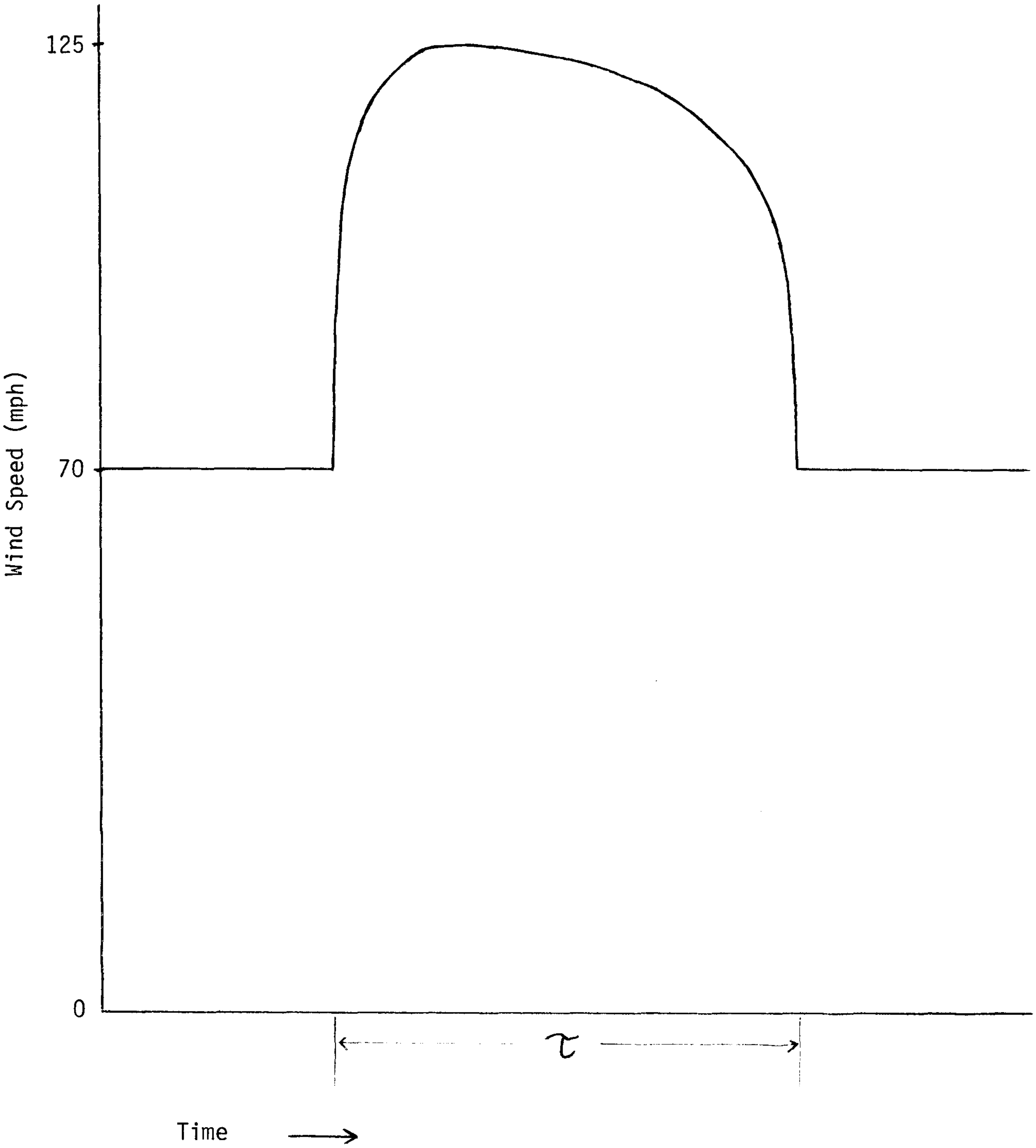


Figure 12
Gust Shape Used in Design

1. Mean wind speed - A mean wind speed of 18 m/s was chosen for the fatigue design condition because this is the highest mean wind speed at which the machine would normally be running.

2. Gusts - Gusts up to 22 m/s and down to 14.3 m/s were chosen. These represent second standard deviation gusts from the mean wind speed of 18 m/s (ref. 2). The dynamic response of the blades to these gusts was considered.

3. Yaw - An elementary yaw analysis was performed to determine the maximum loads on the machine due to wind direction changes. The analysis included gyroscopic effects of the spinning rotor and aerodynamic yaw damping. Aerodynamic moments (due to cross-flow) tending to tilt the rotor up or down were determined analytically. Aerodynamic moments tending to yaw the machine were scaled up from the values measured in the yaw tests on the Enertech 4 kW prototype wind machine.

4. Wind Shear - A design wind shear profile was determined based on the recommendations in Reference 1. During every revolution of the rotor, each blade sees a cyclic force due to the wind shear. The dynamic response of the blades to these forces was calculated.

5. Tower Shadow - The tower shadow effect at the rotor was modeled as a 20% wind velocity reduction occurring over 24° of rotor travel as the blade passes behind the tower. The shape of the tower shadow forcing function was assumed to be a square wave. Each blade experiences this tower shadow excitation once per revolution. The dynamic responses resulting from this excitation were calculated using a method described in Reference 3.

After the responses of the individual blades were calculated, the loads due to the individual blades were combined to give the total loads on the rotor.

3.1.3 Aeroelastic Analysis

An aeroelastic analysis of the Enertech 15 kW rotor was performed by Dr. Norman D. Ham of the Massachusetts Institute of Technology. The flutter boundaries resulting from this analysis are shown in Figure 13. The analysis was done for three chordwise center of gravity (C.G.) locations: 35%, 40%, and 45%. According to Dr. Ham, three types of flutter are responsible for the shape of the flutter diagram. The nearly horizontal part of the flutter on the left-hand side boundary represents classical flutter which involves coupled flatwise bending and torsion of the blade. The nearly vertical dip represents flutter due to coupled edgewise bending and torsion of the blade. Dr. Ham stated that he had never witnessed this type of flutter. The third type of flutter, represented by the right-hand part of the curve sloping up and to the right, is stall flutter.

The flutter boundary for the blade C.G. at 45% of the chord represents the blade as specified at CDR. This blade was shown to be free of flutter and divergence in the operating range (50.5 - 52.8 rpm). However, in an overspeed condition, the tip brakes had been specified to deploy at 75 rpm. This means that there could be possible flutter problems if overspeeding occurred when the wind was between approximately 13 and 17.4 m/s.

To avoid this possibility, the blade chordwise C.G. has been moved to the 40% position and the tip brake deployment has been lowered to 70 rpm (see Figure 14). In this way, the blade will always be below the flutter boundary, even in the overspeed condition. Also, 70 rpm is sufficiently above the operating speed so that spurious deployment is unlikely. The blade fabricators will achieve the 40% C.G. location by repositioning laminates from the tail to the nose.

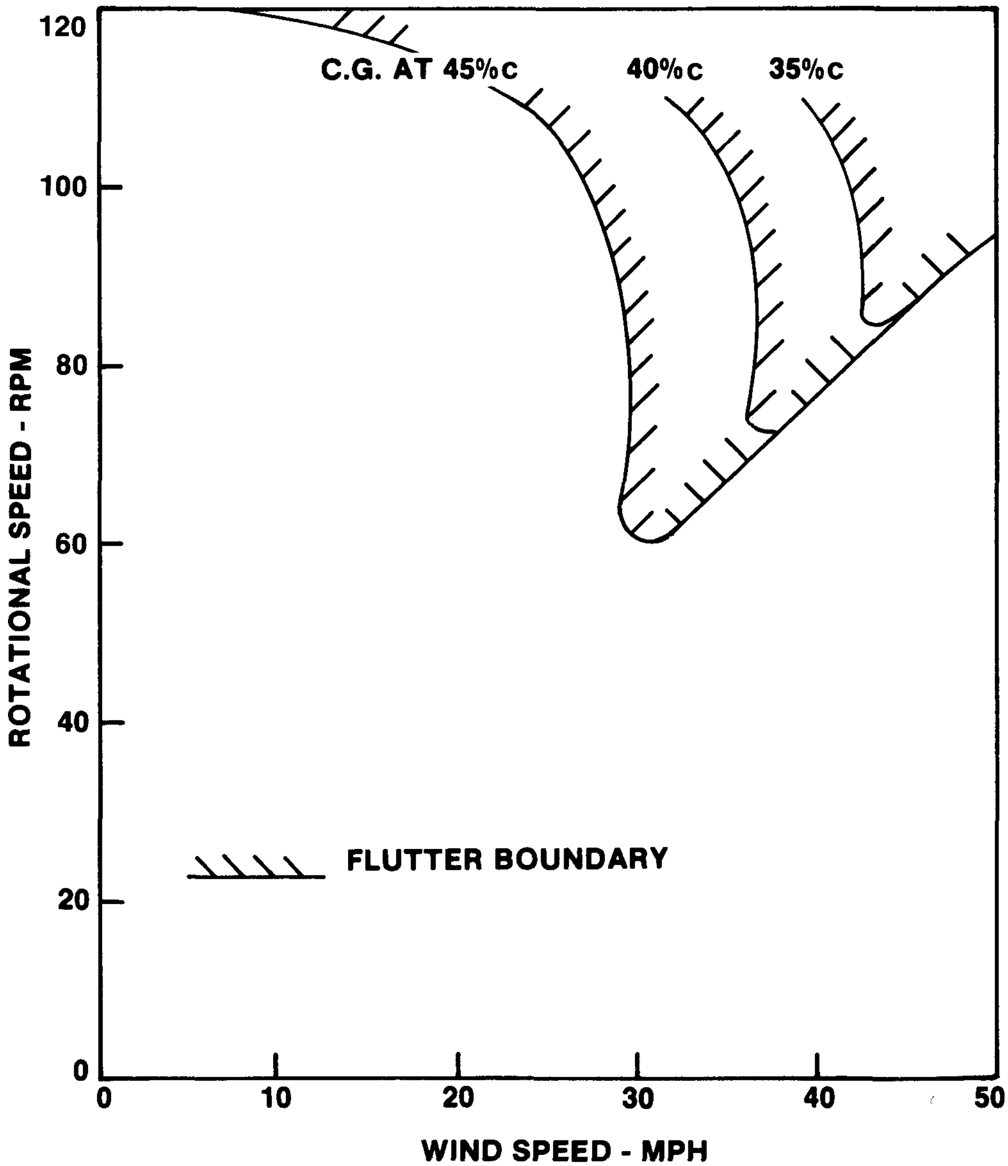


Figure 13

Flutter Diagram - Enertech 15 kW Blade

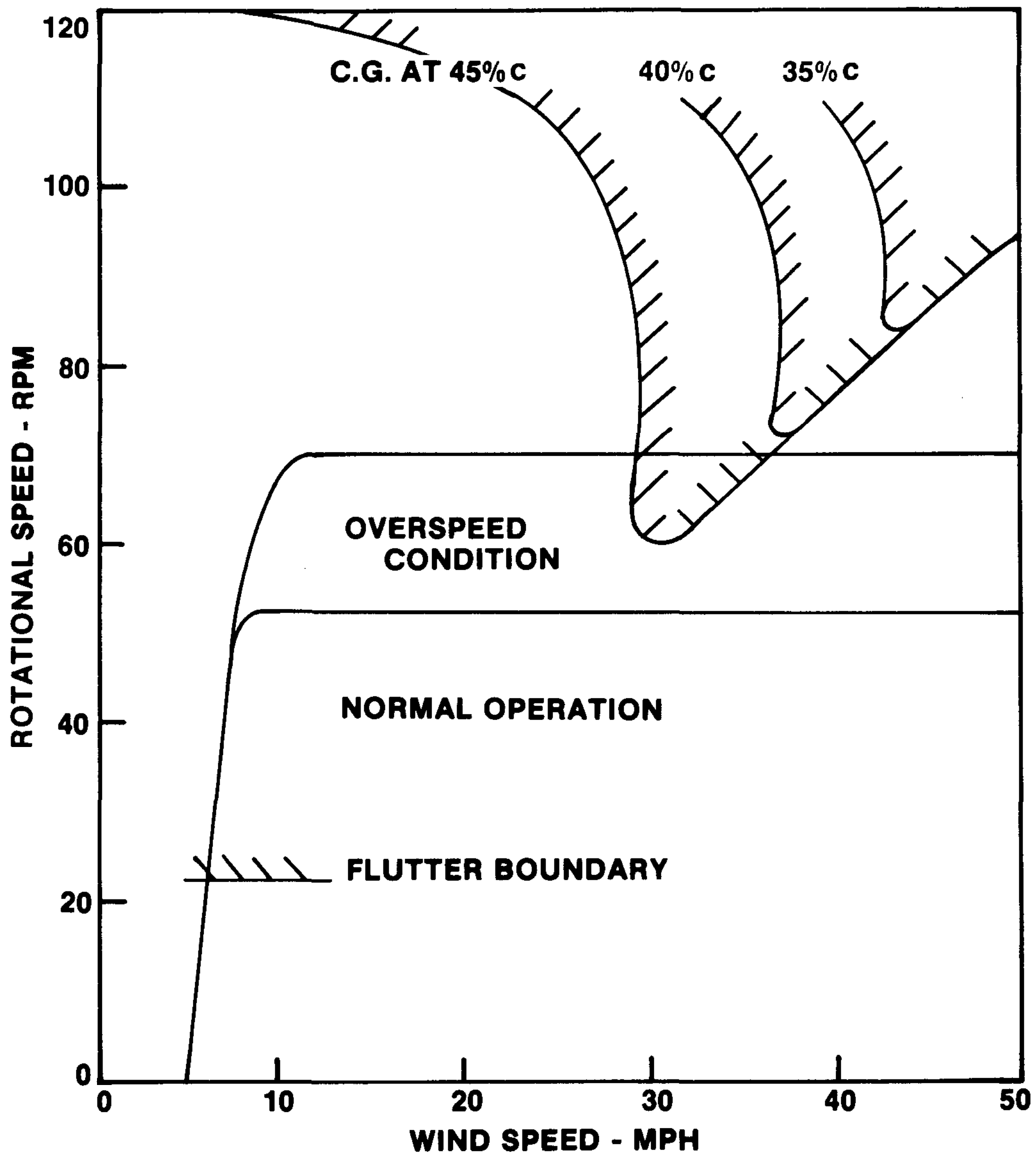


Figure 14

Flutter Diagram - Enertech 15 kW Blade

3.2 Mechanical Subsystem Design

3.2.1 Hub Assembly

The hub assembly is a four piece casting consisting of a central hub attached to the gearbox shaft and three legs which are bolted to this hub and carry the blades (see Figures 15-18). The legs are bolted on with a circular bolt pattern which allows pitch changes for various average air densities. By slotting the holes in the prototype hub, pitch changes can be made on the installed unit for performance evaluation. The material selected is AISI 8632. This alloy is easily cast and is heat-treatable to 160 KSI tensile strength levels. It is metallurgically similar to AISI 4130, having .25% carbon. Properties of AISI 8632 are shown in Table VI.

The central hub is attached to the gearbox shaft with a tapered bushing. The bushing is keyed to the shaft and the hub, and is wedged in place by six one-half inch bolts. The hub is further secured by a one-inch left hand thread bolt, threaded into the end of the gearbox shaft, and a thick washer. Tower shadow creates a small alternating moment which tends to wiggle the hub loose from the shaft, bushing and hub. The tendency for this beating down is proportional to the bearing stress given by:

$$\delta_M = \frac{12M}{\pi RL}$$

where:
- M = tower shadow moment
- R = radius of shaft
- L = length of tapered bushing

The bearing stress for the 15 kW is only one seventh that of the Enertech 1500 (which has never had any problems of this type) and is therefore a very conservative design in this respect.

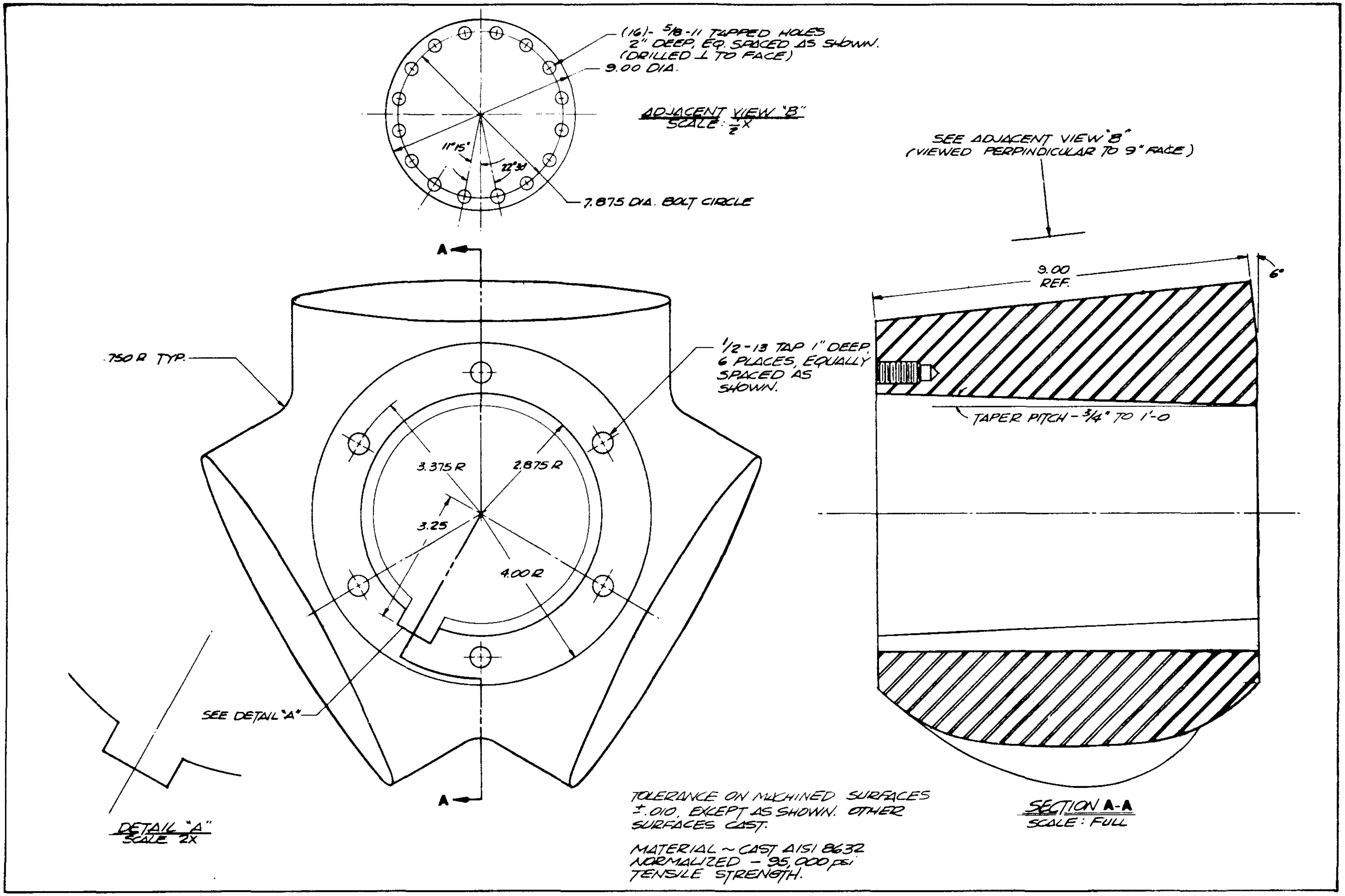
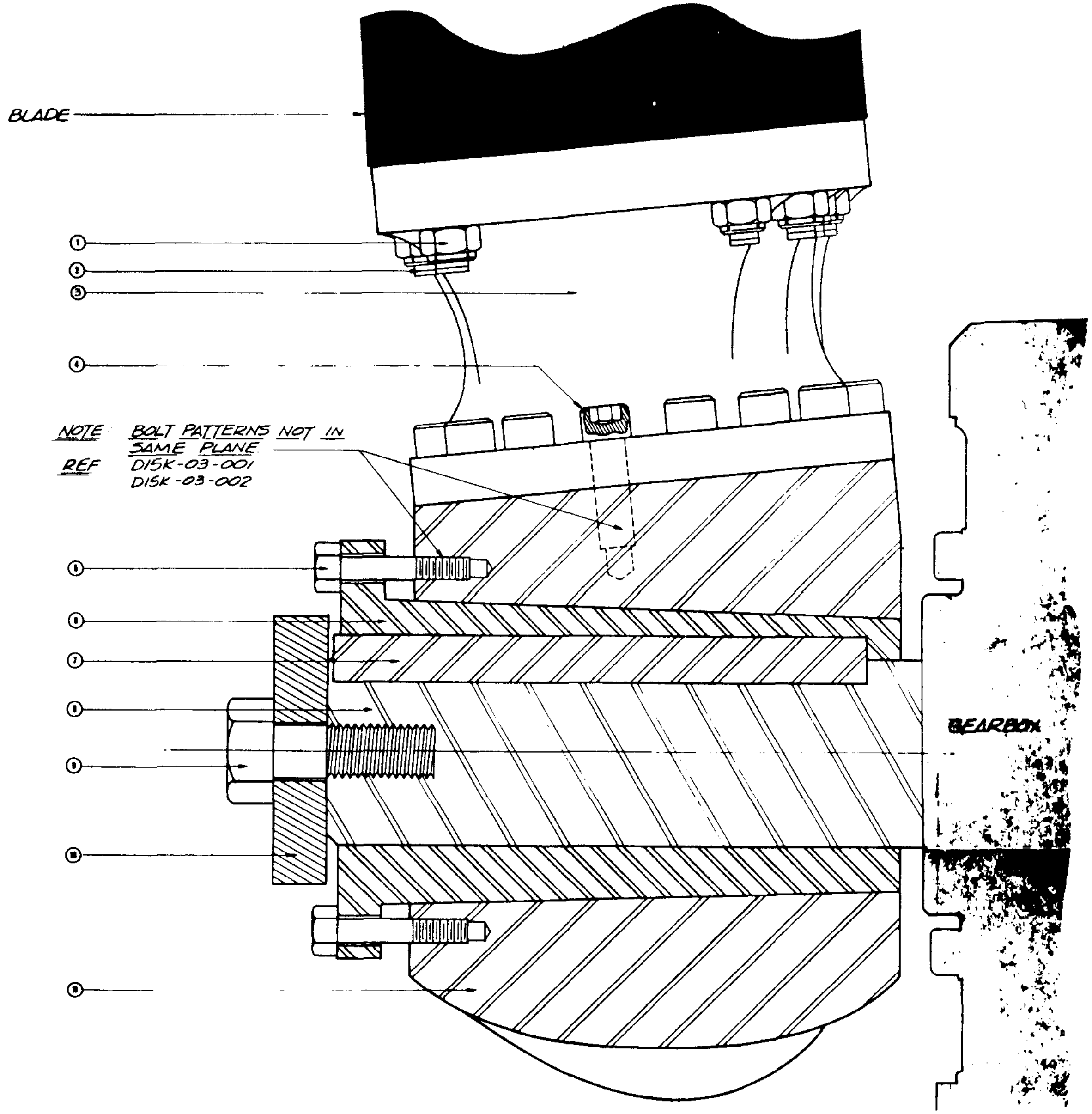


Figure 15

Main Hub



ITEM	PART #	QTY.	DESCRIPTION
1		10	5/8-18 UNJF-3B HEX LOCKNUTS
2		10	5/8-18 STUDS
3		3	HUB ARM
4		16	5/8-11 SOCKET HEAD CAPPED SCREW
5		6	1/2-13
6		1	TAPERLOCK BUSHING
7*		2	KEY
8		1	3 1/2" SHAFT
9		1	1" DIA x 3" L L.H. BOLT
10		1	1" WASHER
11		1	HUB

* 2 SEPARATE KEYS ARE REQUIRED FOR ASSEMBLY. SEE DWG DISK-03-001 - FOR LOCATION AND DIMENSION

Figure 16

Hub Assembly

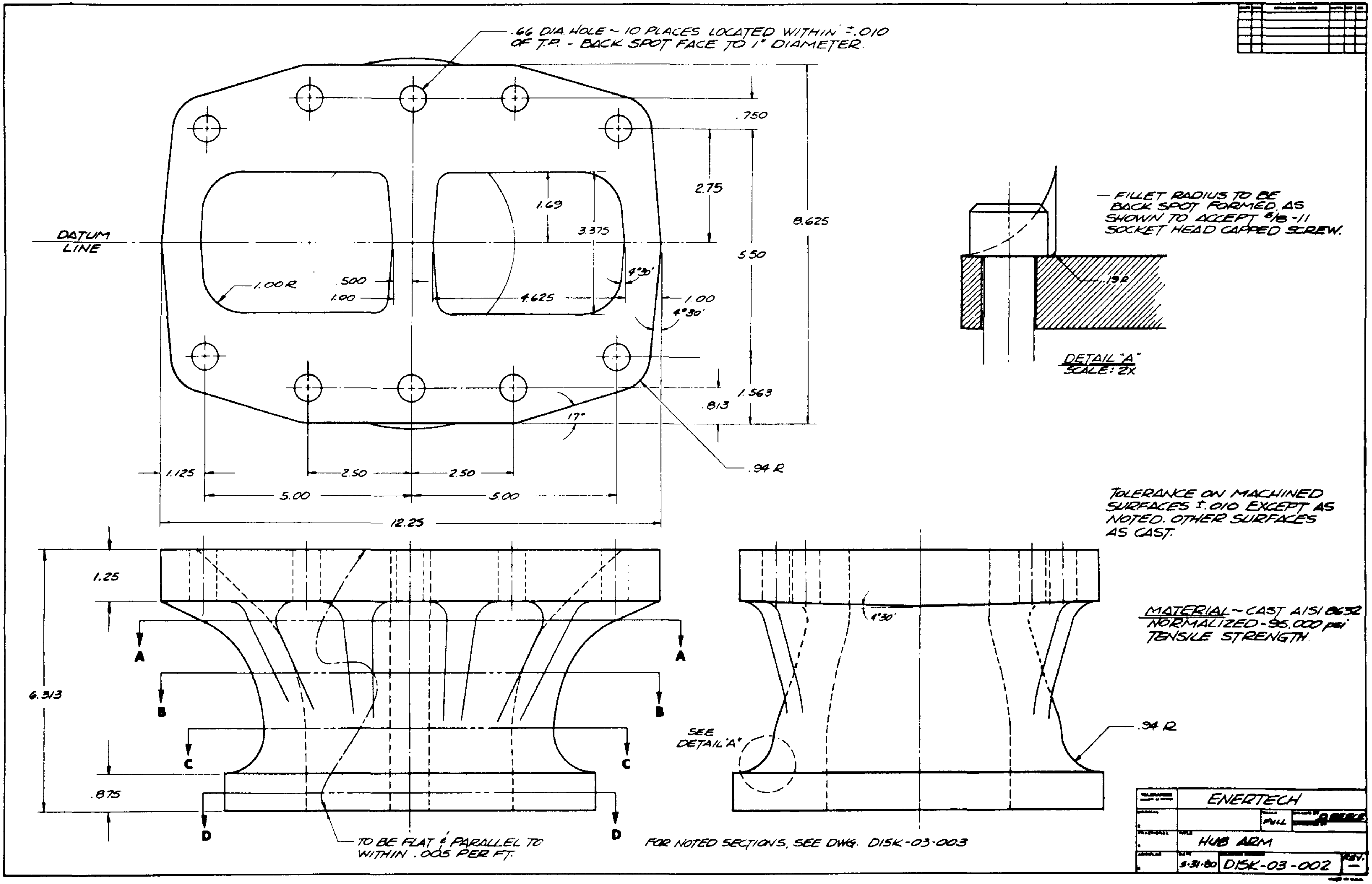


Figure 17

Hub Arm

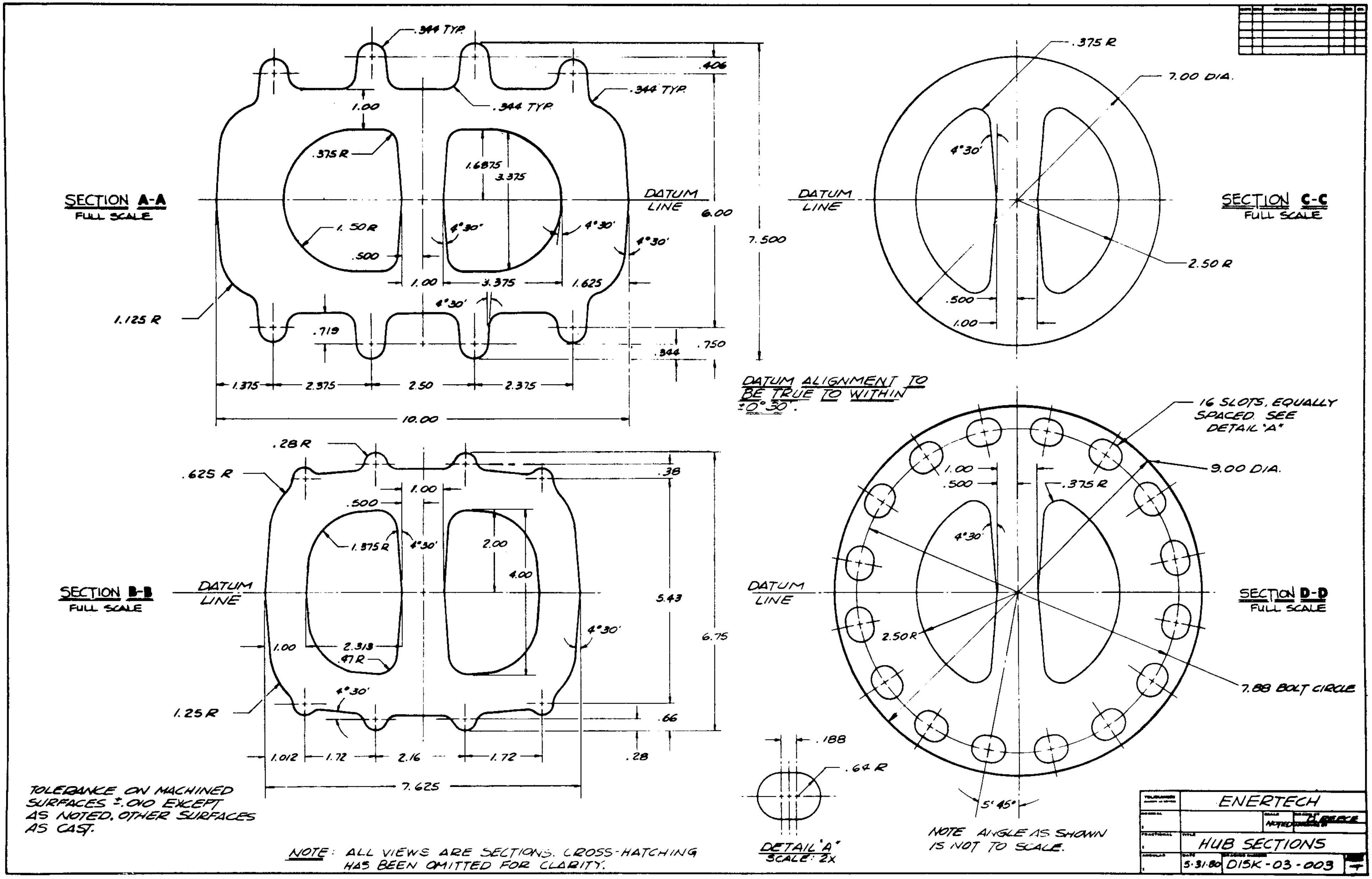


Figure 18

Hub Sections

Table VI

Cast AISI 8632 Properties*

	Normalized	QtT at 800°F
Yield Strength (psi)	86,000	160,000
Tensile Strength (psi)	95,000	180,000
Elongation (%)	16	12
Reduction in Area (%)	52	60
Smooth Endurance 10^7 x (psi)	54,000	85,000*T
Notched Endurance 10^7 x (psi) (K_υ = 2.2)	33,000	50,000*T

* (Sources - Ref. 6 & 7.)

T* These values were estimated from data on normalized and QtT at 1200°F.

For small bushing taper angles the normal load on the hub or shaft is given by:

$$N = \frac{P}{\text{Tan}\phi + \mu}$$

P = axial assembly load - lb
ϕ = taper angle relative to center line - degrees
μ = coefficient of friction between the taper lock bushing and the hub

By experiment with the Enertech 1500, μ was found to be about .10. This assumes that the bushing is torqued and rapped into position in accordance with normal assembly procedure.

The load to pull the assembled hub off of the shaft is given approximately by:

$$P^1 \approx \mu^1 N$$

where: μ^1 = the coefficient of friction between the bushing and shaft. Again, by experiment, μ^1 = .19.

A safety factor for hub pull-off can be defined as: P^1/Thrust. For the Enertech 1500 this number is 12.0; for the 15 kw, 16.0.

The 74 m/s design will require 8 bushing bolts to bring the safety factor up to 12.0 or 10 bolts to bring the safety factor up to 16.0. These comparisons are tabulated in Table VII. It can be seen from this Table that the 15 kW is a slightly more conservative design than the 1500 in the attachment area.

The gearbox shaft torsional shear stress and key stresses are tabulated in Table VIII. Again it can be seen that the 15 kW is slightly more conservative than the 1500, which has had no problems in this area.

The central hub and legs and the various attaching bolts were analyzed for all of the loads shown in the loads table and all possible combinations, including gravity on the blades. All areas of stress concentration were checked for fatigue. The bolt preloads were set so as to prevent flange separation under all conditions and checked for fatigue of the threads and any notched areas.

Table VII

Taper Lock Bushing Load Comparison

	E 1500	15 kW
Radius of Shaft R (in)	.625	1.75
Length of Taper Lock L (in)	1.875	9.0
Tower Shadow Moment M (in-lb)	740	6700
Bearing stress δ_M (psi)	1300	180
Bushing Normal Load N (lb)	45,800	687,000
Assembly Bearing Stress δ (psi)	6200	6900
Pull Off Load P^1 (lb)	8700	130,000
Thrust (56 m/s, stopped - lb)	723	7900
Safety Factor (P^1/Thrust)	12.	16.

Table VIII

Comparison of Shaft and Key Stresses

	E 1500	15 kW
Gearbox shaft torsional shear stress - max. continuous output	6260	4825
Key shear stress - max. continuous output	10,200	4400
Key shear stress - braking	14,500	5800
Key bearing stress - braking	29,000	11,600

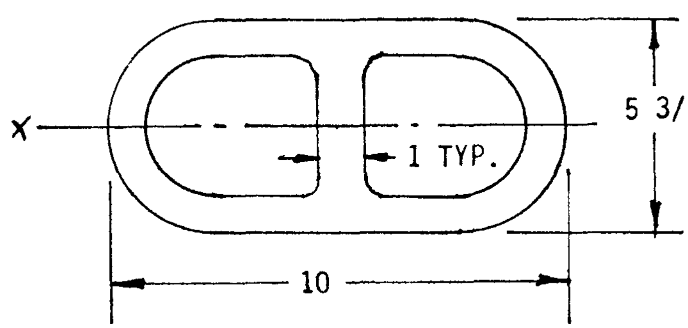
Typical sample calculations follow.

1. Blade attachment flange of legs:

It is assumed that the 56 m/s (125 mph) gust load acts perpendicular to the chord line at the root which is the same as the major axis of the leg at that point. The most highly stressed area is just below the flange. The moment at this point is:

368,400	in-lb aerodynamic
+ 2,850	in-lb blade weight
371,250	in-lb total

The moment of inertia at this point is:

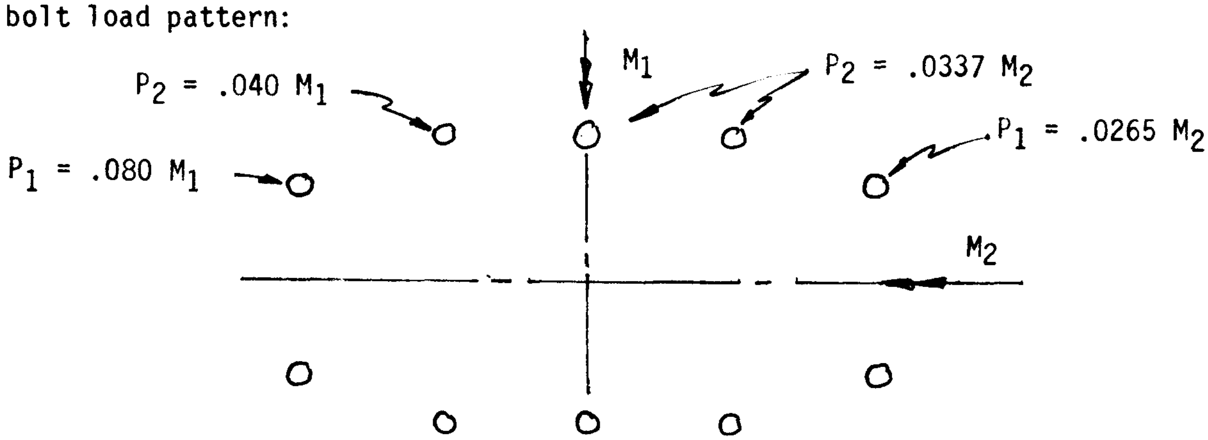


$$I_x = \frac{\pi}{64}\left[(5\ 3/8)^4 - (3\ 3/8)^4\right] + \frac{4\ 5/8\ (5\ 3/8)^3}{12} - \frac{3\ 5/8\ (3\ 3/8)^3}{12}$$

$$= 106\ \text{in}^4$$

$$\delta = \frac{MC}{I_x} = \frac{371{,}250}{106}\left(\frac{5\ 3/8}{2}\right) = 9400\ \text{psi}$$

Taking the moment of inertia of the bolt pattern provides the following bolt load pattern:

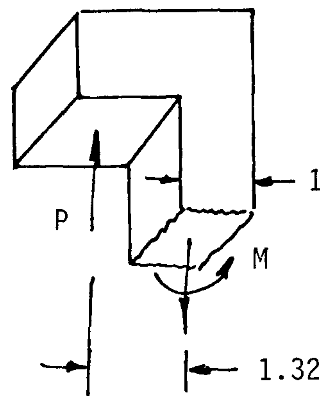


It can readily be seen that the most highly loaded section is at the center bolt. The gussetts are neglected for stress analysis and a beam of one pitch width is taken as carrying the bolt load:

$$P = (.0337)\ 371{,}250$$
$$= 12500$$

$$I = \frac{2\ 1/2\ (1)^3}{12} = .208$$

$$\delta = \frac{MC}{I} = \frac{12500\ (1.32)\ .5}{.208}$$
$$= 39{,}700$$



The total stress at the upwind corner is:

39,700 + 9400 = 49,100

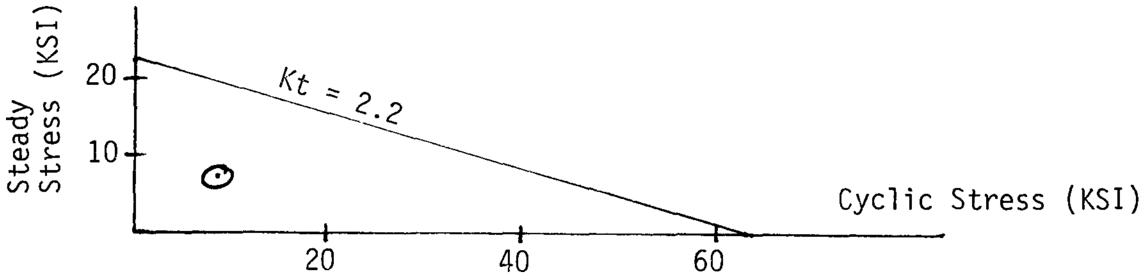
or Safety Factor (SF) = 1.75

This analysis is conservative in that it neglects the gussetts and the plate effect and assumes a pinned boundary condition at the bolt, whereas that point has some fixity due to the stud shoulder. It is estimated that the actual stress at that point might be as low as one half of the calculated value.

Similarly for the 18 m/s (40 mph) yawing condition the stress at this point is:

$$= 6980 \pm 8700$$

The stress concentration in the corner of the flange is 1.88 (Ref. 4). The stresses are plotted on the Goodman diagram below. The point is well below the K_t = 2.2 line and therefore is adequate.



The remainder of the leg, including the circular flange, was analyzed in a similar fashion and the results are summarized in Figure 19. For the 56 m/s (125 mph) design the casting can be used in its normalized condition, but for the 74 m/s (165 mph) design, it should be quenched and tempered at 800°F.

2. Blade Attaching Studs:

The blade attaching studs are made of A1S1 4340, tempered to 140 KSI tensile strength. The end of these studs, which protrudes from the blade, is 5/8-in diameter and has a 5/8 fine thread. The blade is secured by SPS heavy duty "Flexloc" nuts and Locktite®.

From the previous stud load analysis, the most highly loaded stud for 56 m/s is P_2 = 12,500 lb. For 18 m/s yawing the worst bolt is:

$$P_2 = (52{,}800 \pm 66{,}000)\ .0337 = 1780 \pm 2220 \text{ lb.}$$

plus driving torque (10,890) .040 =	435
Total Cyclic Loads	2215 ± 2220

The preload for these studs was selected at 18,400 lb. This preload is the highest that can be used and still ensure that the loads in the studs will be maintained less than the yield load, divided by 1.5. Based on test data, the coefficient of friction for clean threads and wet Locktite® is .20. Using this friction factor, the studs should be torqued to 238 ft-lb. Since this torque may be difficult to achieve on the tower top, the "angle-of-turn" method will be used to preload these studs.

The critical load, or flange separation load, is given by:

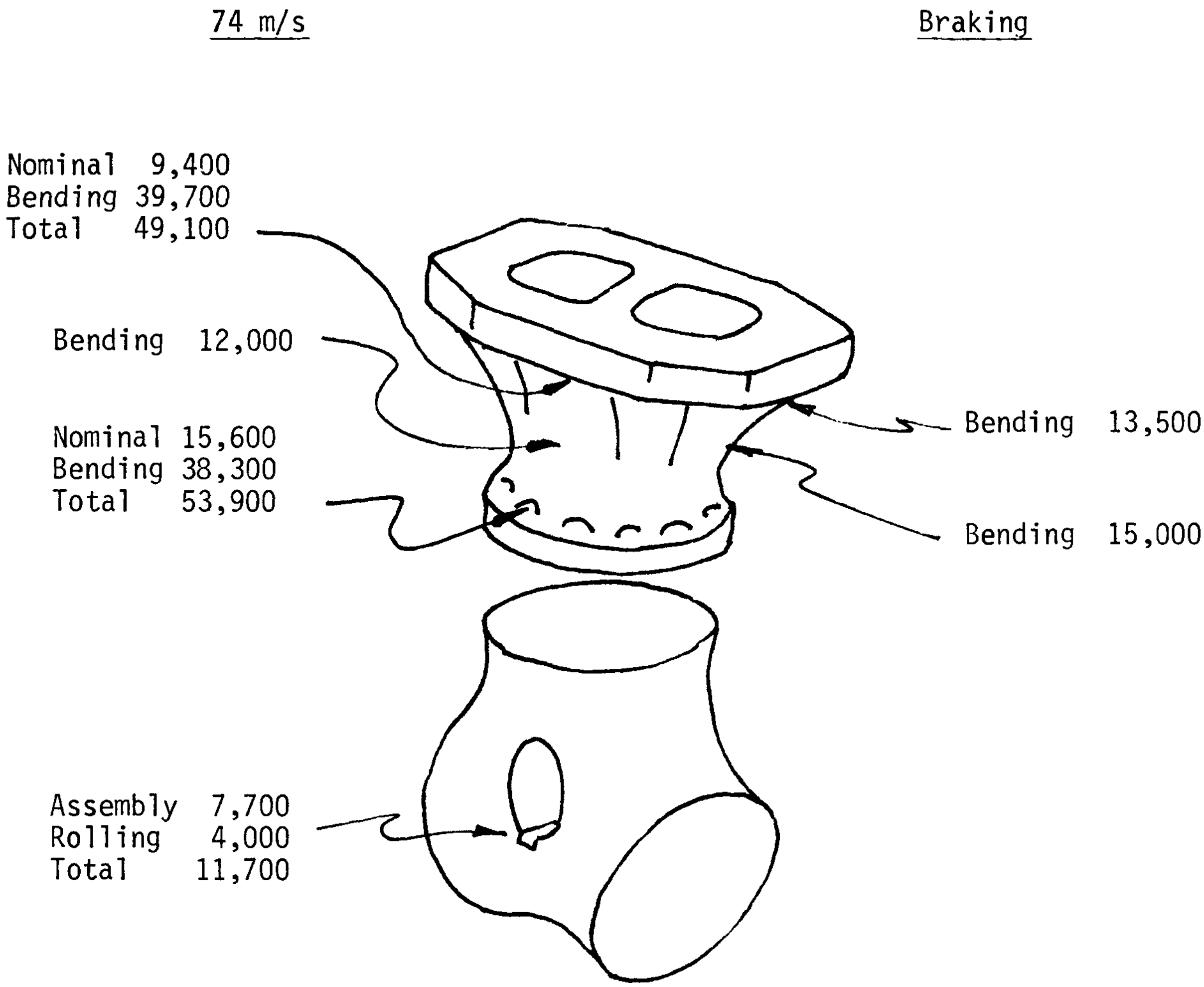


Figure 19
Summary of Hub Stresses

$$Fc = \frac{K_b + K_m}{K_m} P_bp$$

K_b = bolt spring rate, lbs/in

K_m = clamped material spring rate,

P_bp = bolt preload, lb.

$$Fc = \frac{7.24 + 45.8}{45.8} \; 18{,}400 = 21{,}300$$

Therefore, the margin on separation is:

$$\frac{21{,}300}{12{,}500} = 1.70$$

The load in the bolt for an externally applied load F, is given by:

$$P_b = \frac{Kb}{K_b + Km} F + P_bp$$

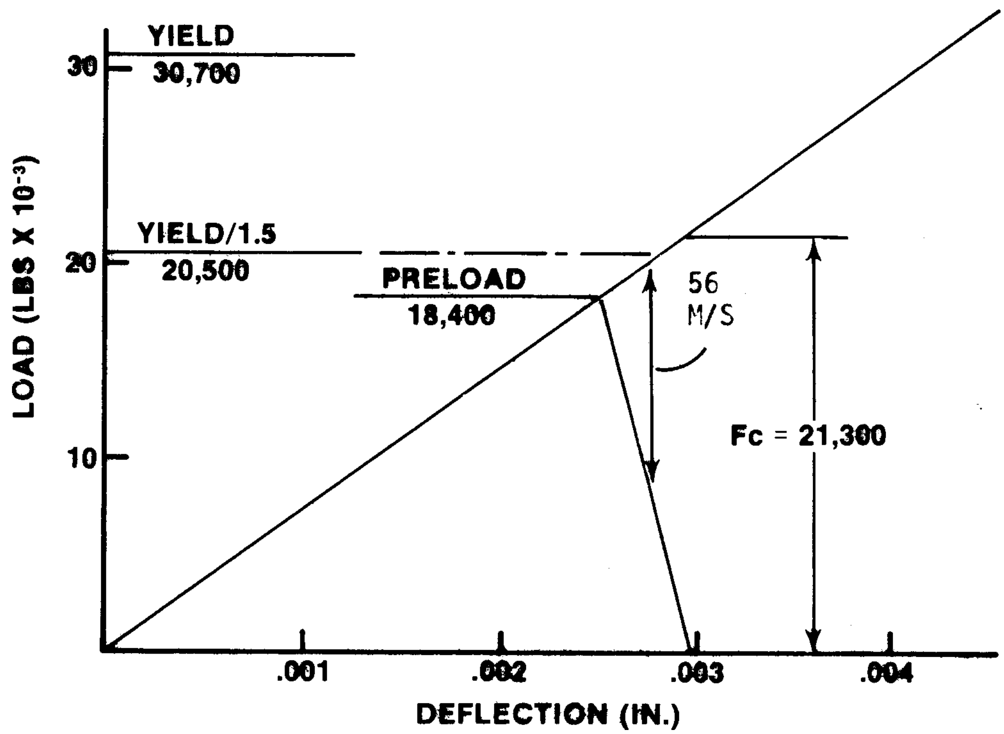
$$= .136F + 18{,}400$$

for 56 m/s:

$$P_b = .136\ (12{,}500) + 18400$$

$$= 20{,}100 \text{ lb.}$$

This analysis is presented graphically below:



The worst condition for fatigue of the studs is 18 m/s yawing.

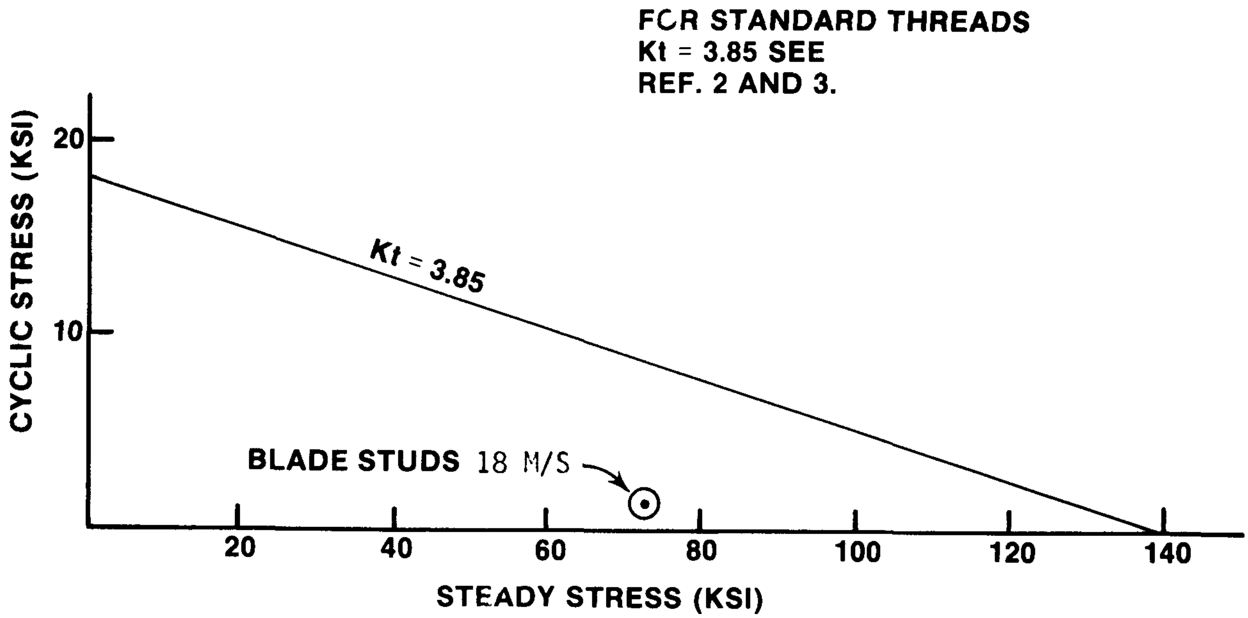
$$P_b = .136\ (2215 \pm 2220) + 18,400$$

$$= 18,700 \pm 300$$

or, the nominal stress in the threads is

$$\delta = 73,000 \pm 1170 \text{ psi}$$

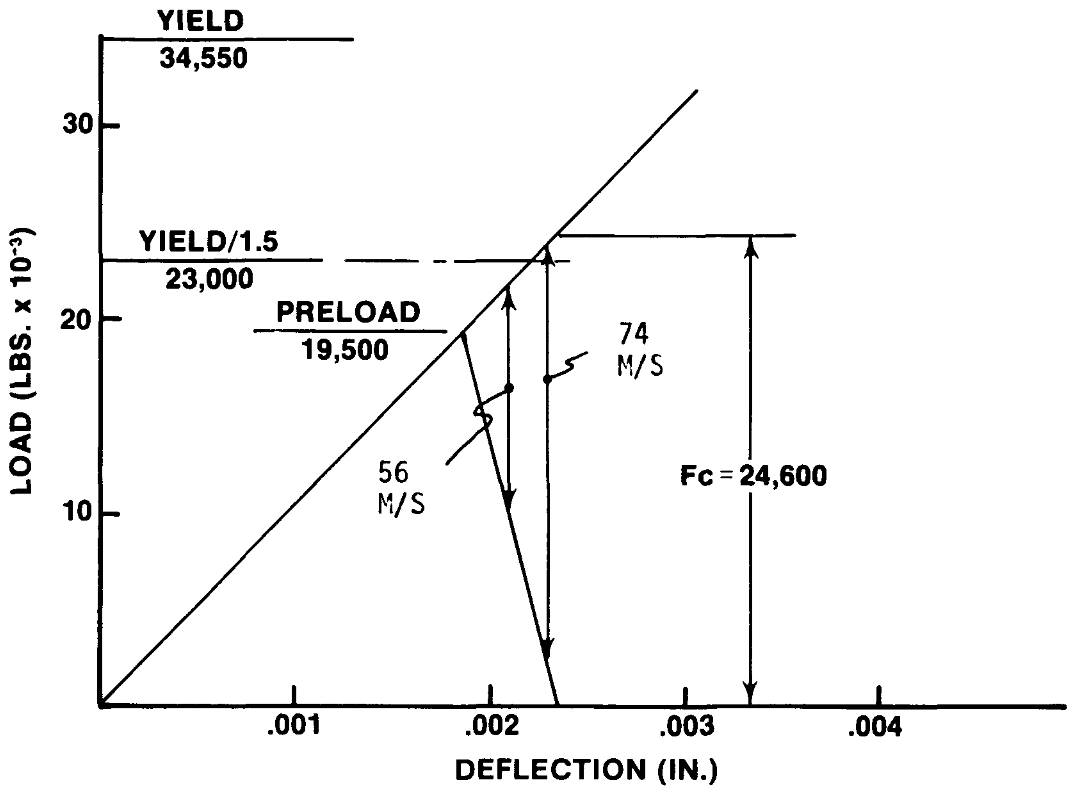
As seen from the Goodman diagram below, the stresses are well below the line and fatigue should never be a problem.

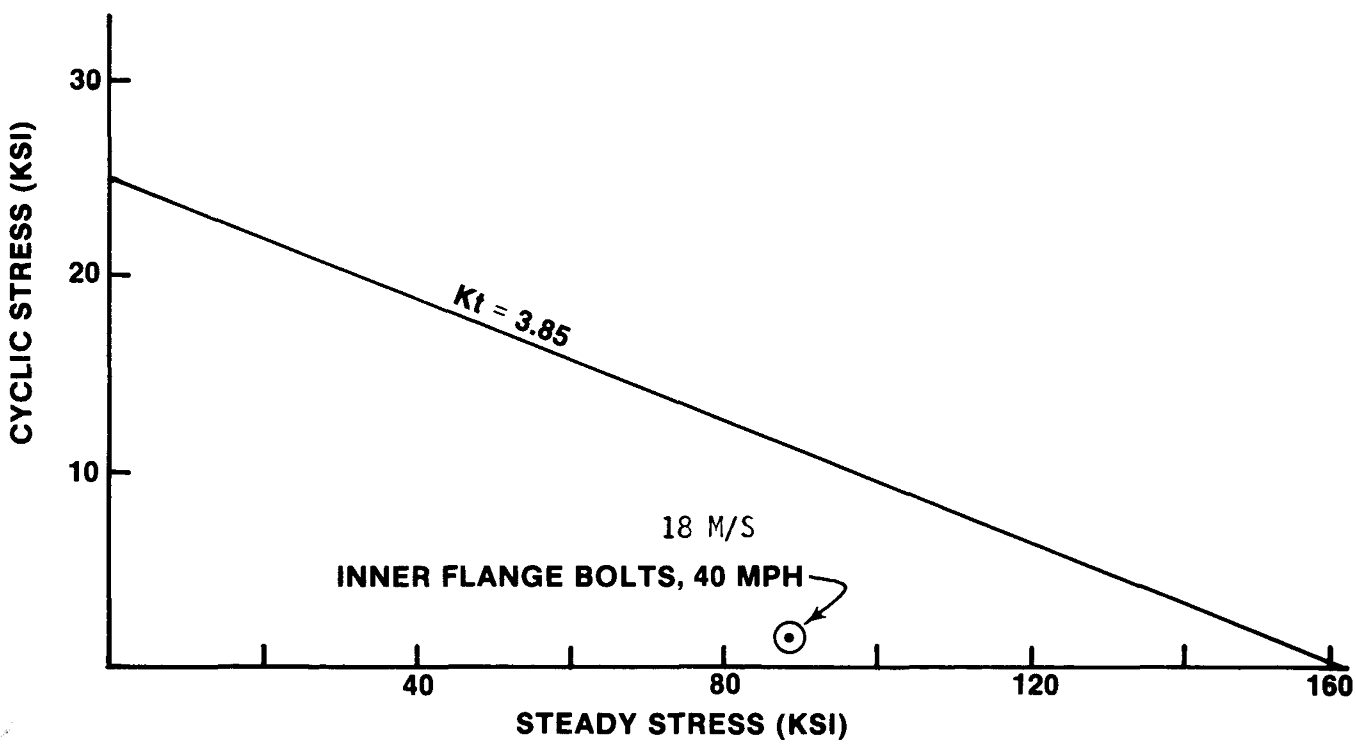


The same studs can be used for the 74 m/s design provided they are tempered to achieve 210 KSI tensile strength and preloaded to 28,500 lb.

The inner flanges of the legs are secured to the hub with 16-5/8-in socket head cap screws. These screws were analysed similarly to the blade studs and the results are shown on corresponding diagrams below.

Again, it can be seen that fatigue is not a problem, and there is adequate margin on yield strength and flange separation load.


YIELD
34,550
YIELD/1.5
23,000
PRELOAD
19,500
74
M/S
56
M/S
Fc = 24,600
LOAD (LBS. x 10⁻³)
30
20
10
.001
.002
.003
.004
DEFLECTION (IN.)


CYCLIC STRESS (KSI)
30
20
10
Kt = 3.85
18 M/S
INNER FLANGE BOLTS, 40 MPH
40
80
120
160
STEADY STRESS (KSI)

For the 74 m/s version the margin on separation is only 1.2 and the margin on yield strength is 1.44. To meet the required margins, special high-strength screws can be used (e.g. 5CRMoV, 280 KSI tensile).

3. Central Hub:

The central hub is subjected to all of the loads that feed down the three legs and, in addition, to the internal pressure generated by the tapered bushing.

The hoop stress due to the internal assembly pressure is given by:

$$\delta\theta = \frac{P\left[(Ro/Ri)^2 + 1\right]}{(Ro/Ri)^2 - 1}$$

Where $P = \frac{N}{2\pi\ RL}$

$$\delta\theta = 4400\ (1.75) = 7700 \text{ psi}$$

The 56 m/s gust load puts a toroidal moment on the hub which can be approximated by:

$$\delta\theta = \frac{MRC}{I_r}$$

$$= \frac{33{,}500\ (4)\ 4.5}{152}$$

$$= 4000 \text{ psi}$$

These stresses add on the upwind end of the hub:

$\delta total = 11{,}700$ psi

It can be seen that, even with a stress concentration of 3.0 at the keyway, there is adequate margin on yielding and fatigue.

The results of the hub assembly analysis are summarized in Figure 19 (above). The design is somewhat conservative and it may be possible to remove some weight by additional coring or thinning of some of the sections. It is not advisable to take advantage of this weight savings at this time for several reasons: (1) the stiffness of the hub affects the blade dynamics, (2) casting quality must still be determined, and (3) the cost of additional coring is not warranted for the prototype machines.

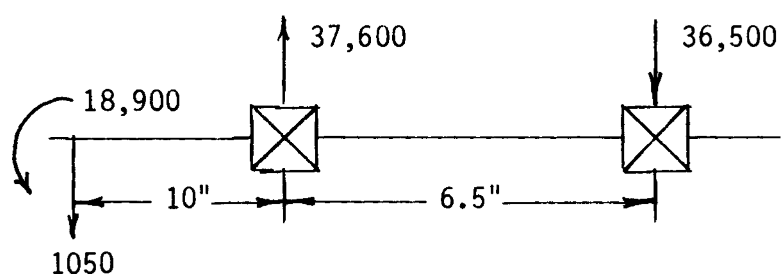
3.2.2 Gearbox

The gearbox selected for this machine is the Winsmith 20YHDC. It is a parallel shaft helical speed increaser with ratios of 5.5625 and 6.4 in the first and second stages, respectively. Helical gears with 25° full depth teeth and a helix angle of 17.728° are used. The low speed shaft is 4140 steel at Rc 28-32. The gears are nitrided with a core hardness of Rc 28-32 and a surface hardness of Rc 50-55. All bearings are Timken tapered roller. A schematic of the gearbox is shown in Figure 20.

1. Analysis at 56 m/s; rotor stopped.

Gearbox Moment	- One blade shadowed =	18900 ft-lb.
Thrust	- One blade shadowed =	5700 lb.
Thrust	- No blade shadowed =	7900 lb.

Bearing reactions are shown below:



The static load capacity of the mainshaft bearings is 149,000 lb. axial and 127,000 radial. Thus it is seen that the bearings are adequate.

Peak bending moment on the main shaft is 237,300 lb/in.

$$\text{Shaft stress} = \frac{MC}{I} + \frac{P}{A}$$

$$= \frac{237{,}300\ (32)}{\pi\ (3.5)^3} + \frac{5700\ (4)}{\pi\ (3.5)^2}$$

$$= 56{,}900 \text{ psi}$$

$$\text{Safety factor (S.F.) on yielding} = \frac{85{,}000}{56{,}900} = 1.5$$

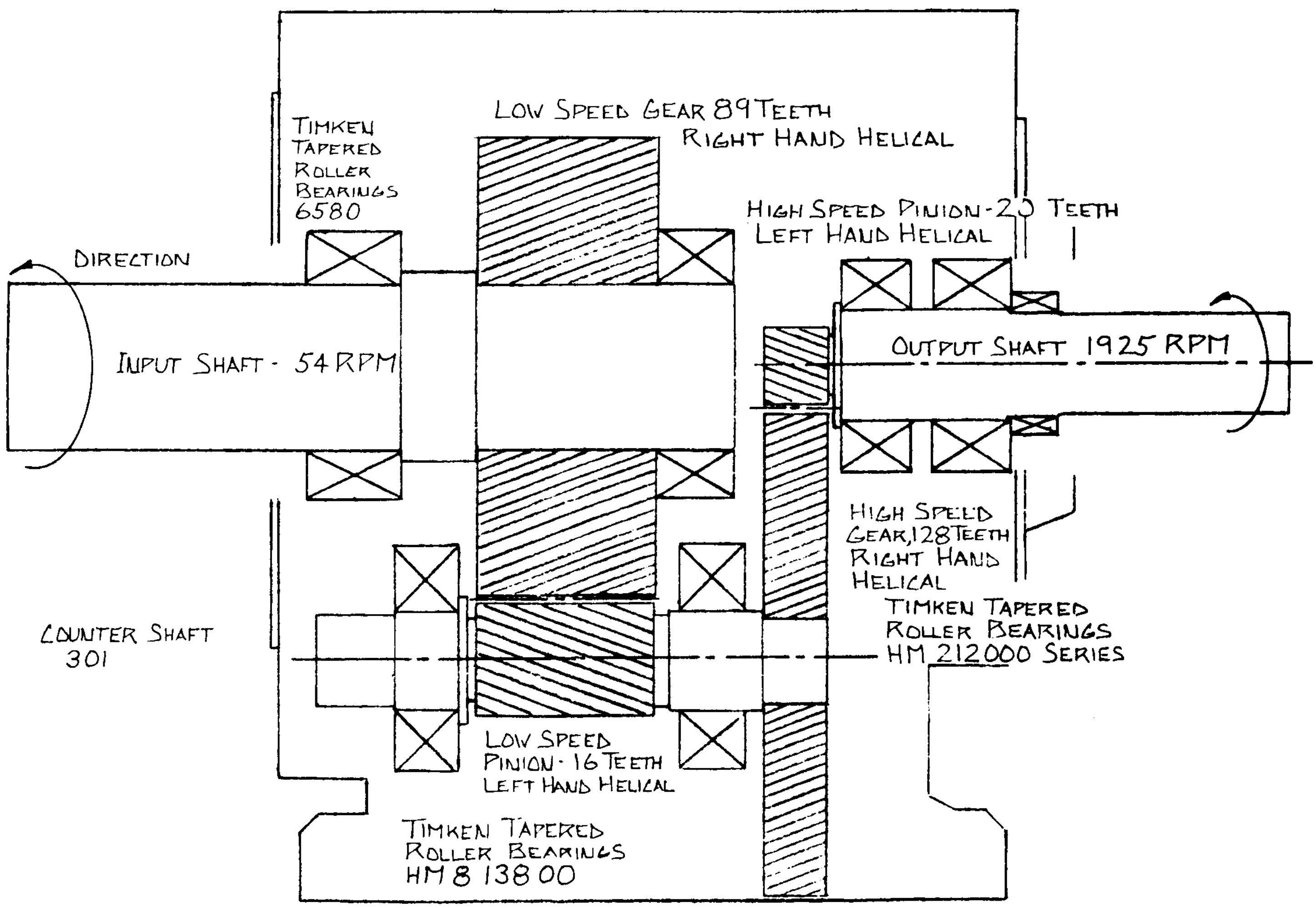


Figure 20

Gearbox Schematic

2. Fatigue Loads at 18 m/s Sustained Wind Speed.
Torque due to 15 kW output = 32340 lbf/in
(Efficiencies assumed are gearbox 0.93, Motor 0.78)
Thrust = 2040 lbf
Gearbox shaft = ± 68760 lbf/in

Gear tooth loading was calculated for spur gearing of equivalent dimensions. This is a conservative approximation which avoids the lengthy helical gear calculations (see Ref. 9). Winsmith has provided factors of safety, arrived at by their A.G.M.A. Standards, and they are slightly larger in all cases.

Dynamic Velocity Factor $kv = \dfrac{50}{50 + \sqrt{v}}$

where v = dn = linear ft/min

Form Factor $y = JK_f$,
where J = geometry factor (from tables)
and, $K_f = K_t$ = stress concentration factor taken as 1.5 (from tables)

Transmitted Load $Wt = \dfrac{2T}{D}$

where T = torque on gear
D = pitch diameter

Tooth Bending Stress $= \dfrac{Wt\ P}{KvFY}$

where P = diametral pitch of gear $= \dfrac{N}{D} = \dfrac{\text{no. of teeth}}{\text{pitch diameter}}$
F = face width of gear

For the low speed gear,

$$\text{Torque} = \frac{(15\ kW)\ (1.34\ \frac{HP}{kW})}{0.78\ (0.93)} \times \frac{33000\ \frac{ft\text{-}lb}{sec\ HP}}{2\pi \times 54\ (\frac{1}{sec})} = 2695\ ft\text{-}lb$$

$$Wt = \frac{2695 \times 12}{11.25} = 2875\ lb$$

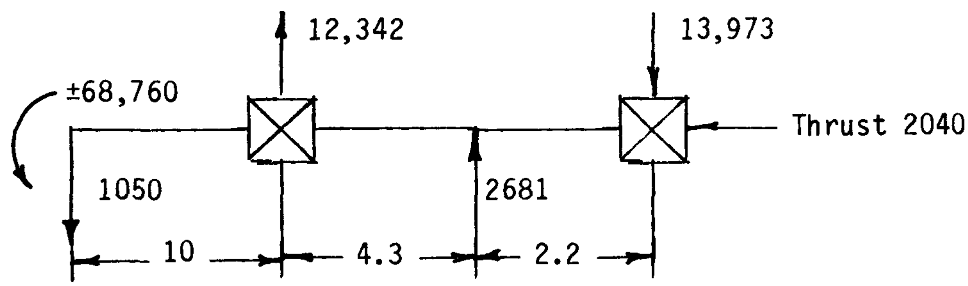
$$\delta = \frac{(2875)\ (\frac{89}{11.25})}{(0.799)\ (4)\ (0.51)} = 13952\ lb/in^2$$

$$Se = k_a k_b k_c k_d k_e k_f \; Se'$$
$$= (0.77)\ (1)\ (0.814)\ (1.00)\ (1.00)\ (1.35)\ (45000) = 38100 \text{ lb/in}^2$$

S.F. = 2.7

Other gears were calculated similarly and the results are shown in Figure 21. Shafting calculations have been made for the steady torque and alternating bending.

For the mainshaft running at 18 m/s; torque = 32,340 in-lb.



Tooth tangential load = $Wt = 2 \dfrac{(2695 \times 12)}{11.25} = 5749$ lbf

Tooth axial load = $Wa = Wt \tan (17.73°) = 1838$ lbf

Tooth radial load = $Wr = Wt \tan (25°) = 2681$ lbf

Shaft stress was calculated for the steady torque and alternating moment.

$$\delta = \frac{MC}{I} = \frac{79{,}260(32)}{\pi\ (3.5)^3} = \pm 18{,}800 \text{ psi}$$

$$\tau = \frac{TC}{J} = \frac{32{,}340(16)}{\pi\ (3.5)^3} = 3800 \text{ psi}$$

The Von Mises stress is: (stress due to thrust is negligible)

$$\delta v = \sqrt{\delta^2 + 3\tau^2}$$
$$= \sqrt{18.8^2 + 3\ (3.8)^2} = 19{,}900 \text{ psi}$$

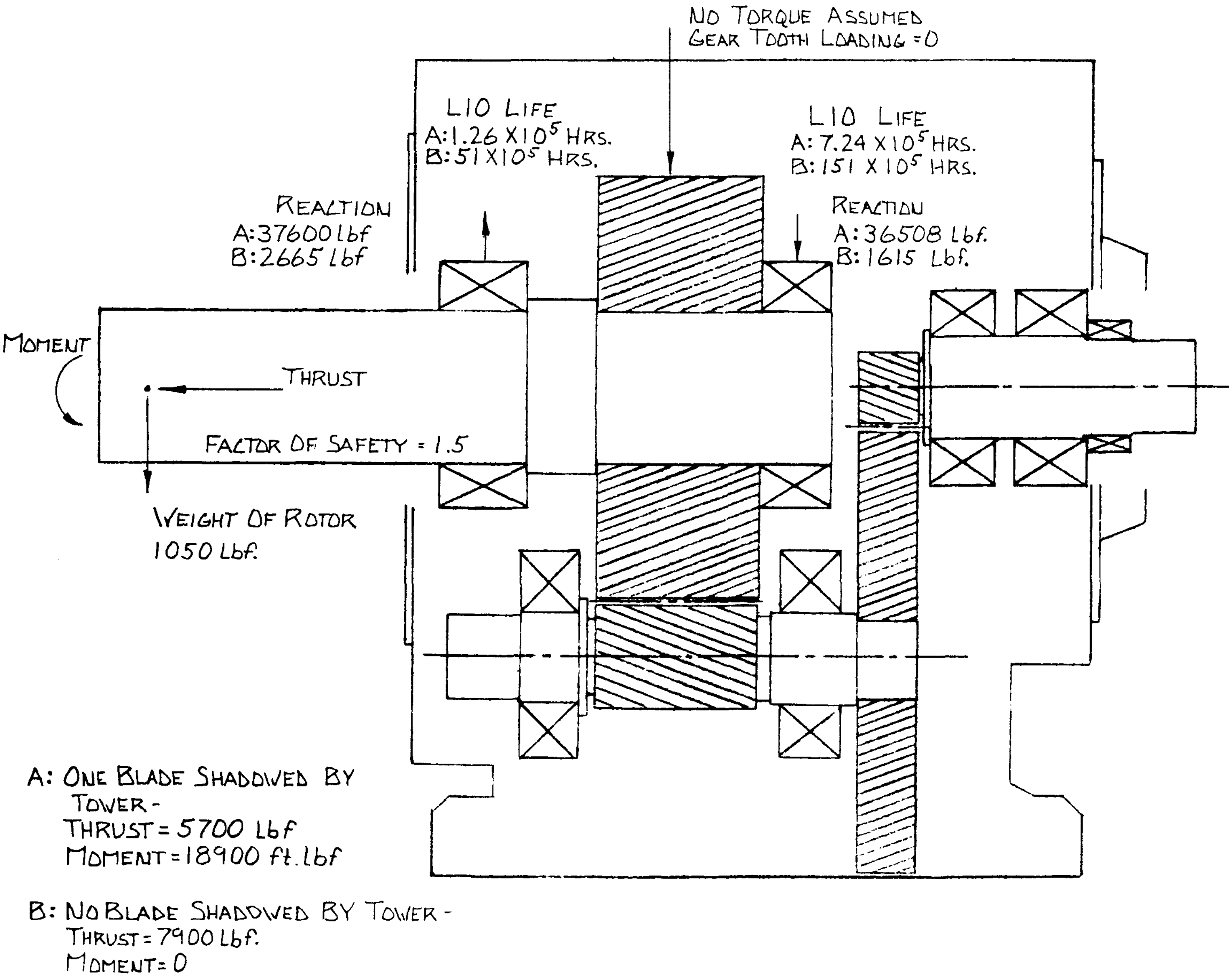
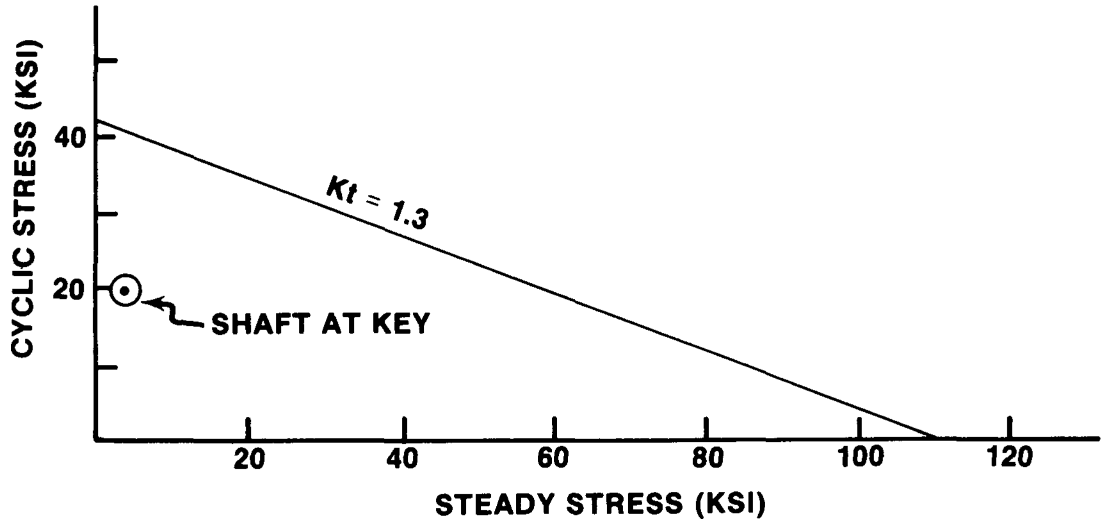


Figure 21

Gear Fatigue Loading - Static at 56 m/s

The stress concentration at the end of the key way is 1.3 (Ref. 4). This stress point is shown on the Goodman diagram below:



Wear on the two pairs of mating gear surfaces have also been calculated. The wear stresses do not exceed the design stresses. Figure 22 summarizes the results of the 18 m/s analysis.

3. Braking Action

The peak torque seen by the transmission is 125 ft-lb at braking. At this point the reverse sides of the gear teeth are brought into play and, similarly, the reverse sides of keys in all shafting. The braking action is expected to be seen only during power outages and in windspeeds averaging 18 m/s with cycles numbering less than 100 a year. Thus, fatigue is not considered a problem.

All bearings, keys, shafts, and gearing have been shown, through calculations, to be able to withstand the braking action. Safety factors are illustrated in Figure 23.

For the 74 m/s design the low speed shaft diameter will need to be increased to 4.25 in. This will require larger bearings and a modified gearbox casting.

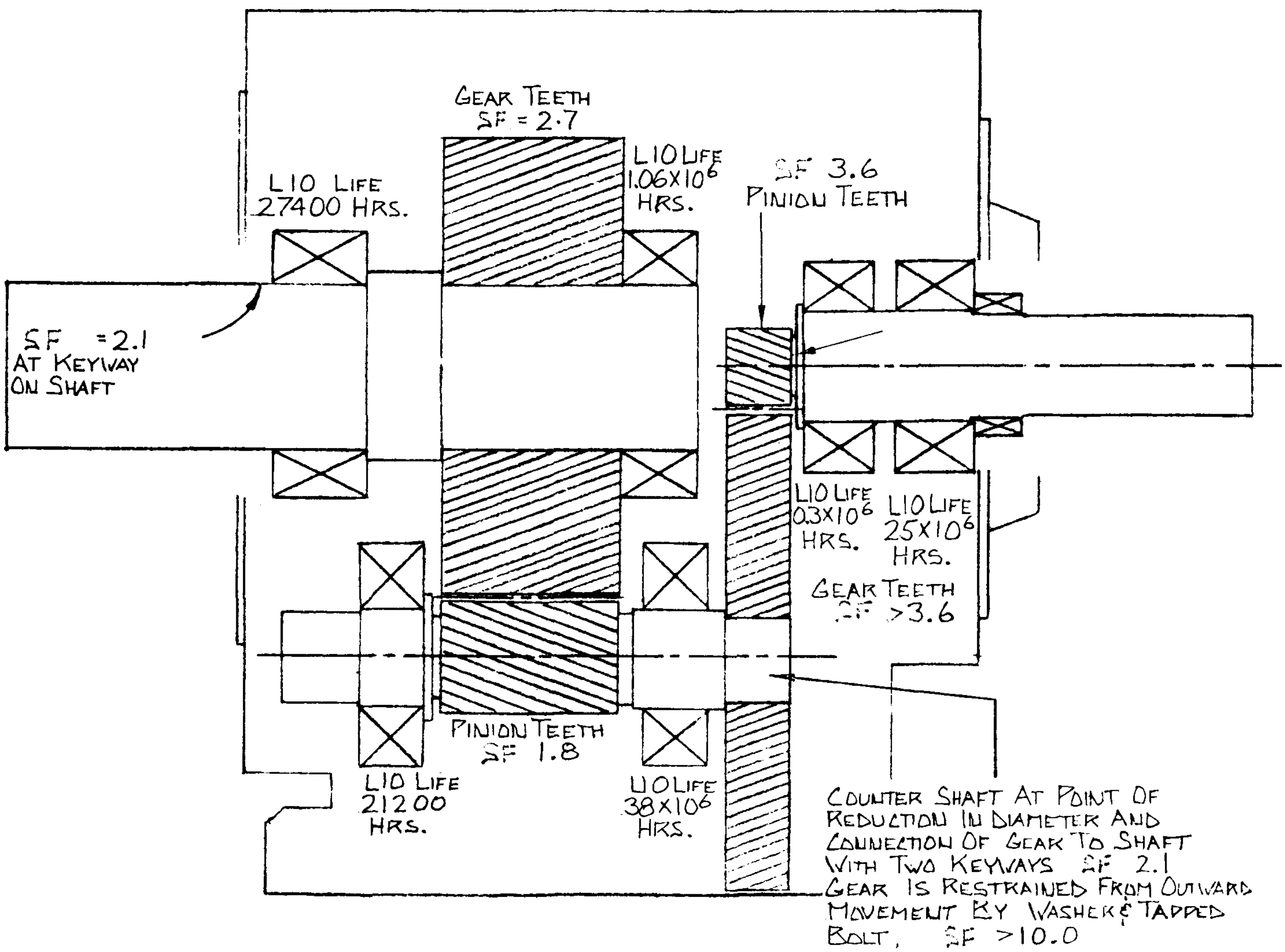


Figure 22

Gear Fatigue Loading - Yawing at 18 m/s

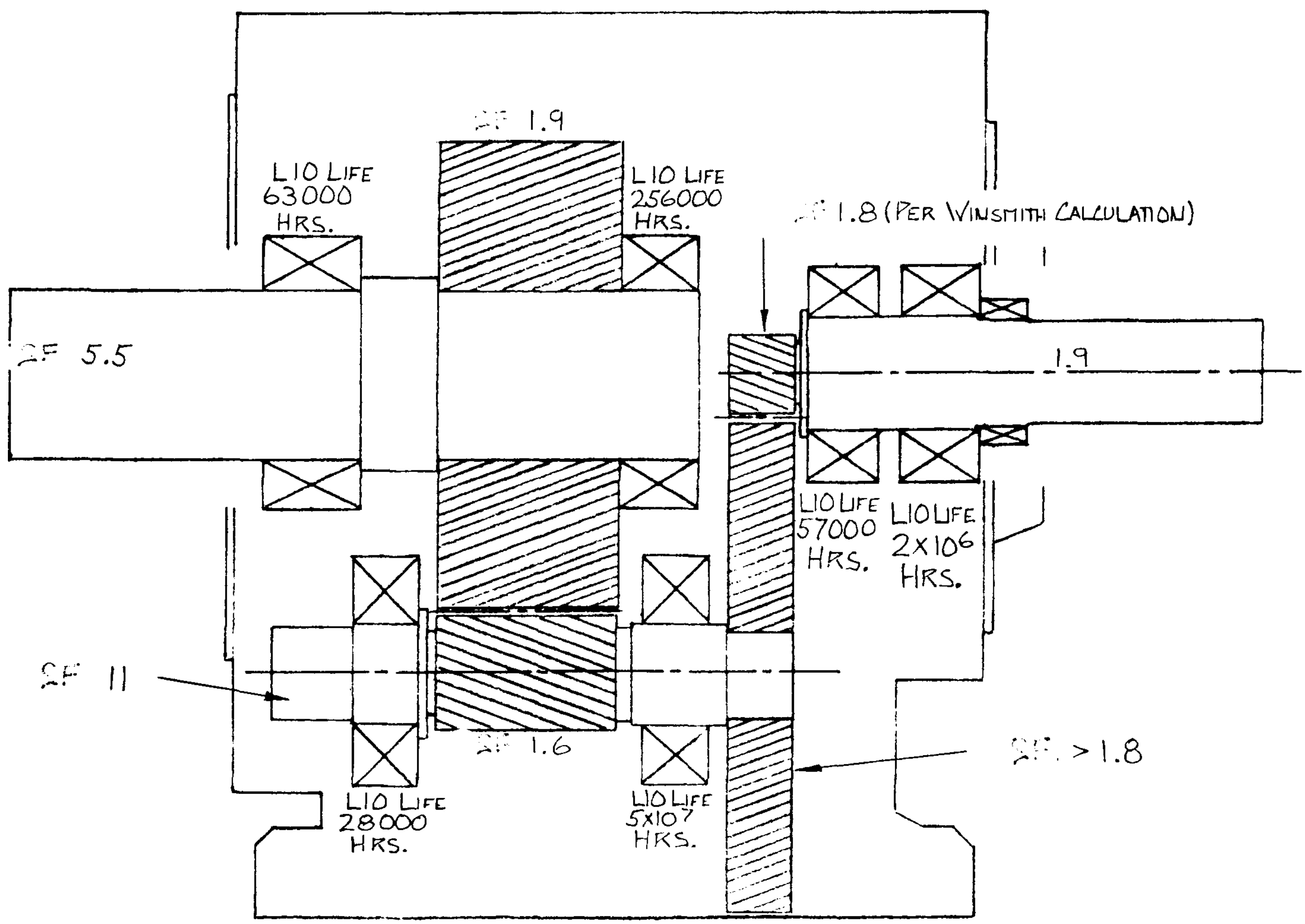


Figure 23

Gear Fatigue Loading - Braking Condition at 18 m/s
With Braking Torque 125 ft-lb

3.2.3 Mainframe and Yaw Bearing

The mainframe and yaw bearing have undergone extensive redesign since PDR. The reasons for this are:

1. The loads were found to be higher than previously thought.

2. A torsional vibration was discovered in the test model when the brake was applied at low wind speed. This vibration appeared to be a complex coupled mode that involved twisting of the frame.

3. It was discovered that a turntable type bearing was less expensive than two pintle shaft bearings and the supporting structure that is inherent in that type of design. In addition the use of the turntable bearing lowers the machine profile and reduces loads on the tower top.

The frame, leveling plate and tower top are shown in Figures 24, 25, and 26.

The mainframe consists of two 3/8-in thick A36 steel plates bent into U shapes and welded together to form a 6-in x 12-in box section beam 52-in long. To prevent buckling of the 12-in wide plates, two 3/8-in thick vertical stiffening plates are welded inside the box section. The reason for using the plate bent into U shapes instead of channel iron is to ensure flat and parallel surfaces at the yaw bearing attachment plate.

A Kaydon MT0210 turntable is used for the yaw bearing. The bearing is sealed and a grease fitting is provided. This type of bearing must be mounted on a flat rigid surface to ensure free yawing. To achieve the required rigidity a 1-in thick plate is mounted inside the 6-in x 12-in box frame. This plate is threaded to receive the bearing mounting bolts, and welded to the frame and stiffener plates. A spacer plate is sandwiched between the bearing and the box frame and carries the slip-ring assembly and ensures that the slip-rings are concentric with the machine axis. Therefore, the lower plate of the box frame is sandwiched between, and welded to, two rigid plates which carry the yaw bearing. This insures a good transfer of load from the frame to the bearing.

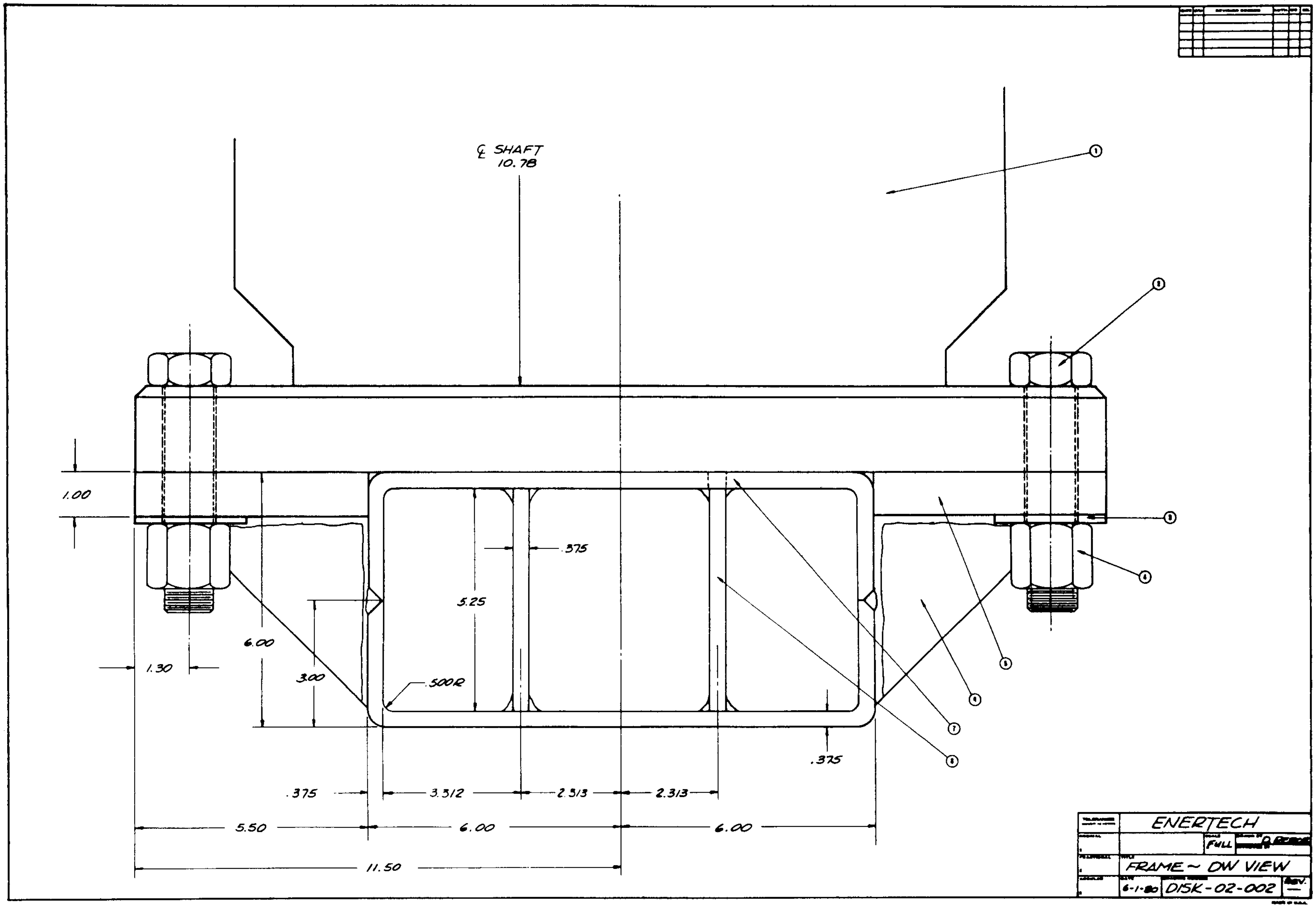


Figure 24

Main Frame

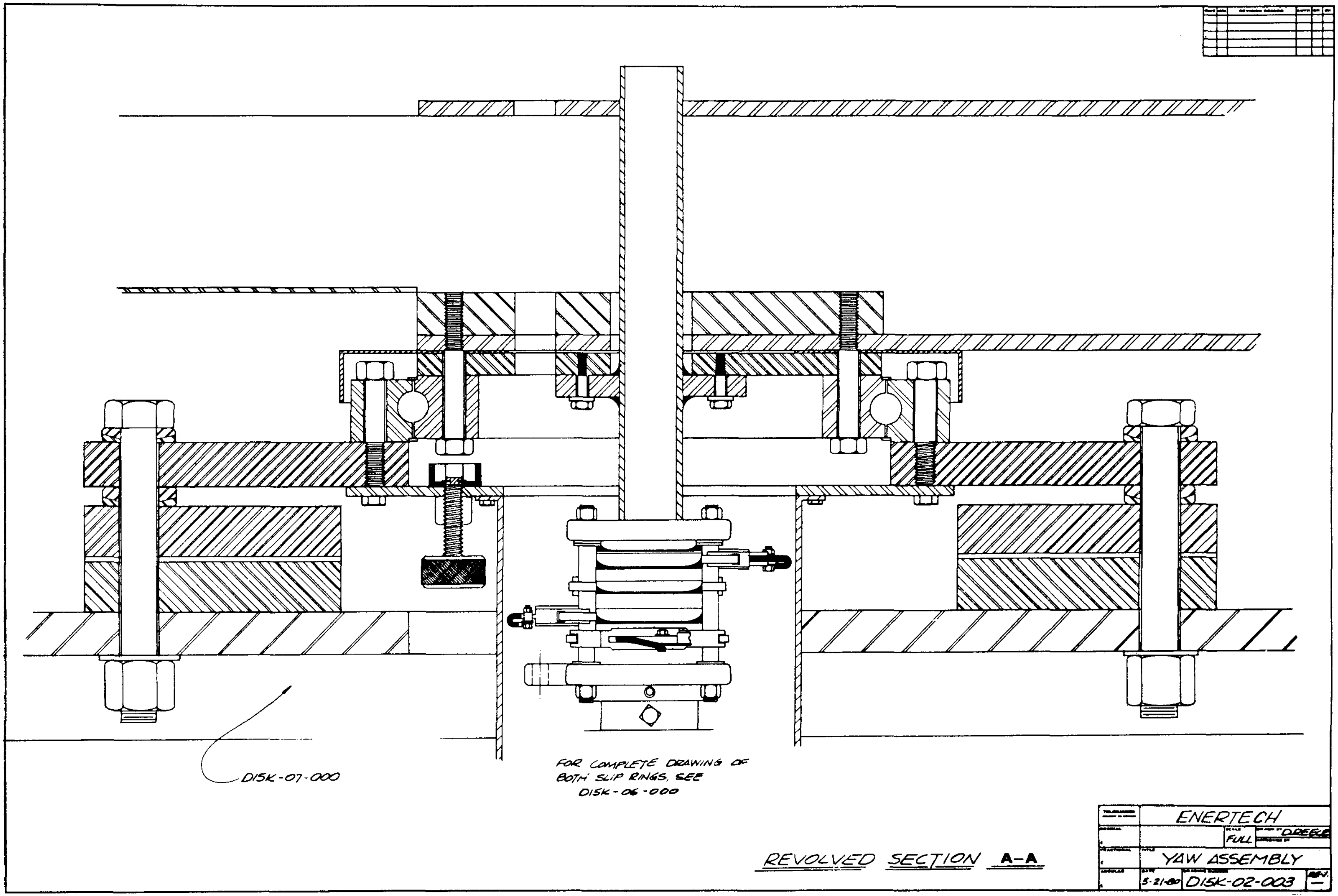
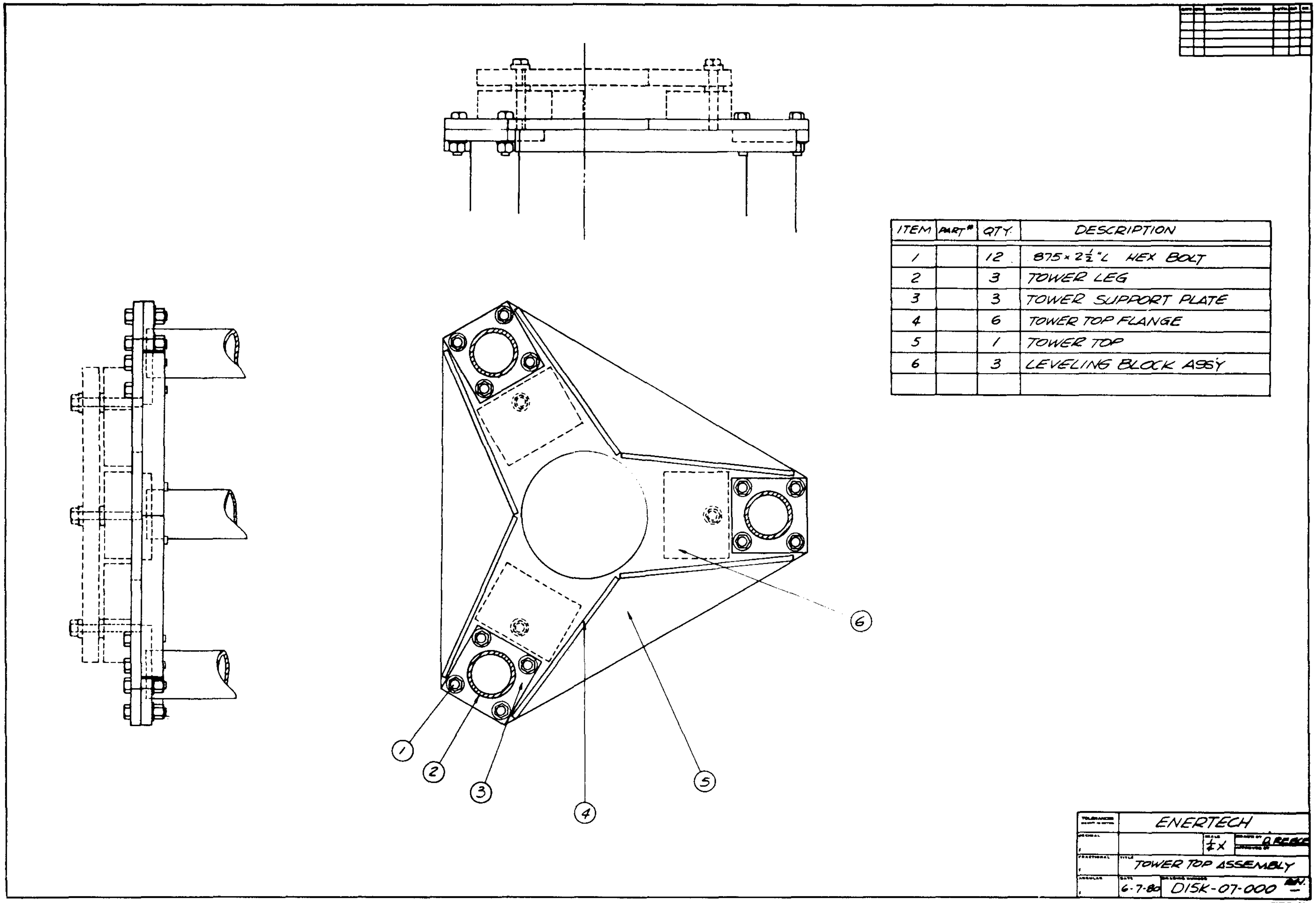


Figure 25
Yaw Bearing Assembly



ITEM	PART #	QTY.	DESCRIPTION
1		12	.875 × 2½"L HEX BOLT
2		3	TOWER LEG
3		3	TOWER SUPPORT PLATE
4		6	TOWER TOP FLANGE
5		1	TOWER TOP
6		3	LEVELING BLOCK ASSY

Figure 26

Tower Top Assembly

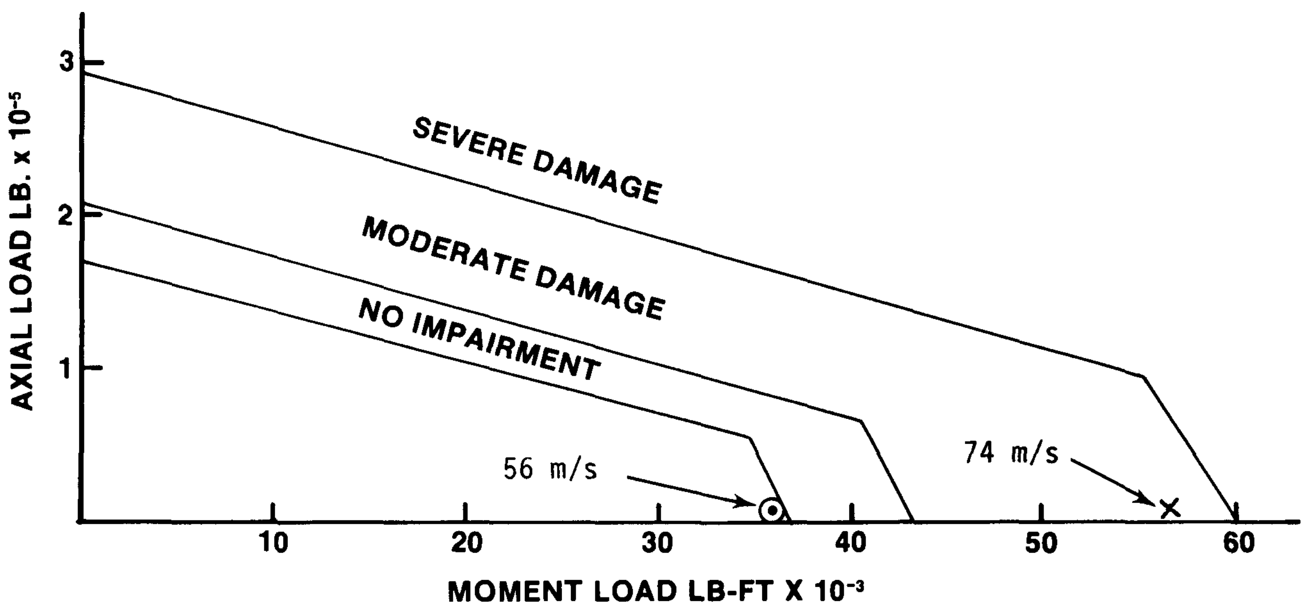
Yaw Bearing Design Analysis The yaw bearing was checked by Kaydon engineers and found to be suitable for this application. The worst load on the bearing occurs at 56 m/s (125 mph) with one blade shadowed. The moment at the bearing for this condition is:

Over-hung Weight	95,000 in-lb
Thrust	114,000
Shadowed Blade	226,800
Total	435,800 in-lb

The axial load on the bearing is the total machine weight

Weight = 2,400 lb

This loading point is shown on the Kaydon graph below along with the 74 m/s point, (see Ref. 12).



It can be seen that the loads for the 56 m/s design are within the allowable. The loads for the 74 m/s design fall in the area of moderate damage. The moderate damage area on the graph is where slight brinelling occurs. Fracture is about five times the load at 56 m/s.

For the 74 m/s design a larger bearing must be used to prevent brinelling. The Kaydon MT0265 is recommended.

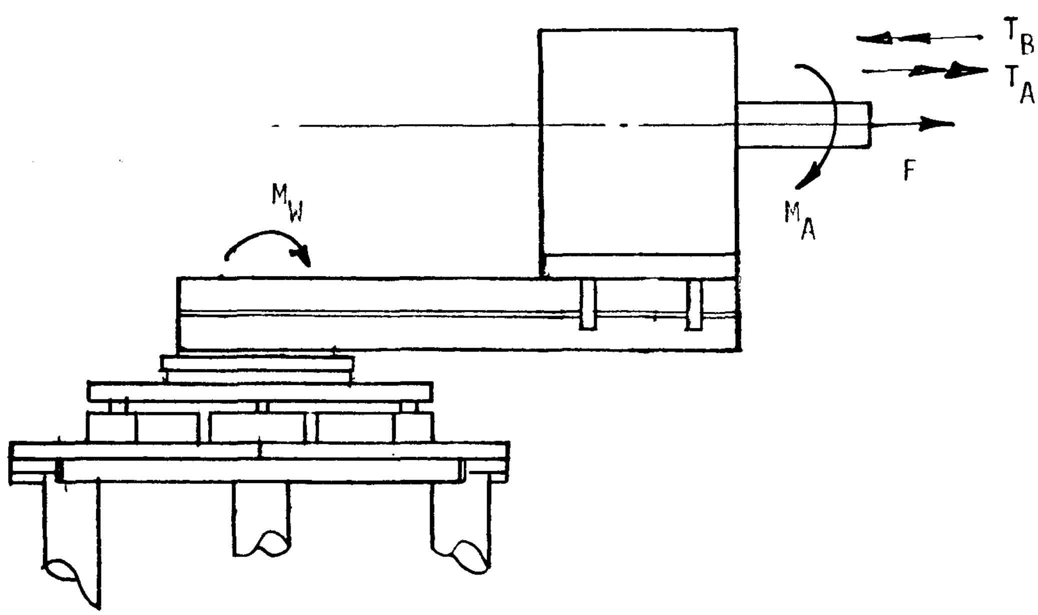
This bearing experiences no impairment for up to 60,000 ft-lb load and sustains only moderate damage up to 87,000 ft-lb load.

The maximum allowable deflection (out of flat) of the bearing mounting surface is .013 in. to insure long life and free yawing. The calculated deflections are much less than this, being only .006 at 56 m/s.

<u>Main Frame Design Analysis</u> The loads on the mainframe are summarized in the sketch and table below:

	56 m/s (125 mph) gust	73.8 m/s (165 mph) gust	18 m/s (40 mph) 30° yaw
Thrust, F (lb)	5700	9990	1400 ± 640
Moment, M_A (in-lb)	226800	395000	11760 ± 57000
Weight Moment, M_W (in-lb)	95000	100000	95000

Aerodynamic Torque, T_A = 40,560 in-lb
Braking Torque, T_B = 53,400 in-lb



The maximum bending stress in the main beam occurs just downwind of the yaw bearing attachment.

$$I = \frac{1}{12}\left[4(5.25)^3 .375\right] + 2(.375)12(2.812)^2$$

$$= 18 + 71 = 89$$

for 56 m/s:

$$\delta = \frac{MC}{I} = \frac{403{,}500(3)}{89} = 13{,}600 \text{ psi}$$

The yield strength for A-36 steel is 36,000 psi

Therefore SF = 2.64

The maximum shear stress occurs at the weld.

$$\tau = \frac{3}{2}\frac{V}{A}$$

$$= \frac{3}{2} \cdot \frac{2400}{13}$$

$$\simeq 300 \text{ psi}$$

Under braking torque the frame experiences a torsional stress:

$$\tau = \frac{T}{2At} \quad \text{(Ref. 14)}$$

$$= \frac{53{,}400}{2(65.4).375}$$

$$= 1100 \text{ psi}$$

For the fatigue condition the loads are:

Gearbox bending moment	-	11,760 ± 57,000 in-lb
Moment due to thrust	-	22,090 ± 10,100
Weight moment	-	95,000
		128,800 ± 67,100 in-lb

Bending stress in the frame is:

$$\delta = \frac{MC}{I} = \frac{(128{,}800 \pm 67{,}100)\ 3}{89}$$

$$= 4300 \pm 2300 \text{ psi}$$

Thus, it is seen that stresses in the frame are well within allowable limits.

The lower plate of the box section was found to have little margin on buckling at the 74 m/s condition. Therefore, the two vertical stiffening plates were added to ensure adequate capacity at 56 m/s and 74 m/s.

The gearbox mounting pads are 1-in thick plates welded to the box frame and gussetted. All of the welds in this area were checked using the methods outlined in Reference 16. The stresses in the welds are well below allowables.

The gearbox is attached to the mainframe with four 1 1/8-in diameter bolts. These bolts are adequate for all conditions including the 73.8 m/s. The preload should be 50,000 lb or about 1000 ft-lb of torque. The angle-of-turn method will be used to preload these bolts.

Bearing Mounting Bolts

The yaw bearing is held in place by 1/2-in diameter grade-8 bolts. There are 20 bolts on the inner race and 16 on the outer. The equation recommended by the manufacturer for determining the bolt load is (Ref. 12):

$$P = \frac{3M}{DN} \pm \frac{F}{N}$$

Where: M = the applied Moment
D = diameter of bolt circle
N = number of bolts
F = axial load

For 56 m/s the moments are:

Thrust	115,000 in-lb
Weight	95,000
Shadowed blade	226,800
	436,800

Pinner = 7010 lb

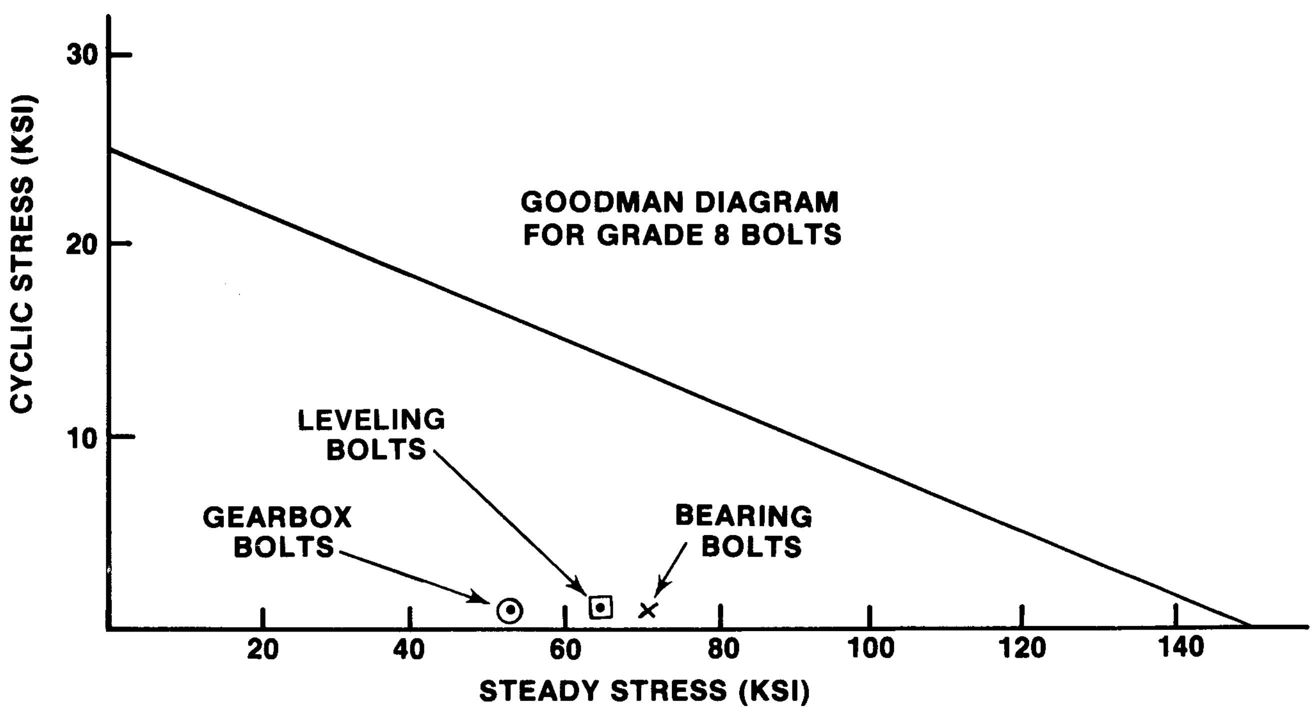
Pouter = 6350 lb

Similarly for the fatigue condition the moments are:

134,800 ± 69,800 in-lb
Pinner = 2240 ± 1200 lb
Pouter = 2050 ± 1130

The preload of all the bearing bolts was set at 10,000 lb. This ensures a 1.5 margin on yield and gives a 1.58 margin on separation. These bolts should be torqued to 110 ft-lb to achieve the 10,000-lb preload.

For the fatigue condition the stresses in the bolts are plotted on the Goodman diagram below:



For the 74 m/s design, the same bolts can be used with the MTO 265 bearing. The increased diameter and number of bolts reduces the individual bolt load.

Leveling Plate The leveling plate is a 1-1/2-in thick A36 steel plate. The outer race of the yaw bearing is bolted to the leveling plate and three 7/8-in diameter bolts mount the plate to the tower top through the leveling blocks. A 1/4-in thick circular plate is bolted to the bottom of the leveling plate which seals the underside of the bearing from the elements and provides the non-rotating structure to support the slip-ring brushes, wires and slip-ring cover. The leveling plate has a 1.83 safety factor on yield and fatigue is not a problem.

The three 7/8-in diameter leveling bolts are grade-8 and have a safety factor of 1.53 on yield and 1.54 on separation. The preload for these bolts should be set at 33,500 lb. or 615 ft-lb. torque.

For the 74 m/s design the leveling plate should be 1-5/8-in thick and the leveling bolts should be increased to one inch.

Tower Top The tower top consists of 1-in thick steel plate with gussetts welded on the bottom. It is bolted to the tower legs with four 7/8-in bolts at each leg. The weldment was sized for the 56 m/s loads and treated as a pinned ended beam which is conservative. The safety factor for the weldment is 1.8. The safety factor for the 12 bolts holding the plate to the tower is about 5.

3.2.4 Nacelle

The nacelle is made of molded fiberglass similar to the Enertech 1500. The lower portion of the nacelle is made in two halves, split axially, so that it can be installed after the machine is bolted to the tower top, (see

Figures 27 and 28). This feature insures that the nacelle will not be damaged during erection and allows the skirt to extend down over the bearing area which helps to keep that area dry. The top piece is hinged at the rear and tilts up for servicing. This allows access to both sides of the machine which, we have found from experience with the 1500, is a desirable feature. The spinner is similar to that of the 1500 and attaches to the hub. It is planned to use Southco rubber draw latches to secure the top of the nacelle. These are currently being tested for use on the 1500.

Stresses in the nacelle are very low and no structural problems are foreseen.

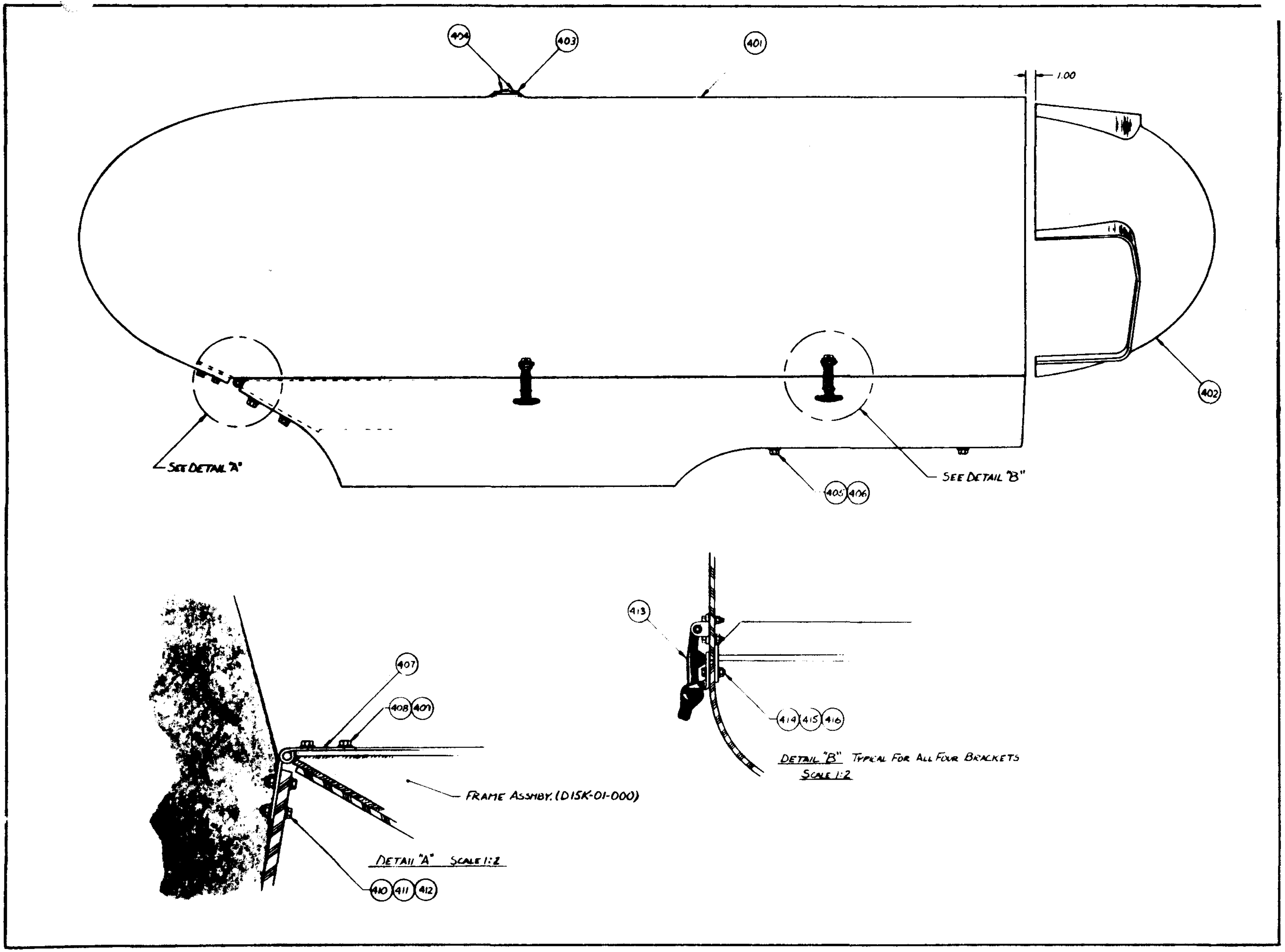


Figure 27

Nacelle Assembly

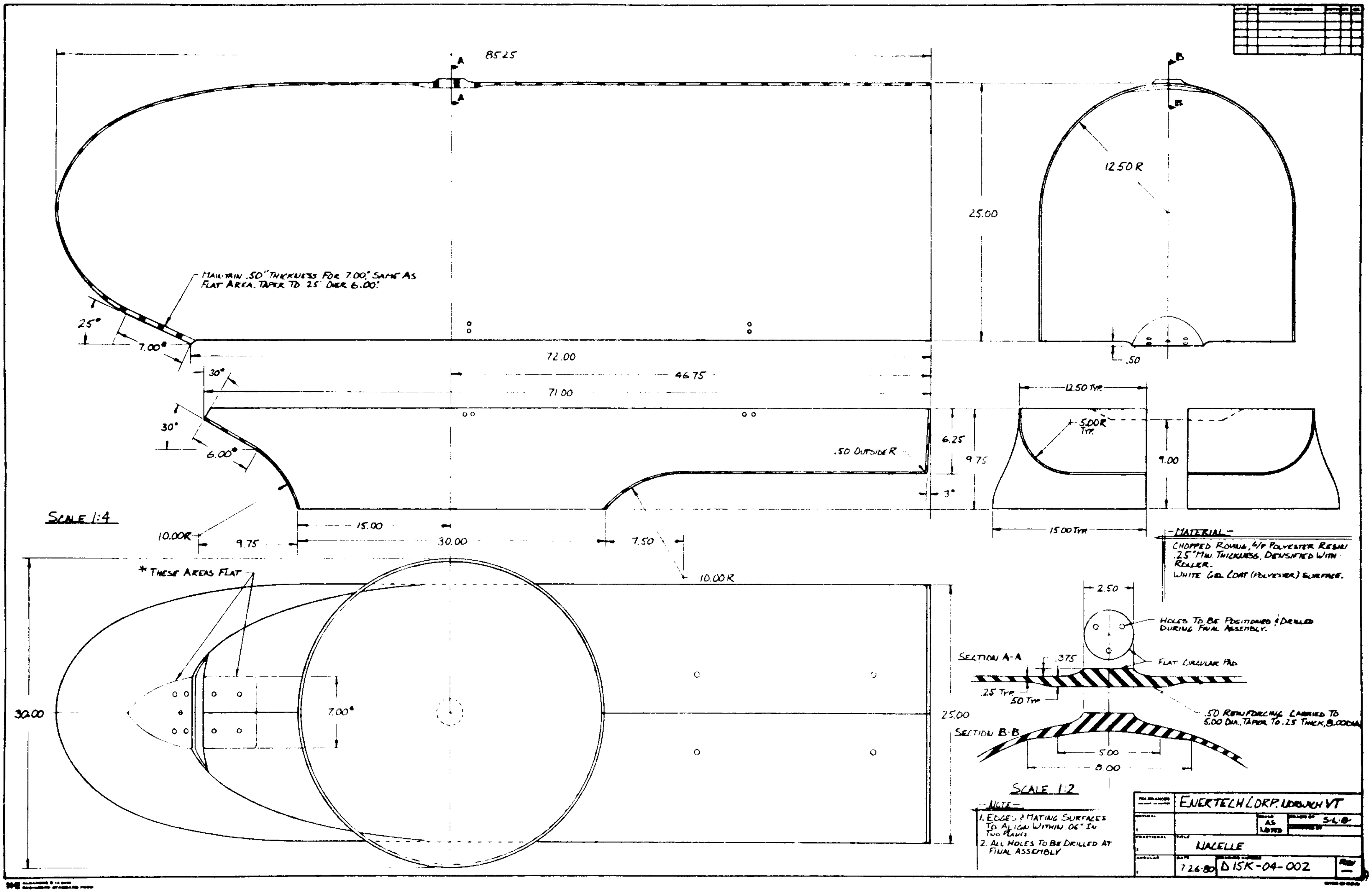


Figure 28
Nacelle

3.2.5 Tip Brakes

Since CDR the tip brakes and latching mechanism (see Figure 29) have been redesigned. First, the end plate was changed from a fiberglass molding to a plain aluminum plate. The main reason for this change is reliability. If the fiberglass part does not unlatch for any reason it becomes useless. If the aluminum plate does not unlatch, it soon yields and bends outward, becoming almost as effective a brake as if unlatched . This yielding was demonstrated during truck tests on the test model tip brakes in August as well as during tests on Mt. Washington in New Hampshire.

In addition, fiberglass tip brakes would have been more expensive. Molds would cost $4000 and the brakes $120 each, while the aluminum parts cost only $25.

The latching mechanism was changed slightly after CDR. The spring was moved from the radial position to the tangential position. The reason for this change is to facilitate adjustment of the spring tension and avoid having to fish for the eye of the spring inside the blade at assembly.

All parts have been made of stainless steel or aluminum to prevent corrosion. The areas where unlatching motion is required are stainless on stainless or plastic on stainless to prevent any possibility of freezing up due to electrolytic action. Aluminum parts will be anodized.

See Table IX for tip brake design parameters.

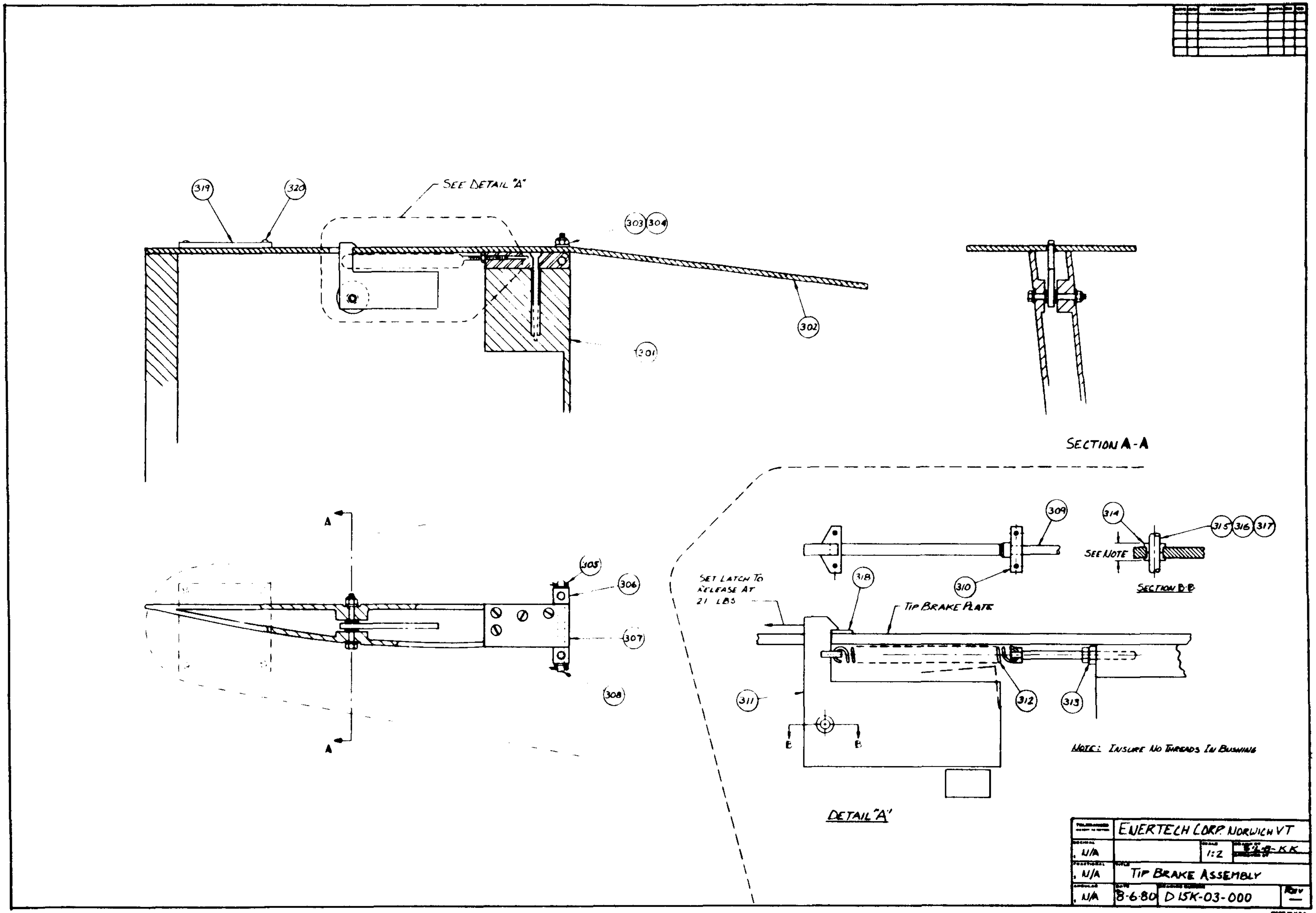


Figure 29

Tip Brake

Table IX

Tip Brake Design Parameters

Parameter	E-1500	E-4000	15 kW
Assumed power at rotor for maximum permitted speed	7200 W 9.6	15,300 W 20.4	57.6 kW 76.8
Generator/rotor rpm at above condition	2825 250	2750 162	2672 75
Resulting torque on Rotor (ft-lb)	202	662	5380
Equivalent force at each blade tip (lb)	10.2	22.4	81.5
Plate areas required at each blade tip (CD = 1.5, 2/3 redundancy)	42 in^2	97.5 in^2	330 in^2
Tip speed at above condition (ft/sec)	172	167	173
g force at blade tip at above condition	140	88	42
normal max g force	70	44.3	22

3.2.6 Slip-Ring Enclosure and Yaw Lock

After CDR it was found that a larger diameter slip-ring enclosure was required. This change also necessitated redesign of the yaw lock. The slip-ring cover is a round 16-gage steel can which slips off downward with the removal of 3 screws exposing the entire slip-ring assembly. The slip-ring enclosure mount plate has 8 holes for attaching to the leveling plate; thus, there are 8 possible angular locations for the incoming wires from the tower. Slip-ring shaft eccentricity has been held to a minimum to prevent motion and possible chafing of the wires.

The new yaw lock is of the same type as the old one; that is, it engages the bolt heads of the bearing mounting bolts. The new design (see Figure 25) is simpler and less expensive than the previous one.

3.3 Electrical Subsystem Design

Designs for single and three-phase (240 Vac) electrical power subsystems are presented in this section. Note that there is no change in electrical design for the 74 m/s machine. Figures 30 and 31 illustrate the subsystems and components described in this section.

3.3.1 Description of Operation

It is the nature of an induction motor to become an induction generator when driven beyond synchronous speed. Upon control system command the generator is connected to utility power. The system motors up to synchronous speed, is driven beyond synchronous speed by the wind, and begins to generate power.

Various system components are needed between the generator and the utility line for safety, power, transmission, and control purposes. Slip-rings accomodate wind machine yaw, a cable transmits power, a magnetic contactor functions to disconnect and connect the generator from utility power under

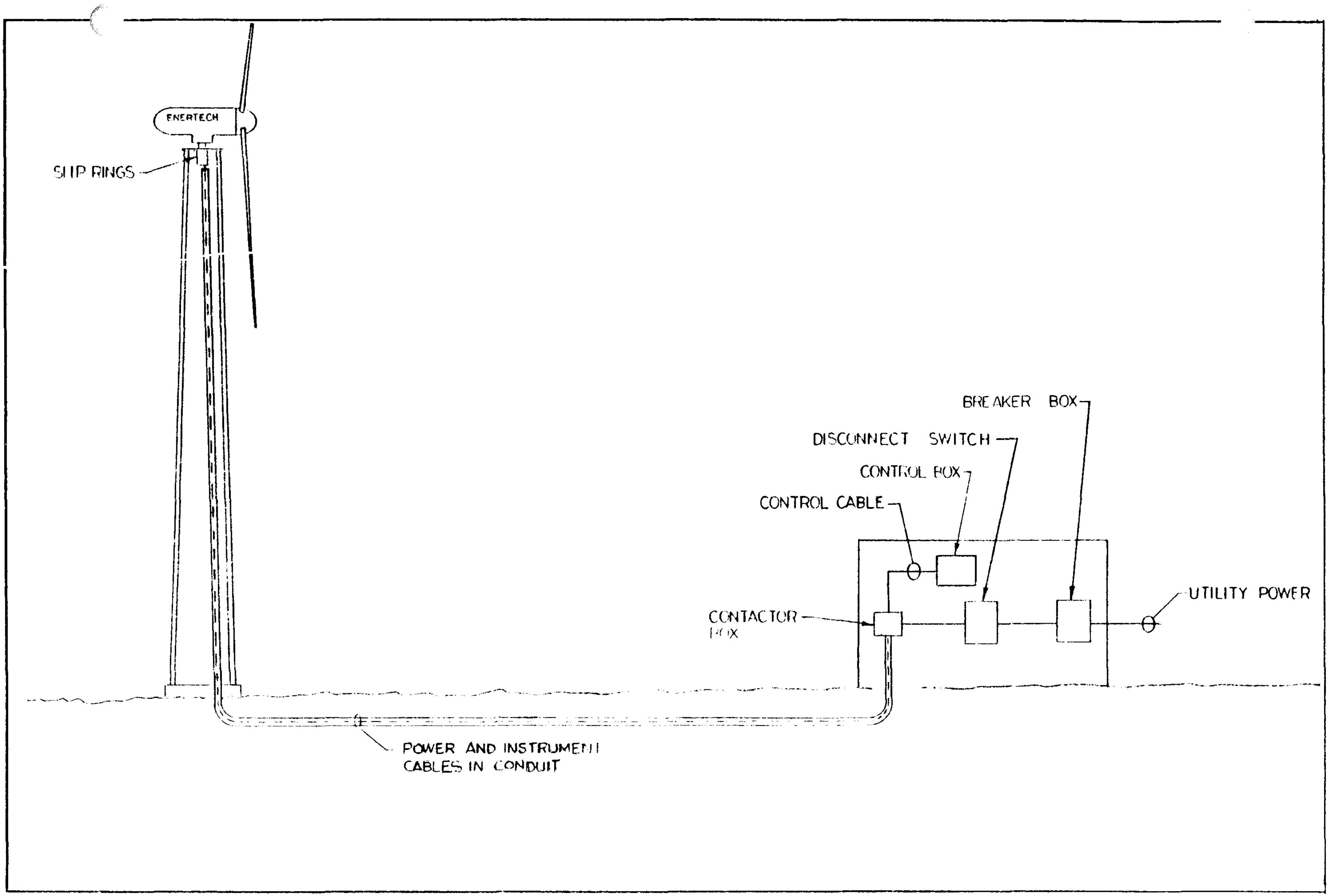


Figure 30

General Electrical System Layout

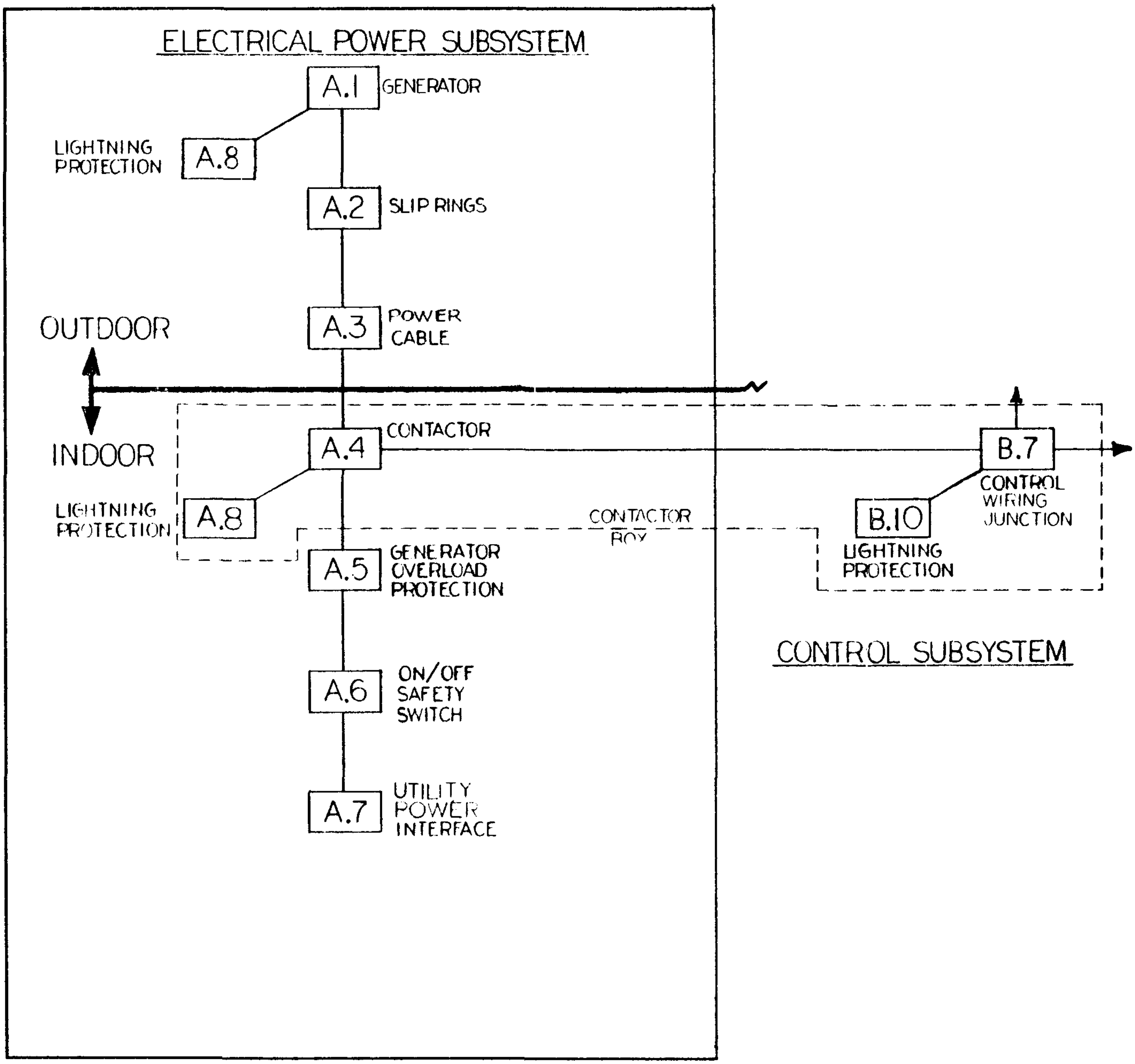


Figure 31
Electrical Subsystem Block Diagram

control system command, and overload protection device prevents generator damage from excess current, and a safety switch provides a positive disconnect for service purposes. A means of protecting generator and utility lines from lightning is provided, and a circuit protective device (fuse, circuit breaker) is provided at the utility interface.

3.3.2 Single Phase Design - Power Subsystem (see Figure 32)

Generator (A.1)

Gould Inc. is the supplier of the generator. It is a single-phase wound machine designed for 17 kW continous output. An auxiliary winding is used to start the unit. The primary concern has been the magnitude of the inrush current as the wind turbine motors up to speed. We have taken advantage of Gould's computer simulation capability to generate reliable data on a simple yet elegant design. It was determined that a reduced voltage starting scheme would be quite effective. The generator will be energized with 115 Vac upon startup and motored to 1000 rpm. It will free-wheel to synchronous speed at which time it will be connected to 230 Vac. The auxiliary winding alone receives the low voltage current. More detail about this winding is not available except that it is required to possess at least 10 ft-lb locked rotor torque. The generator power factor at 20 kW is to be .8 minimum. Gould Inc. has agreed to supply these machines to Enertech for $1000 - 1200, which is apparently 25% higher than a comparable 3-phase 25 kW motor but well in line with what a single-phase motor of this size would cost.

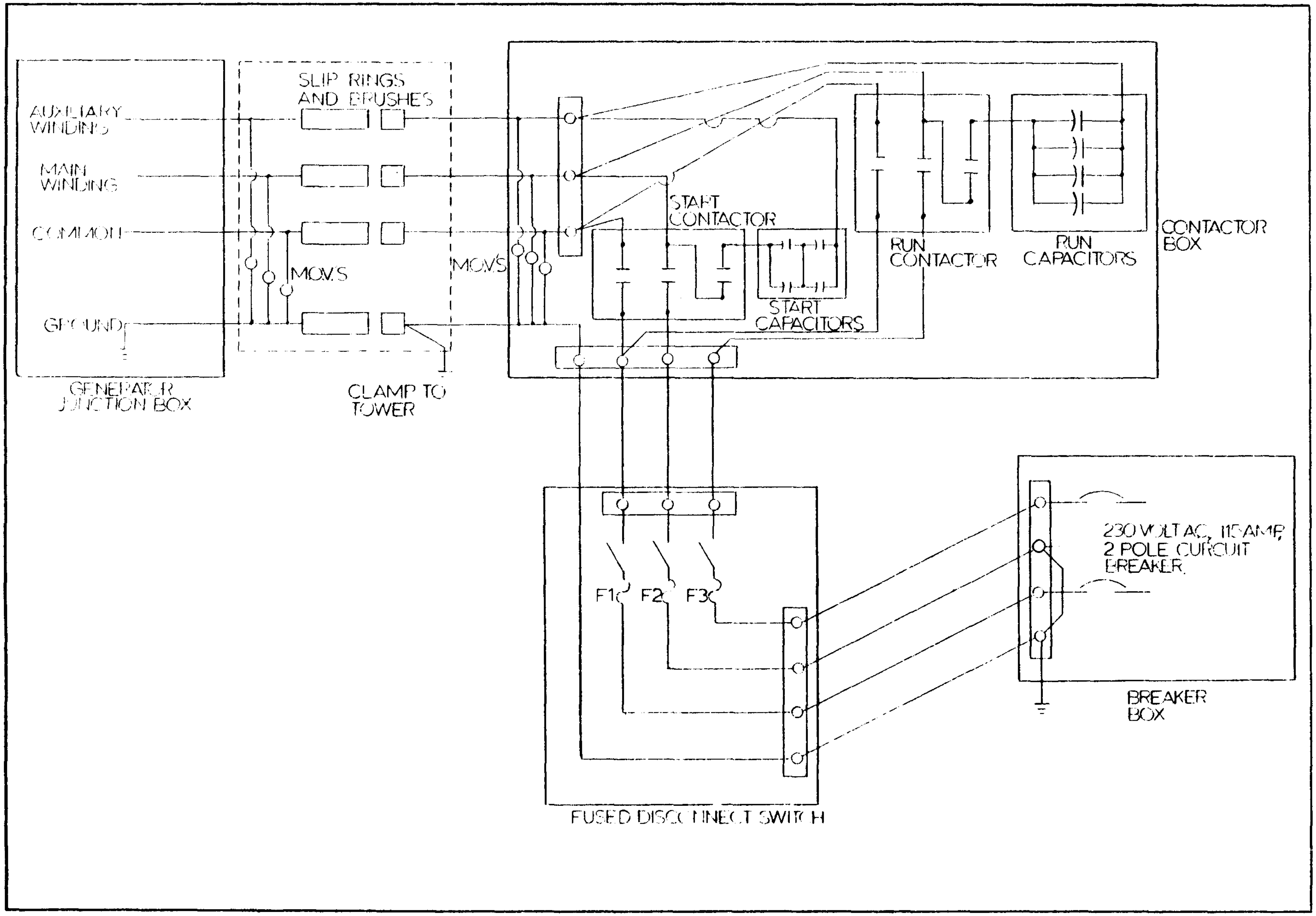


Figure 32

Single-Phase Electrical Power Subsystem Wiring Diagram

Enertech also had a generator custom-wound to similar specs by a local motor shop, and it yielded similar data. Both generators' specifications are as follows:

Frame	268T Double C Face, Open Drip Proof
Speed	1800 rpm (4 pole)
Design	Two Value Capacitor, Reduced Voltage Start
Weight	Approx. 250 lb.
Other	Sealed bearings, Class F Insulation.

Slip-rings (A.2)

Slip-rings have been upgraded to the following:

Aeromotive Manufacturing Co.
Model No. SR-904
200 Amp, 600 Vac, 1 1/2-in bore
Copper Graphite Brushes, 200 rpm
Stainless steel brush springs
Zinc plated hardware.

Power Cable (A.3)

The electric cable should be suitable for direct burial or for running in conduit, have sufficient capacity for this application, and minimize power transmission losses. The following cable meets these requirements:

Type USE single conductor cable
#2 AWG Copper, 115 Amp rating.

Contactors (A.4)

The start-up inrush current is approximately 80 amps; therefore, a size increase is needed. The Start Contactor is an SS controls CA1-60-240AC, with capacity of 110 Amp, inductive.

The run contactor will be seeing many cycles, especially when winds are near cut-in windspeed. The mercury relay specified has much longer mechanical life than on originally specified unit, operates more quietly, and is well suited to inductive loads. The run contactor is a Durakool, Inc., Model DFC3-303 (135 Amp contact rating at 240 Vac).

Overload Protection and Safety Switch (A.5, A.6)

The lockable disconnect switch allows servicing of the turbine without fear of "hot" wiring. The fuses in the switch are designed to protect against generator overload. A suitable disconnect switch is:

> General Electric Model TG 4324
> Nema 1 Enclosure, 200 Amp contact rating
> 4 pole, 3 fuse; solid grounded.

Utility Interface (A.7)

The machine will be utility-interfaced through a circuit breaker. The circuit breaker must be sized to protect the conductors, meaning a 115 amp or less double pole breaker is needed. A suitable breaker is:

> General Electric Cat. No. TQAL21110, Amp.

Lightning Protection (A.8)

It is necessary to protect the generator and the customer's electric service from voltage surges induced by lightning. Switching surges will also occur as the generator is disconnected from the line. Metal oxide varistors (MOV'S) at the slip-rings and at the contactor box will shunt these surges to ground before they reach the generator or electric service. The following MOV'S are appropriate:

> General Electric V140LA20A (6 pcs)
> 150 Volts Maximum Line to Ground Voltage, 55 Joules peak energy, 6000 peak amps.

It is also recommended that power and control cables be run in metal conduit up the tower. This conduit serves as mechanical protection and as an electrical shield, thereby greatly reducing the magnitude of lightning-induced transients on power and control cables.

The tower and generator are separately grounded, and these grounds are interconnected as shown in Figure 32. Interconnection reduces current flowing through the yaw bearing (which would result in corrosion) due to differing ground potentials. Therefore, the conduit is grounded, allowing it to be an effective shield.

3.3.3 Three-Phase Design - Power Subsystem (Figure 33)

Generator (A.1)

The generator of the three-phase design is slightly more efficient than the single-phase generator and has less slip. The generator produces a great deal of startup torque and draws equivalent amperage to a 20 horsepower, 260 Vac motor, the purpose of the greater number of winding turns being to improve low output efficiency. A second generator (25 hp) has already been tested successfully. This generator appears to be a low efficiency design, but its frame may be used for the single phase generator to be custom wound. The three-phase generator expected to be used is:

Gould E-plus motor/generator
20 hp 3Ø wound to 260 Vac
1800 rpm, Open Drip Proof
Sealed bearings, Class F Insulation.

Slip-Rings (A.2)

Three slip-rings are needed for power and one for ground. Each ring must have a minimum capacity of 60 amps. The following slip-rings are satisfactory:

Aero-Motive Series SR-40
75 Amp, 600 Volt, Copper Alloy Rings
Copper Graphite Brushes, 200 rpm.

Power Cable (A.3)

The cable must have sufficient capacity and be suitable for underground as well as conduit installation. The following cable is suitable for an installation with up to 350 feet between contactor box and wind turbine:

Type USE Single Conductor Cable
#4 AWG Copper, 85 amp rating.

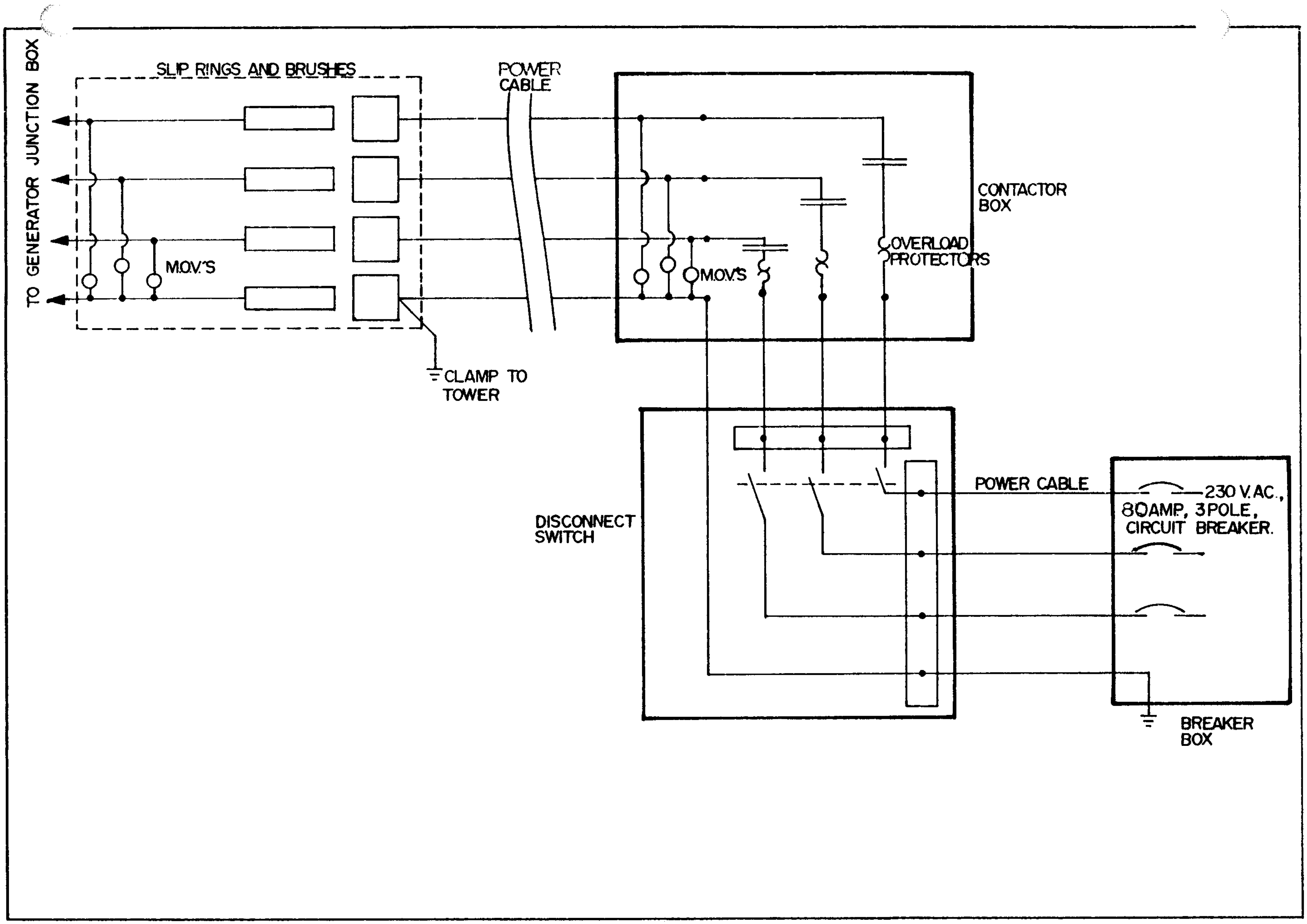


Figure 33

Three-Phase Electrical Power Subsystem Wiring Diagram

Contactor and Overload Protection (A.4, A.5)

The contactor must disconnect all ungrounded wires to the generator and have contacts sized for the horsepower rating of the generator. The overload protector must protect against prolonged overloads. A standard motor starter will meet these needs.

S&S Controls Cat. No. 1-40-240AC-72A
25 hp NEMA contact rating
Adjustable Heater 42-72A.

Safety Switch (A.6)

The disconnect switch is located near the connector box and disconnects all power to the wind system. It is lockable in the open (off) position to guarantee tower worker safety. The following switch is appropriate:

General Electric Model TGN3323
3 pole, No fuse, Solid neutral
100A Contact rating.

Utility Interface (A.7)

The power cable is rated for 85 amps, therefore any circuit breaker with a smaller rating will protect it. The following breaker is appropriate:

General Electric Cat. No. THQAL32080
Three Pole, 80 amps.

Lightning Protection (A.8)

As in the single phase design, metal oxide varistors are used in conjunction with grounded metal conduit to protect both the generator and power switching components from lightning - induced voltage transients. MOV'S are placed between each phase and ground at the contactor and slip rings.

6 pcs. GEV250LA404
250 volts maximum line to ground voltage
90 joules peak energy, 6000 amps peak current.

3.3.4 Evaluation and Conclusion

The electrical system (see Figure 34) appears to be flexible, simple and reliable. The induction generator design offers simplicity and low cost with moderate efficiency. The single-phase design will not be as efficient as the three-phase design at all operating levels. The power cable is sized to minimize transmission I^2R losses.

The electrical power quality is similar to that of an induction motor, thus presenting no particular problems to the utilities. Voltage will vary little at the utility tie-in, with no waveform distortion. Current varies with generator output. The waveform is a consistent, slightly distorted sine wave (typical of motors); with the difference being that instead of current lagging voltage by zero to 90 electrical degrees, it lags by 90 to 180 degrees.

Standard, time-proven electrical components characterize the electrical design. Induction motor/generators, slip-rings, cable, motor starters, and safety switches have been used for years in industrial electrical applications. The "bugs" have been worked out of their designs and the components are available as off-the-shelf hardware, making system maintainability excellent. In that all components are already mass-produced for a variety of applications, component costs are low.

Heat dissipation characteristics of the electrical system are excellent. The cable is sized to keep temperatures well below insulation ratings. The generator is of the open drip-proof design (as opposed to totally-enclosed fan-cooled), allowing a 1.15 service factor due to superior cooling characteristics. It should also be noted that the greatest need for heat dissipation occurs when cooling conditions are best. The highest power output (and greater cooling need) occurs when air density is high (i.e. low air temperature) and at high windspeeds.

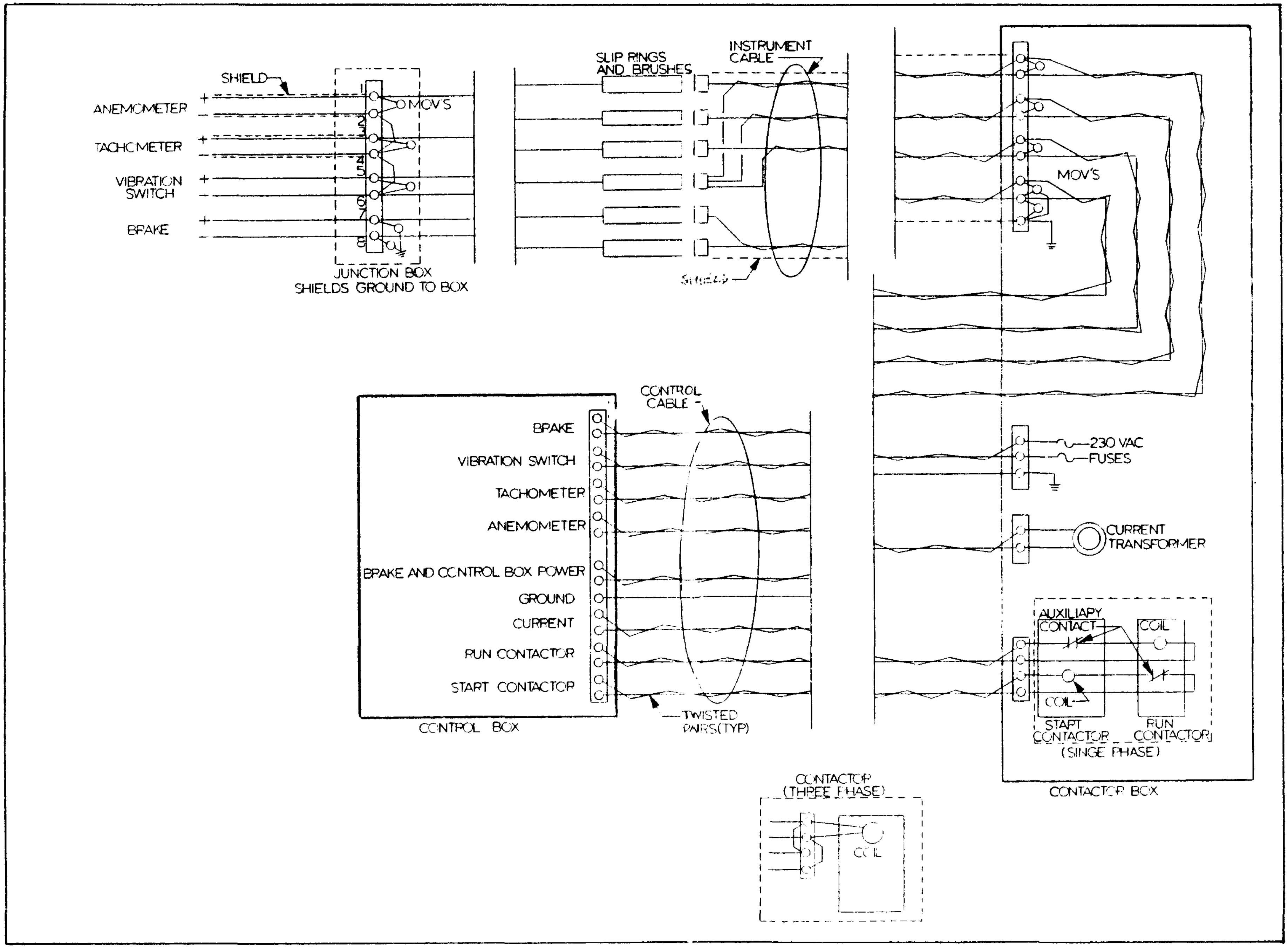


Figure 34

Single-Phase/Three-Phase Wiring Diagram

System voltage is effectively constant, and maximum current conditions are handled by both the cable protective device (circuit breaker) and the generator protective device. Some utilities may be concerned about the potential inrush current of such a large single phase motor. The fact that it will be wound to a higher voltage allows for a "softer" start, however. The systems reduced-voltage starting keeps it well within acceptable limits.

The system is well protected from lightning or other voltage transients as detailed previously. A minimum of electrical equipment in the nacelle improves serviceability and reliability.

An interesting characteristic of the generator is the following. As load increases, line amperage also increases, causing a voltage-drop in the line which lowers system efficiency. Compensating for this, however, is the fact that lower line voltage causes an increase in generator voltage. As generator voltage increases so does generator efficiency. The net result is that higher line amperage does not lower system efficiency as much as might be expected.

3.4 Control Subsystem Design

The function of control system elements is discussed and evaluated in this section, together with electrical design. Figure 35 shows the control system hardware configuration. Note that the tip brakes, although a part of overspeed control, are discussed in the rotor design section.

3.4.1 Description of Operation

The controls have been designed to make their operation easily understood by the customer. Looking at the control panel (Figure 36) the customer can immediately see the state of the system. If it is running, the "run" light is on. If it is shut down, (i.e. generator disconnected from the utility line), one of the "shutdown" lights is on. If the shutdown requires resetting, the reset backlighted push button is on. System power output, rotor speed and windspeed are read from the meters. Although it is operationally simple, the control logic is more complex.

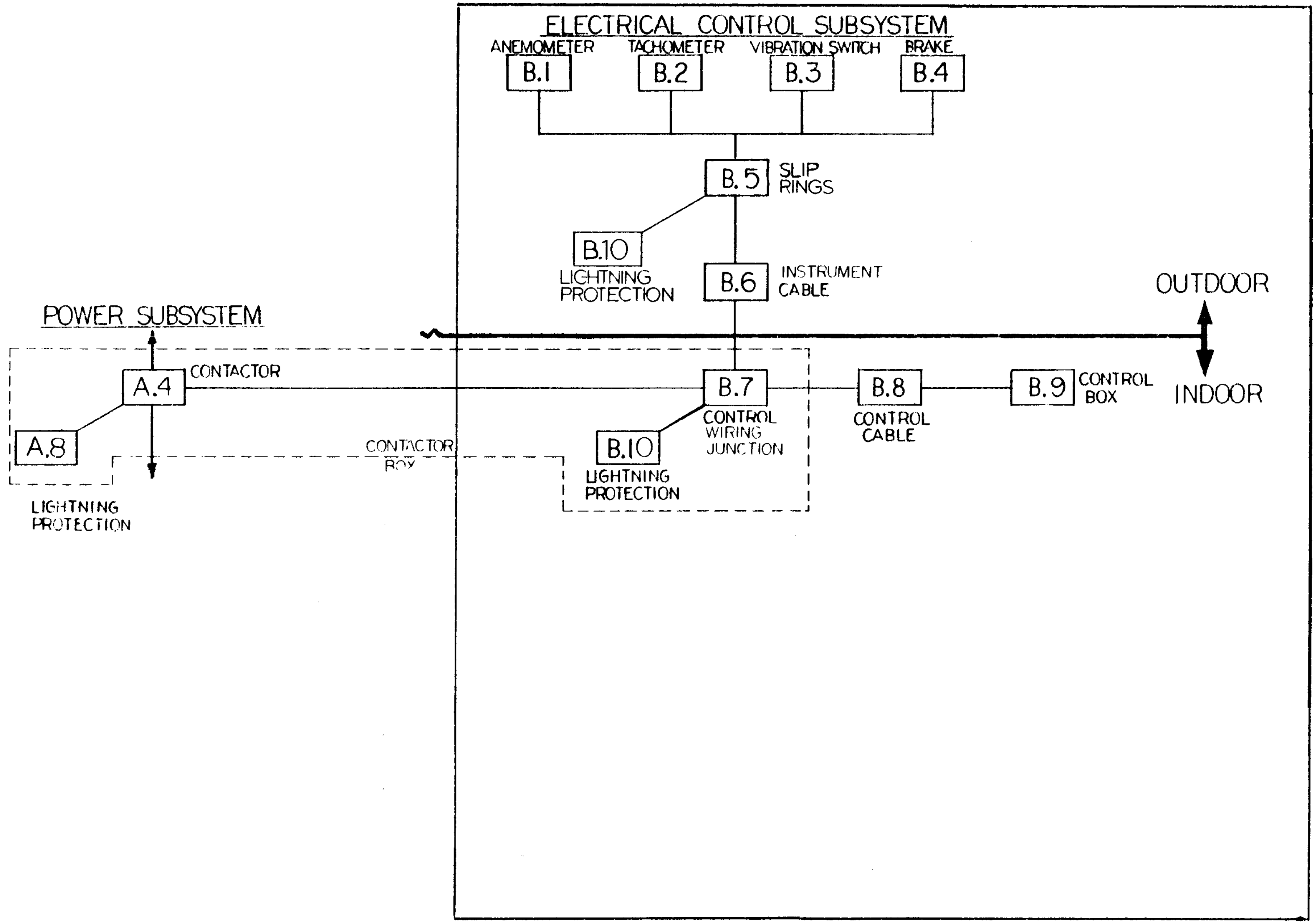


Figure 35

Electrical Control Subsystem Block Diagram

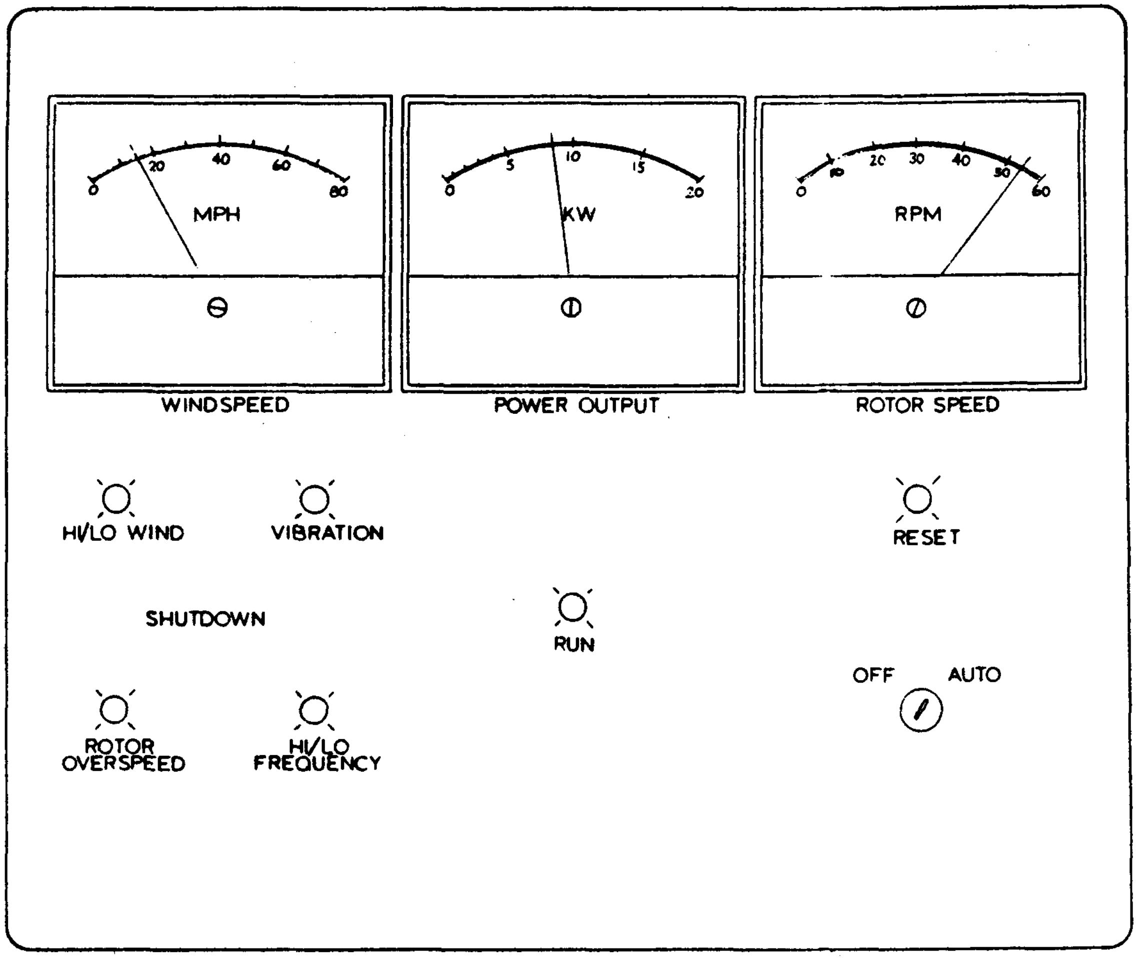


Figure 36
Control Box Layout

3.4.2 Control Elements Function

The anemometer is responsible for initiating system startup when windspeed is sufficient for power generation. The windspeed signal also actuates a high windspeed shutdown.

The tachometer senses generator rotational speed and initiates several control functions. A generator overspeed condition (2000 rpm) actuates a resettable system shutdown. When generator speed drops below synchronous (1800 rpm) the machine is disconnected from the utility line and the brake remains disengaged. When generator speed increases above synchronous speed (1820 rpm) it is reconnected to the utility line. The control system is returned to windspeed-based control when generator speed drops below 1000 rpm. The brake comes on at very low generator speeds (100 rpm) requiring a windspeed-based restart. The machine therefore freewheels at low windspeeds, lessening the number of stop/start cycles and reducing brake wear and energy consumed in motoring the generator to operating speed. Control System Operation is summarized in Table X.

The vibration sensor, upon excess vibration of the bed frame, initiates a resettable shutdown in the control electronics.

The brake is disengaged whenever it receives a 230 Vac coil signal from the control electronics. Otherwise it is engaged. This means that the brake is automatically engaged when a utility power outage occurs. The brake is the final control element for all machine shutdowns. Whenever a shutdown is initiated the brake is engaged. The brake is disengaged, however, when the machine is operating on generator speed-based control whether the machine is generating power or not. The brake is rated at 125 ft-lb of torque. Calculations have shown that a shutdown at maximum torque (maximum output) necessitates 102 ft-lb of torque.

The "hi-lo" frequency shutdown logic is located in the control box. It senses utility line frequency and initiates a shutdown when frequency

Table X

15kW Control System Operation

System Condition:	Control Action: Start Contactor	Run Contactor	Brake	Comments
Startup functions				
1. Gen. stopped. Avg. windspeed at 10 mph for 50 seconds.	closes	already open	disengages	Turn over to speed control at 1000 rpm
2. Gen. freewheeling below 1000 rpm avg. windspeed at 10 mph for 1 minute	closes	already open	already disengaged	
3. Gen. Freewheeling above 1820 rpm	already open	closes	"	
4. Gen. freewheeling between 1000 & 1800 rpm	opens	already open	"	
Shutdown Functions:				
1. Gen. speed drops below 1800	already open	opens	already disengaged	
2. Gen. speed drops below 1000	already open	already open	"	System returned to windspeed control
3. Gen. speed drops below 100 rpm	"	"	engages	
4. Gen. speed exceeds 2000 rpm	"	opens	engages	Manual reset needed.
5. Windspeed exceeds 40mph for 50 seconds	"	opens	engages	
6. Excessive machine vibration	"	opens	engages	Manual reset needed.
7. Utility frequency <59 or >61 Hz	opens	opens	engages	
8. Utility power loss	opens	opens	engages	
9. Machine fails to reach 1000 rpm 20 sec. after start contactor closes	opens	already open	already open	

exceeds or drops below preset limits. This is primarily to provide positive prevention of utility backfeed upon power outage.

The contactors are the second final control element and are used to connect the machine to utility power. The connection command can be the result of windspeed, generator speed, or hi-lo frequency data.

Testing of the system is facilitated by the use of a "test" switch located in the control box. It is a momentary contact push button which, when depressed, gives a simulated 45 m/s (100-mph) test signal to the anemometer input circuitry. This causes the machine to start up (if stopped) and then immediately shutdown as if high windspeeds were occurring. This allows the machine to be started in no (or low) wind conditions for testing purposes.

3.4.3 Control System Hardware

Anemometer (B.1)

We have had excellent experience with the anemometer recommended below. It is of the rotating permanent magnet type. Its ouput frequency is linear with windspeed and extremely immune to noise contamination. This is important in that at low windspeeds the generated signal is a low level voltage, but the frequency of that signal will not vary with noise. The following anemometer is recommended:

Maximum Model 40
2 Cycles per revolution

Tachometer (B.2)

The tachometer will also be of the proven rotating permanent magnet design. The frequency signal is converted to a useful form for the control logic and meter readout. It will mount in the brake housing, sensing motor shaft-end angular velocity. The following tachometer is specified:

Bunting 6-pole side-pole rotor
Catalog No. SP947
3 Cycles per revolution

Vibration Switch (B.3)

The vibration switch activates a resettable shutdown function in the control logic whenever potentially damaging vibration occurs at the generator. It will mount on the frame assembly. An adjustable switch is specified for the prototype machine, with a fixed version to be used once a proper setting is determined. The following switch is specified:

PMC/Beta Model 414B
.3-3 in/sec adjustment range

Brake (B.4)

A hydraulic brake has been designed for use on this generator. We have had good experience with hydraulic brakes on our smaller units. The details of the pump are shown in Figures 37 and 38 and in Table XI.

Slip-Rings (B.5)

The control slip-rings transmit four signals: anemometer, tachometer, vibration switch and brake control. The primary concern is signal distortion and the signal most prone to distortion is the anemometer signal. At low windspeeds the anemometer signal has low voltage and low current. Minimal brush resistance is desirable. The following slip-rings are suitable for this application:

Aero-Motive Model AG-312
12 Conductor, 1 1/2 in bore, 35 amp
Silver contact brushes, silver plated rings, stainless steel brush springs and mounting clips, zinc plated steel mounting hardware. Maximum temp 120°C, Maximum speed 200 rpm, continuous duty.

Instrument Cable (B.6)

The control cable must be configured to minimize noise pickup from the adjacent power cable. It must also be sized to minimize voltage drop.

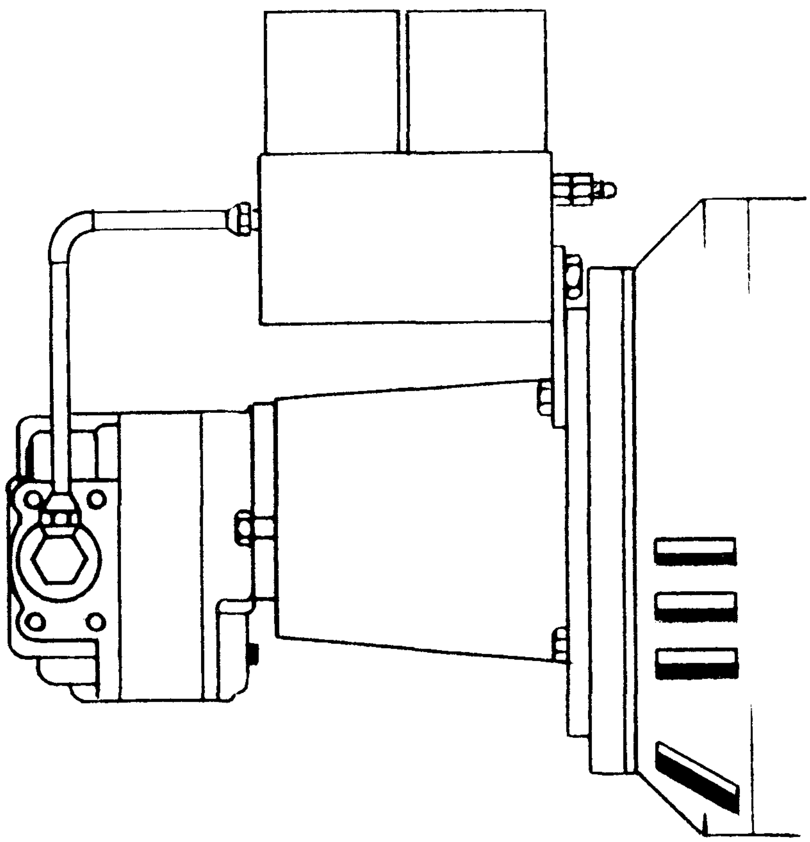


Figure 37
Hydraulic Brake Pump

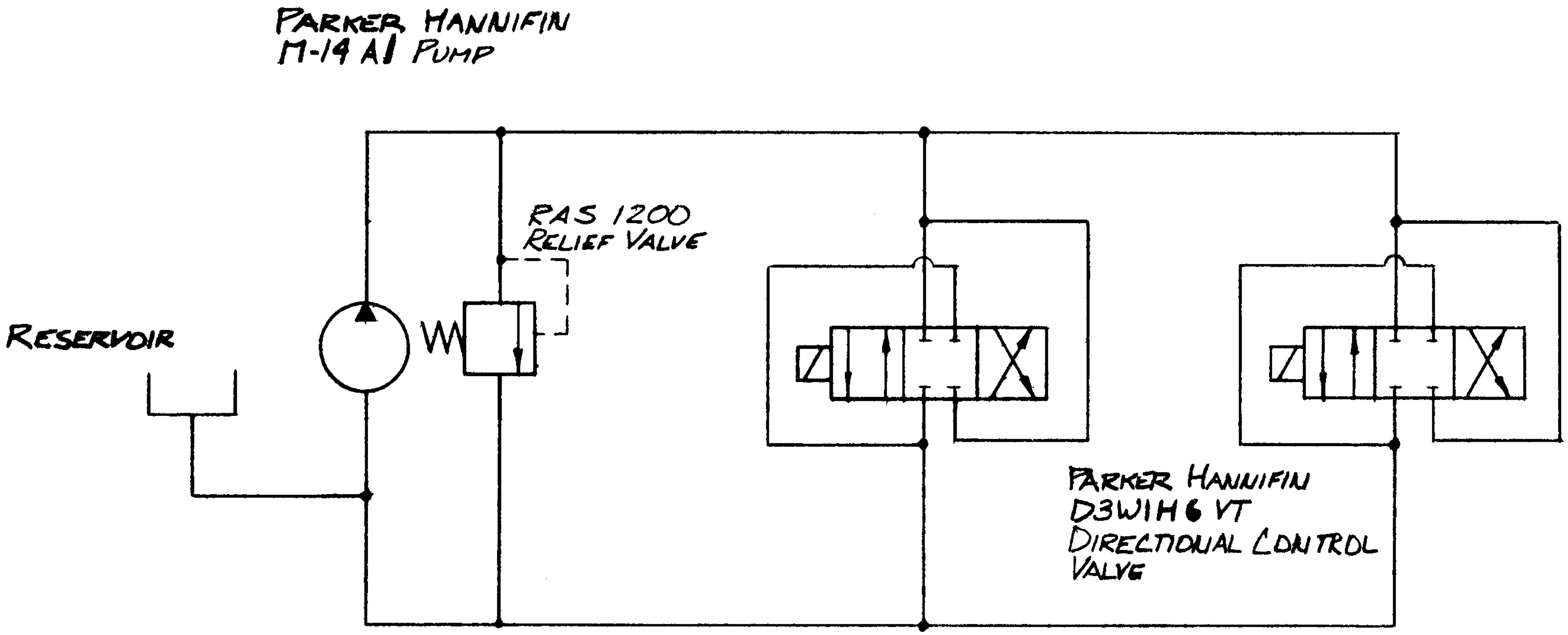


Figure 38

15 kW Hydraulic Schematic

Table XI

Parts List - Hydraulic Brake

Description	Model/Manufacturer
Adaptor Ring	Parker Model AR-1
Pump Adaptor Flauge	Parker Model MM
Solenoid Valves	Parker Model D 3W1H6T
Valve Manifold	Parker SP2D34A
Gear Pump	Parker M14A1A1CB250-00
Pressure Relief Valve	Parker RAS1200

The brake draws .72 amps at 230 Vac when in the disengaged position. It is necessary to maintain brake coil voltage to at least 200 Vac. A #18 AWG copper conductor is adequate to handle inrush and operating current. Capacitive noise pickup is minimized by utilizing shielded cable and inductive noise coupling is taken care of by specifying twisted pairs of conductors.

Alpha Control Cable No. 5376
#18 AWG 6 twisted pair w/overall shield copper
conductor, 600 volts, 60°C

Control Wiring Junction (B.7)

The control wiring junction provides an interface between control box wiring, power wiring, and remote control system elements. It is located in the contactor box.

Control Cable (B.8)

This cable interconnects the control box with the control wiring junction in the control box. A qeneral purpose control cable is sufficient.

Alpha Control Cable No. 5616/1801
8 Twisted pair w/overall shield

Control Box (B.9)

The control box is shown in Figure 36. The box itself is a Nema 12 panel enclosure. It houses necessary meters, switches, lights and control electronics as discussed previously. It is meant for indoor mounting at a location remote from power equipment and wiring, thus minimizing electrical noise contamination.

Lightning Protection (B.10)

It is desirable to protect control electronics from voltage spikes induced by lightning. This is simply accomplished through the use of small metal oxide varistors (MOV's) shunting surges to ground at the control junction box. Appropriate MOV's for this function are:

General Electric MOV,
Model V22ZA3, 14V max line to ground voltage,
3 joules maximum energy dissipation, 1000A max
current Model V130 LA1, 130 V max line to ground
voltage, 4 joules maximum energy dissipation, 500A
max current

(NOTE: The primary means of lightning defense are 1,
the metal conduit surrounding the instrument cable and,
2, the instrument cable shield.)

3.4.4 Control Response

Observation of the response of other induction machines created concern about excessive on/off cycling of the generator during marginal wind conditions. This has led to minor modifications of the control response.

As the machine freewheels up to synchronous speed it is desirable to have the machine connected to the utility as quickly as possible once generation speed is reached. This lessens power surges upon system connection.

Fast response is not nearly as critical when the wind drops and the machine slows to below synchronous speed. By slowing response at this point, excessive cycling will be reduced during marginal wind conditions.

Therefore, the system response time with increasing rpm will (initially) be set at 0.5 second, while a minor circuit redesign will allow a 2-second response time with rpm decreases.

3.4.5 Conclusion

The control system design has been kept as simple as possible without compromising safety or net power generation. Because the electrical power subsystem is very simple, a slight increase in control system complexity to allow for efficient system operation is justified. The system is functionally simple with a self-explanatory control display. The system installation is also simple, requiring only wiring to and from readily accessible and clearly marked terminal points. The solid state control logic will be fairly complex. The logic is extremely reliable and will be tested prior to system installation. Environmental operating conditions will not be a problem in that the control box is mounted indoors.

All components except the control logic circuitry are standard, mass-produced equipment. After much thought and discussion, the system proposed seems to embody the minimum control cost while satisfying operational criteria. Marketability considerations dictate the use of a simple, understandable control system at reasonable cost. Installation costs will be low in that point-to-point wiring is all that is required.

The system is easily maintained, being modular in nature. It is anticipated that maintenance will take place on a "repair by replacement" basis. If a control board fails it will be replaced and the faulty one returned to the factory for inspection and (if necessary) rebuilding. The components located at the machine are quite reliable and easily replaceable. Other components are easily replaced without climbing the tower.

If the control system is not mounted indoors, a weather-proof box would be specified and heater strips mounted within to counteract the effects of temperature extremes on the control electronics.

3.5 Test Model Program

To verify several 15 kW system design and component concepts, a series of tests were carried out with standard and modified components of the prototype Enertech "4000" (E-4000) wind generator. Tests were carried out from March 24 through May 22, 1980, at locations including Rutland, Windsor, Norwich and Union Village in Vermont as well as on top of Mt. Washington in New Hampshire.

The rotor diameter of the E-4000 used for these tests was 6 meters, making it approximately a .45 scale model of the proposed 15 kW design.

Test results for each of the seven tests conducted are summarized below.

3.5.1 Generator Bench Tests

This test was conducted in order to establish the characteristics of the generator when it was feeding into a normal utility line and to learn how to simulate this operation as closely as possible when operating with a remote generating unit and load bank. The characteristics of the generator are shown in Figures 39 and 40.

3.5.2 Performance Tests

Figure 41 shows the output curve obtained from the E-4000 test model both at zero degrees and at 45 degrees yaw angle without tip brakes. Figure 41 also shows the effect of adding tip brakes to the blades. As can be seen these end plates had virtually no effect on performance except above 13.4 m/s (30 mph) where output was lowered very slightly. This was considered, if anything, desirable so the concept of end-plate type tip brakes has been retained for the 15 kW design.

Figure 42 shows the test model output curve corrected to a temperature of -6.7°C (20°F). As can be seen, the shape of the curve is very similar to the E-1500 curve at the same temperature. The actual values obtained were

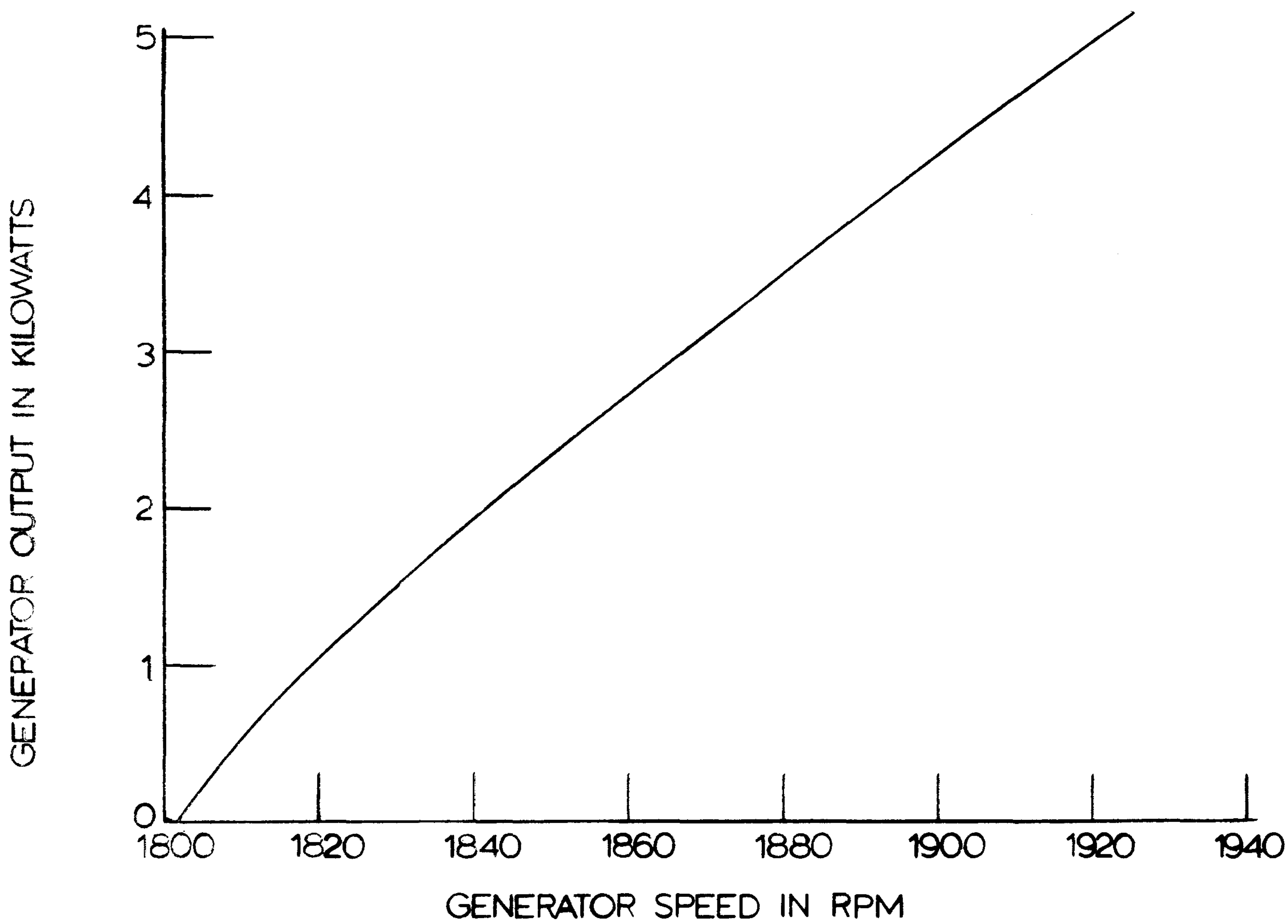


Figure 39

Enertech 4000 Generator Output Characteristics

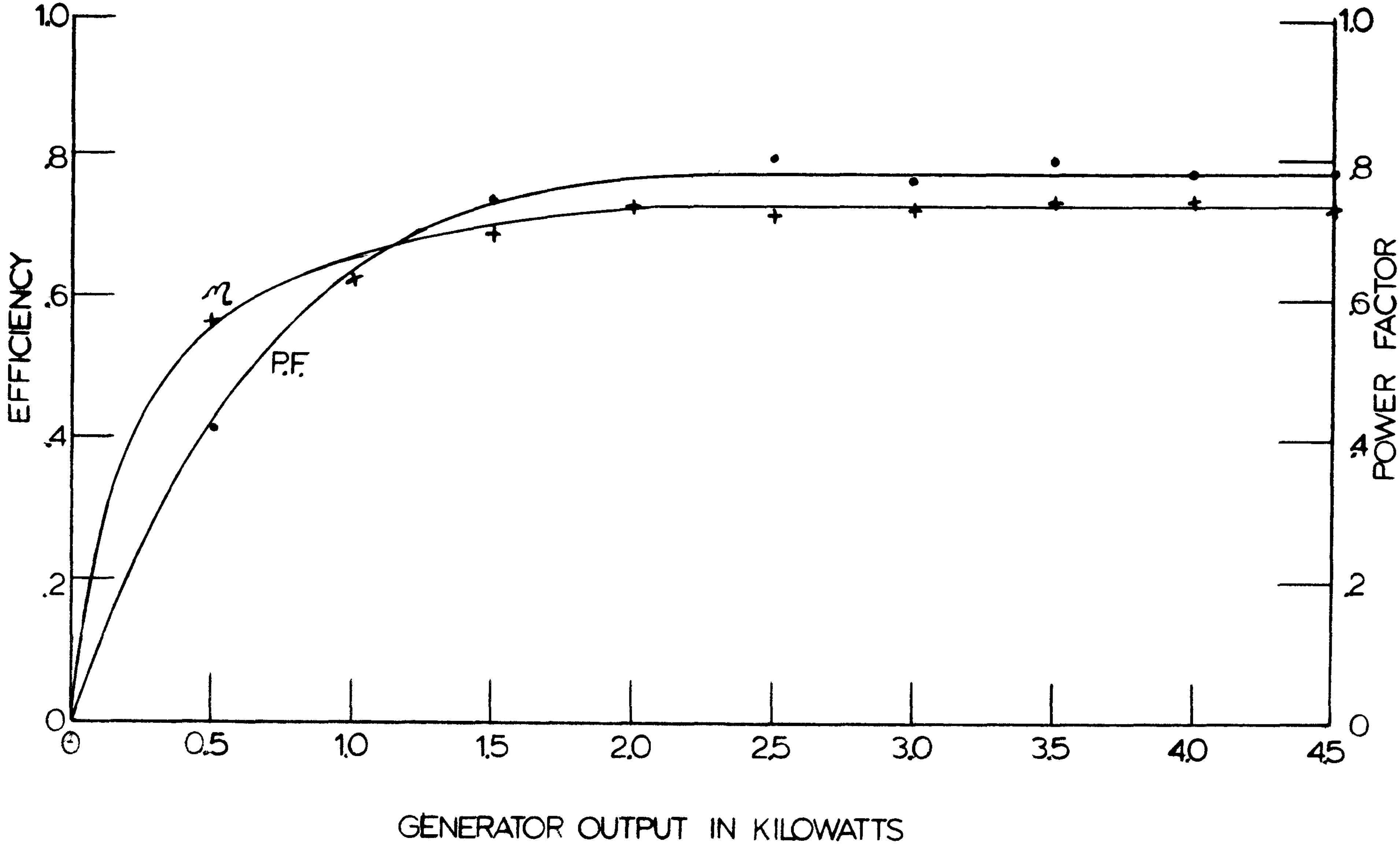


Figure 40

Enertech 4000 Generator Characteristics

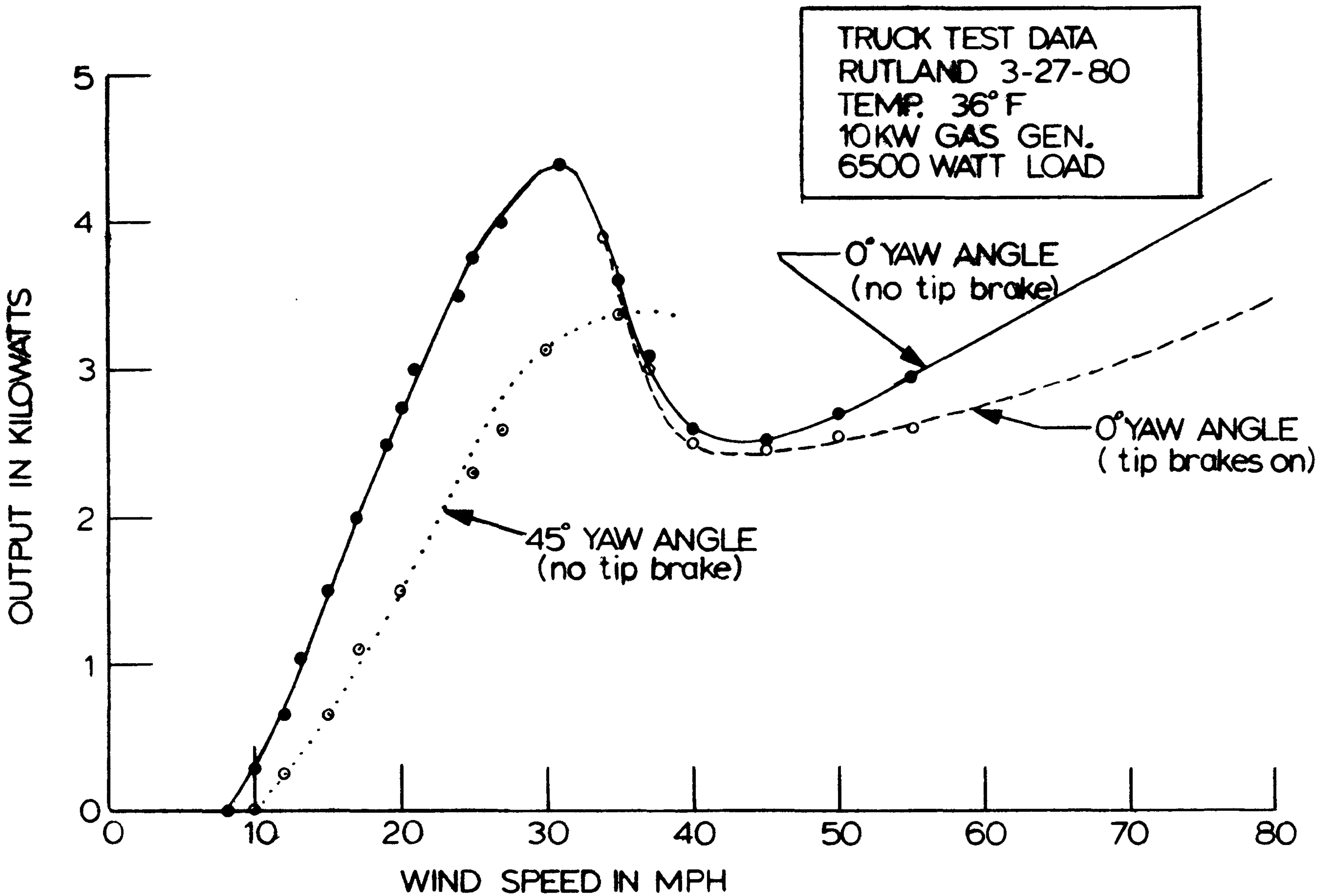


Figure 41

Enertech 4000 Performance Test Results
(2.5° Blade Tip Pitch)

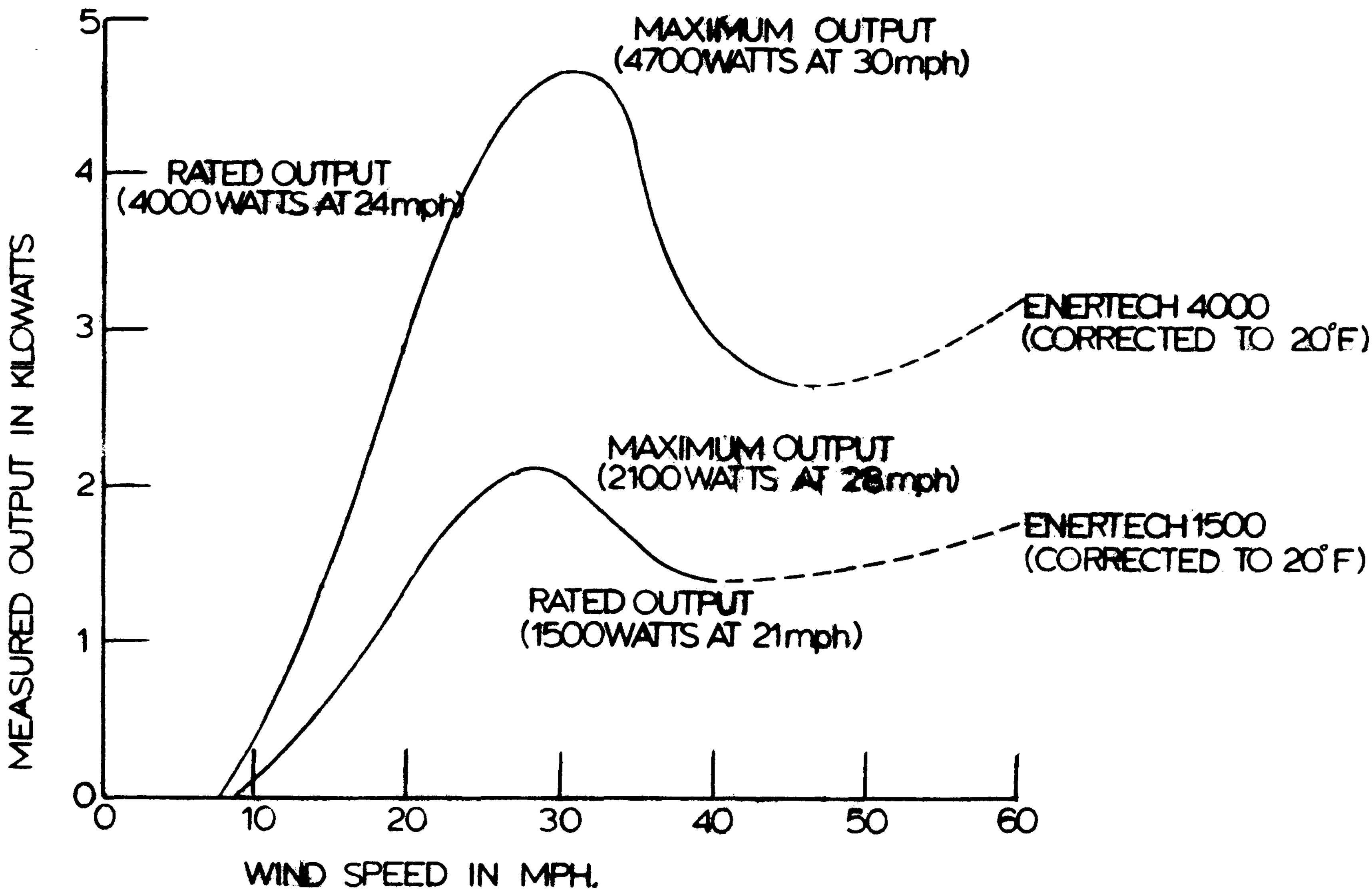


Figure 42

Enertech 4000 Output Curve (Corrected to 20^{o}F)
Compared to E-1500

very close to the values predicted by multiplying the E-1500 data by the swept area ratio of the E-4000 to the E-1500 - or a factor of 2.25. The chief difference noted is that the E-4000 output drops off slightly more after the peak output point than the E-1500 output, indicating a more pronounced stall effect.

The results of this test confirmed Enertech predictions for this size unit and thereby gave added confidence to predictions for the full-scale 15 kW machine. As a result of the model test, the originally proposed effective blade pitch settings were retained. (The actual pitch settings are slightly different to take into effect the thickness variation of the 15 kW blade which runs from 10% at the tip to 28% at the root compared to 12% and 22% respectively for the E-4000.)

The effect of varying the blade pitch is shown in Figure 43. As can be seen a flatter blade pitch results in a lower peak output which is reached at a lower wind speed. Present plans call for the 15 kW blade to be adjustable (plus or minus one degree) from the nominal setting to allow fine tuning of the prototype to the desired peak output value after final assembly.

Finally, a self-starting test was conducted on the E-4000 test model to determine the start-up characteristics of the unit. The results were as follows:

> Conditions: Temp. 10°C, brake released, rotor stopped
> Results: Minimum wind speed to start rotor turning: 9.8 m/s
> Measured breakaway torque on rotor: 20.5 ft-lb
>
> (Note: It is assumed that the breakaway torque would decrease as the drive train components became broken in; however, in its present state the unit clearly does not have acceptable self-starting characteristics.)

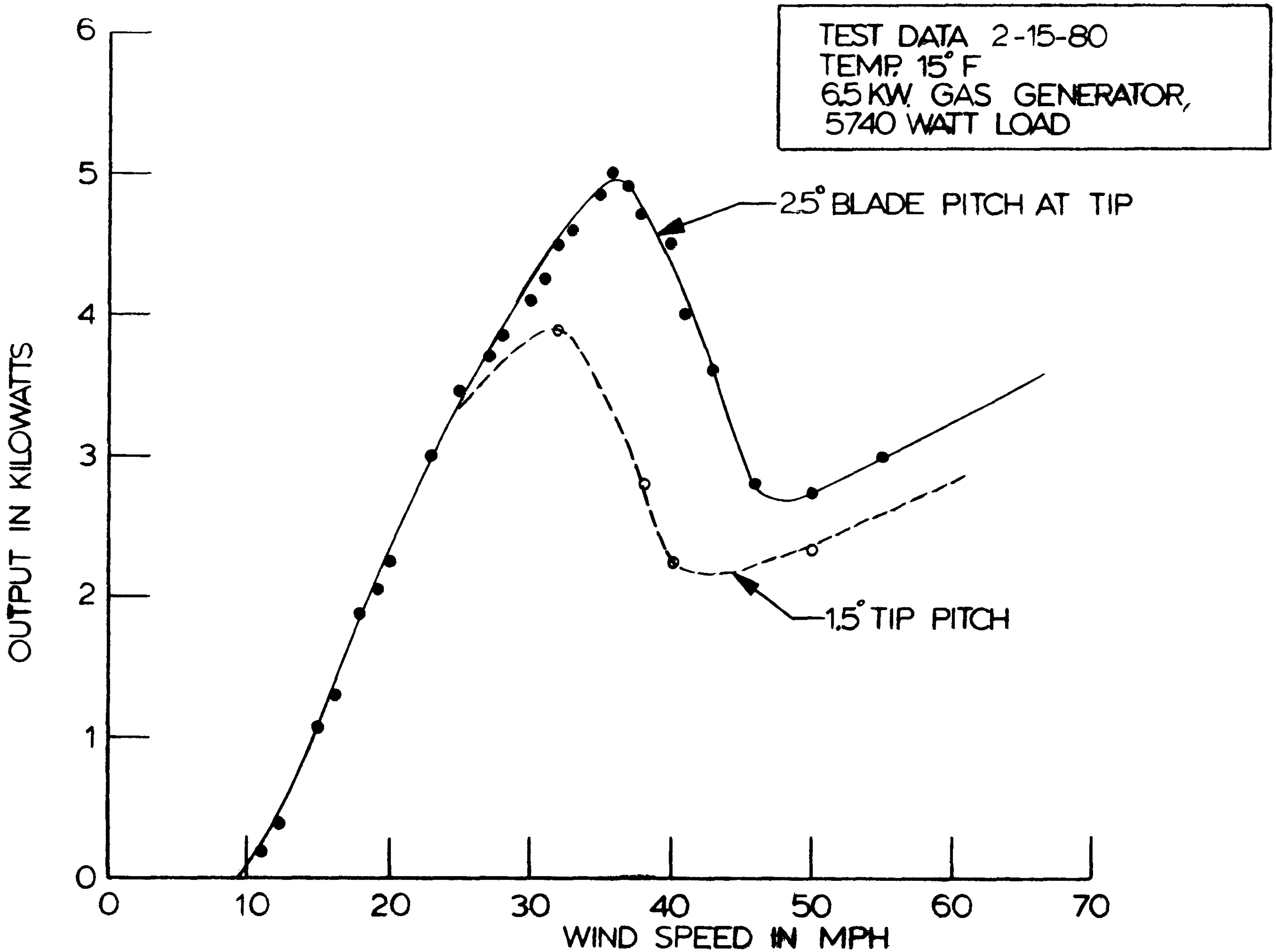


Figure 43

Enertech 4000 Output at Various Blade Pitch Settings

3.5.3 Thrust Tests

Thrust measurements were obtained by mounting the test model drive train assembly on a sliding bed and attaching it to a calibrated hydraulic cylinder. This test set-up is shown in Figure 44.

Results of the thrust tests are shown in Figure 45. These show close agreement with theoretically predicted results, especially around 18 m/s--the speed at which fatigue loadings were calculated--giving confidence in these calculated values.

3.5.4 Yaw Force Tests

The calibrated hydraulic cylinder used for thrust measurements was also used in a different position to measure yaw forces on the Enertech E-4000 test unit when set at various angles of yaw both to the right and left of center (zero yaw position). The test-rig configuration used for these tests is shown in Figure 46.

Figures 47 and 48 show the results of yaw force tests. The data shown are for a bed-frame length of 26 inches. Although longer bed frame lengths were tested, the results showed little variation from the data presented here. The pattern was always the same: restoring force increased up to a certain wind speed, then it began to decrease, actually becoming negative in some instances.

The test results show that the wind machine appears to have positive stability about the zero yaw point at low wind speeds; however, at higher wind speeds it is not clear just what position the unit would assume if it were free to yaw. The results indicate that the unit might track as much as 30° out of the wind in a 20 m/s wind, for example.

Figures 49 and 50 show yaw forces as a function of wind speed for a stopped rotor. This serves as a control test and indicates that the measuring equipment was working properly.

Figure 44
Thrust Test Set-Up on Test Truck

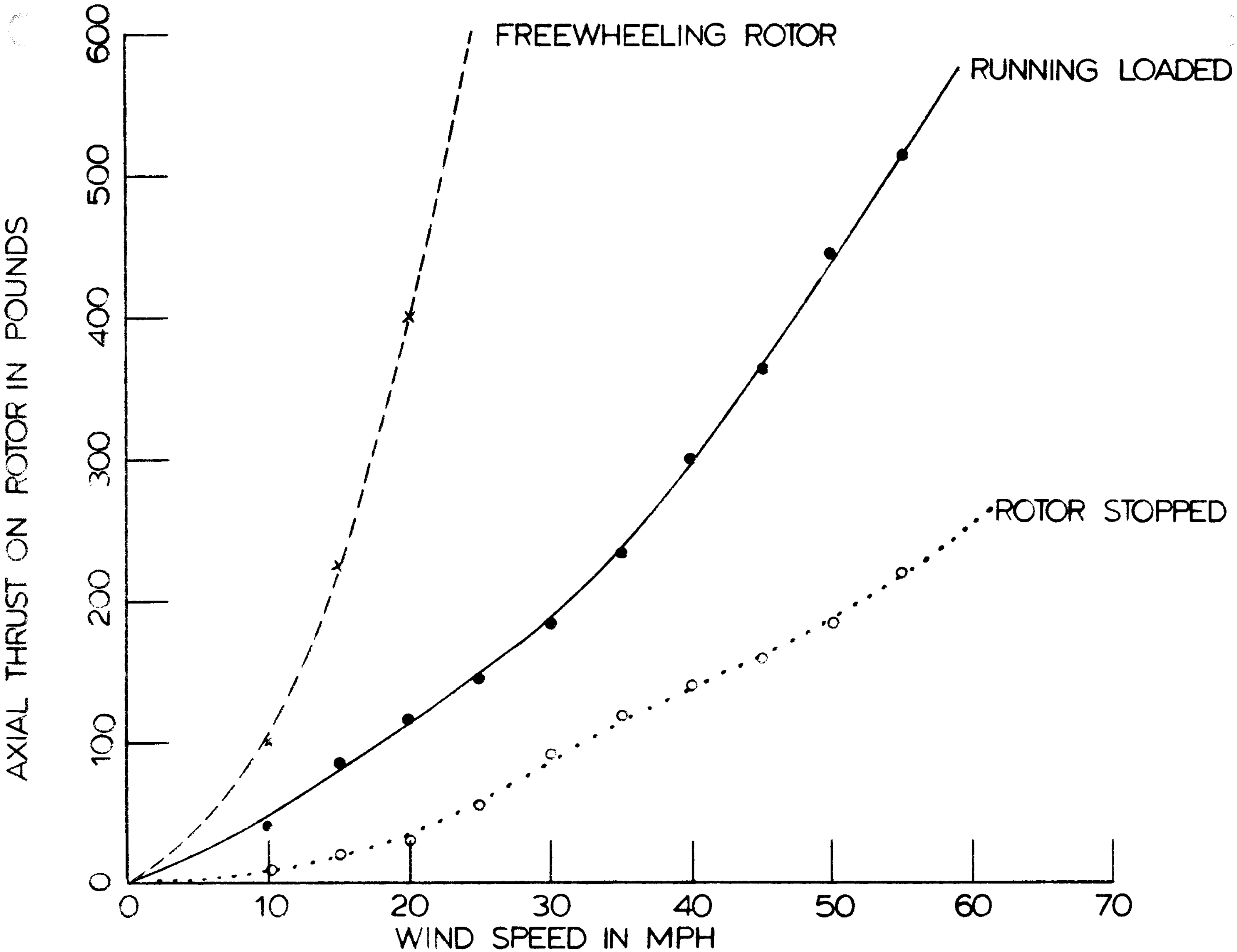


Figure 45

Enertech 4000 Thrust Test Results

Figure 44
Thrust Test Set-Up on Test Truck

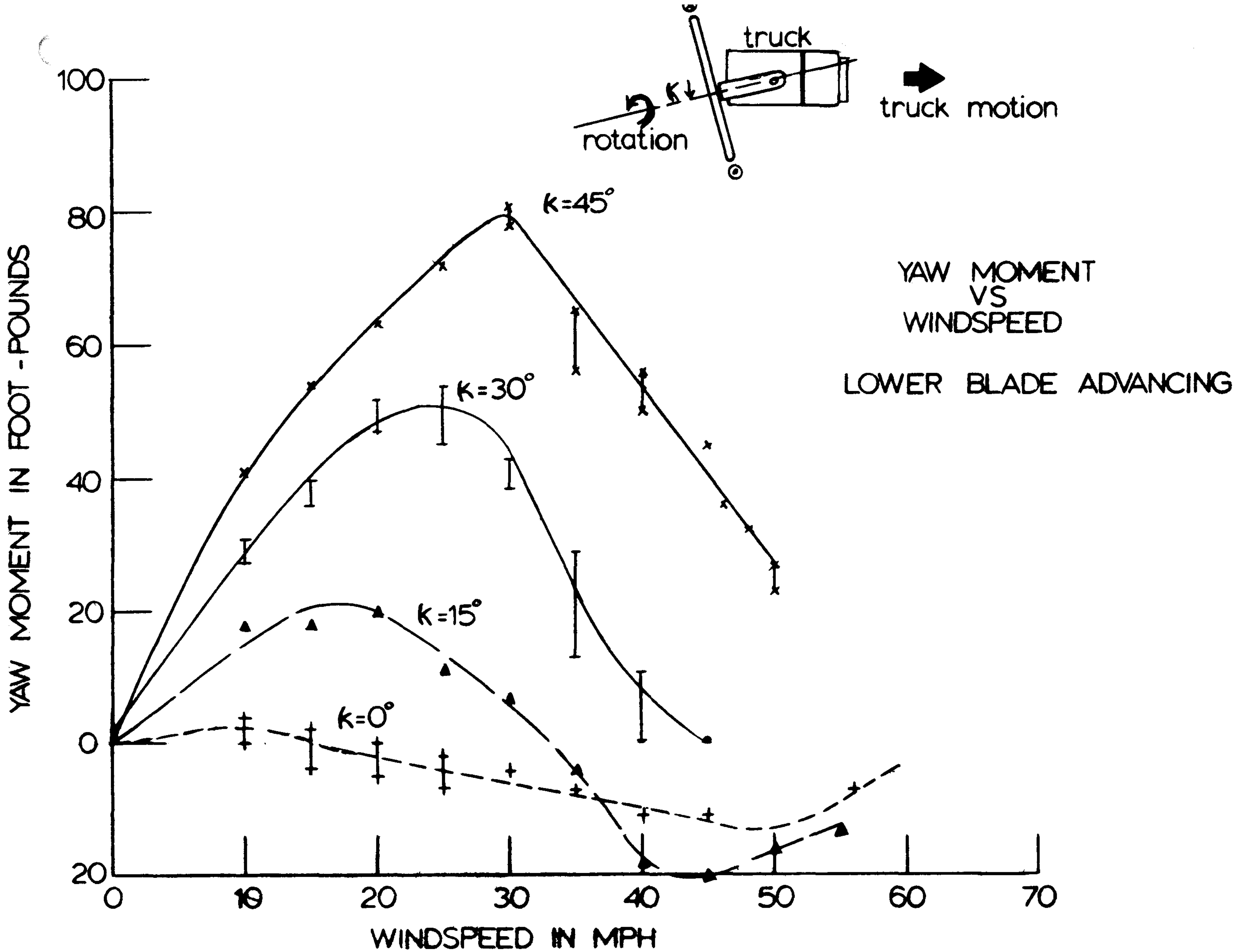


Figure 47
Yaw Moment Test Results - Lower Blade Advancing

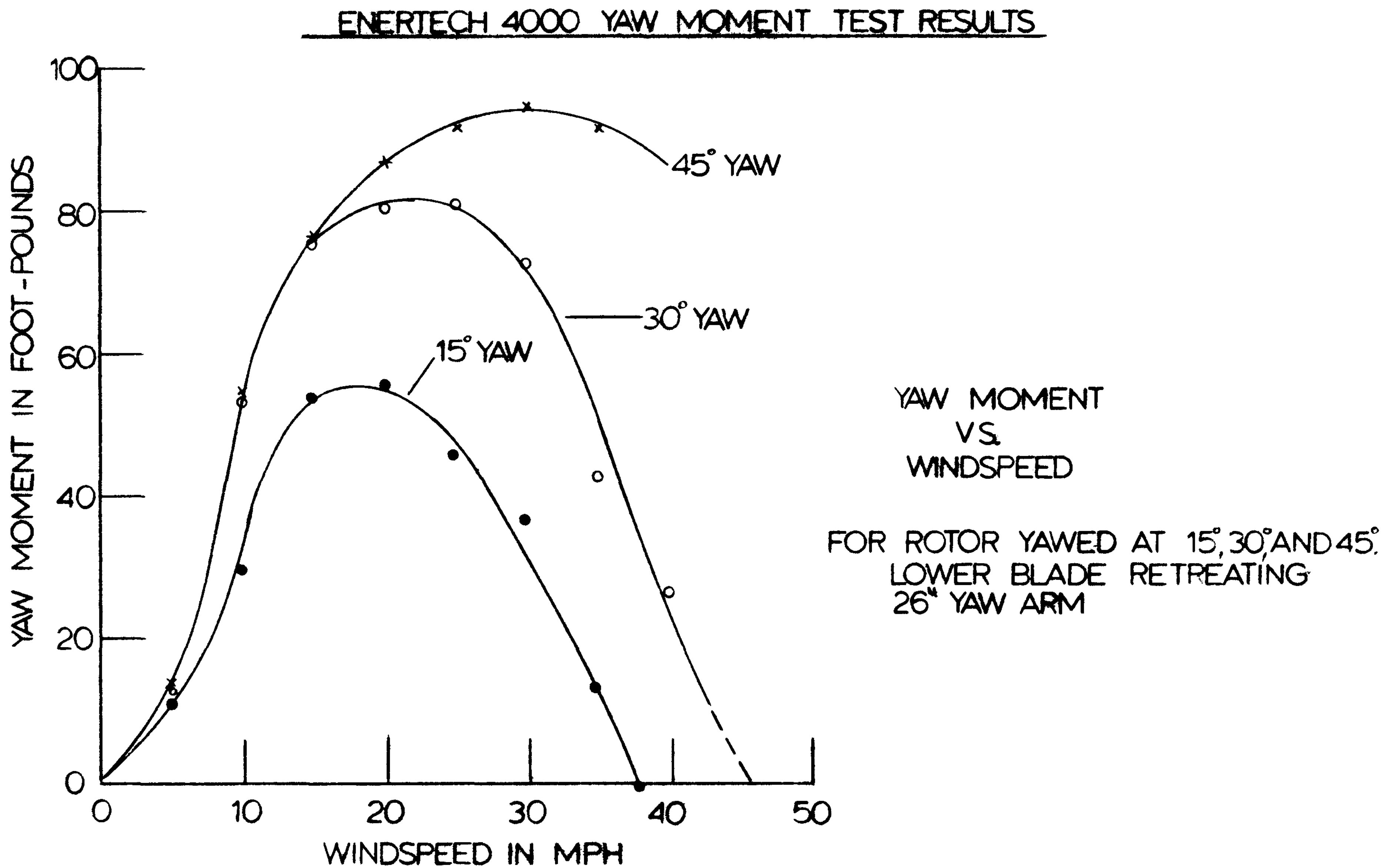


Figure 48

Yaw Moment Test Results - Lower Blade Retreating

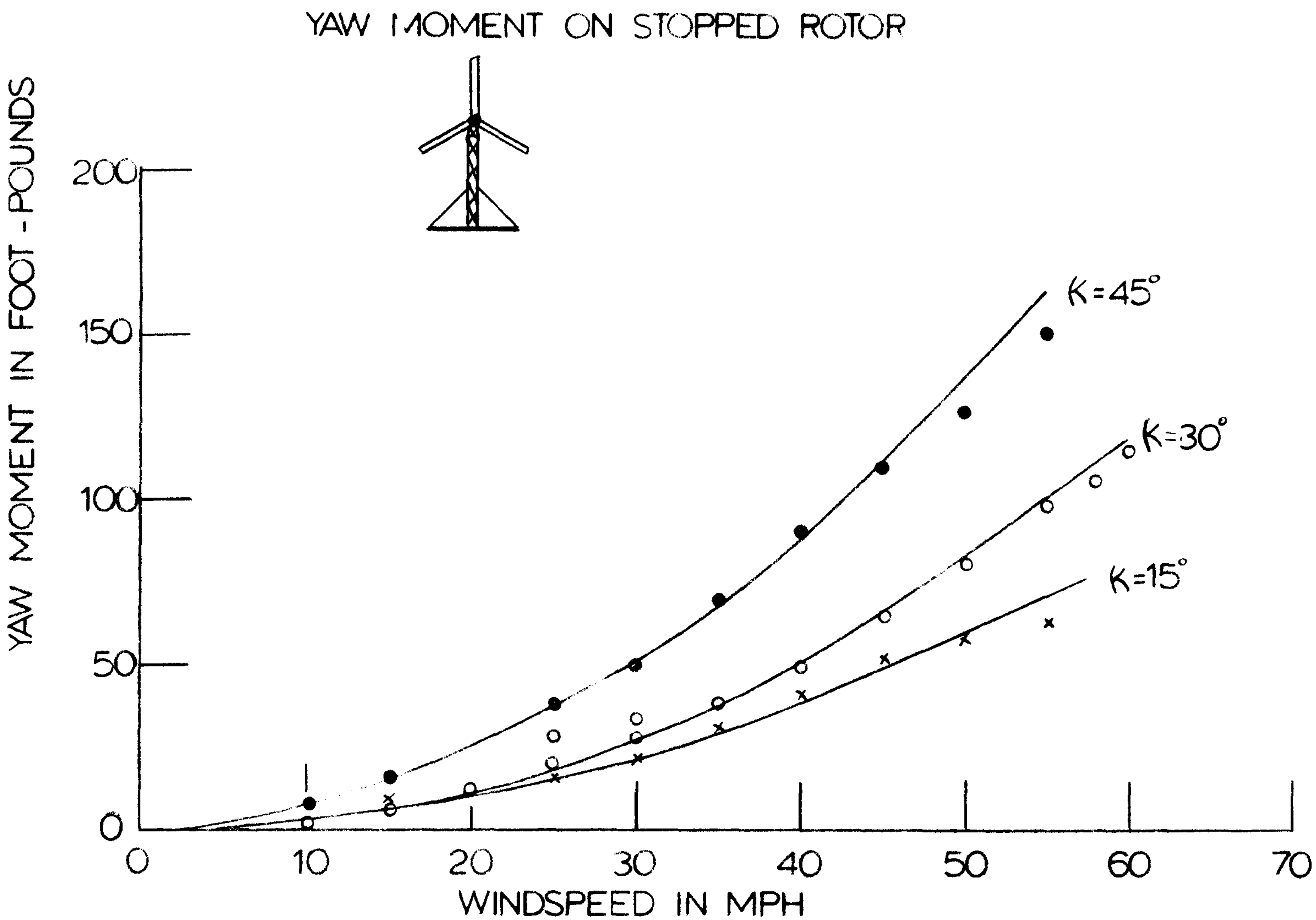


Figure 49

Yaw Moment Test Results - Stopped Rotor

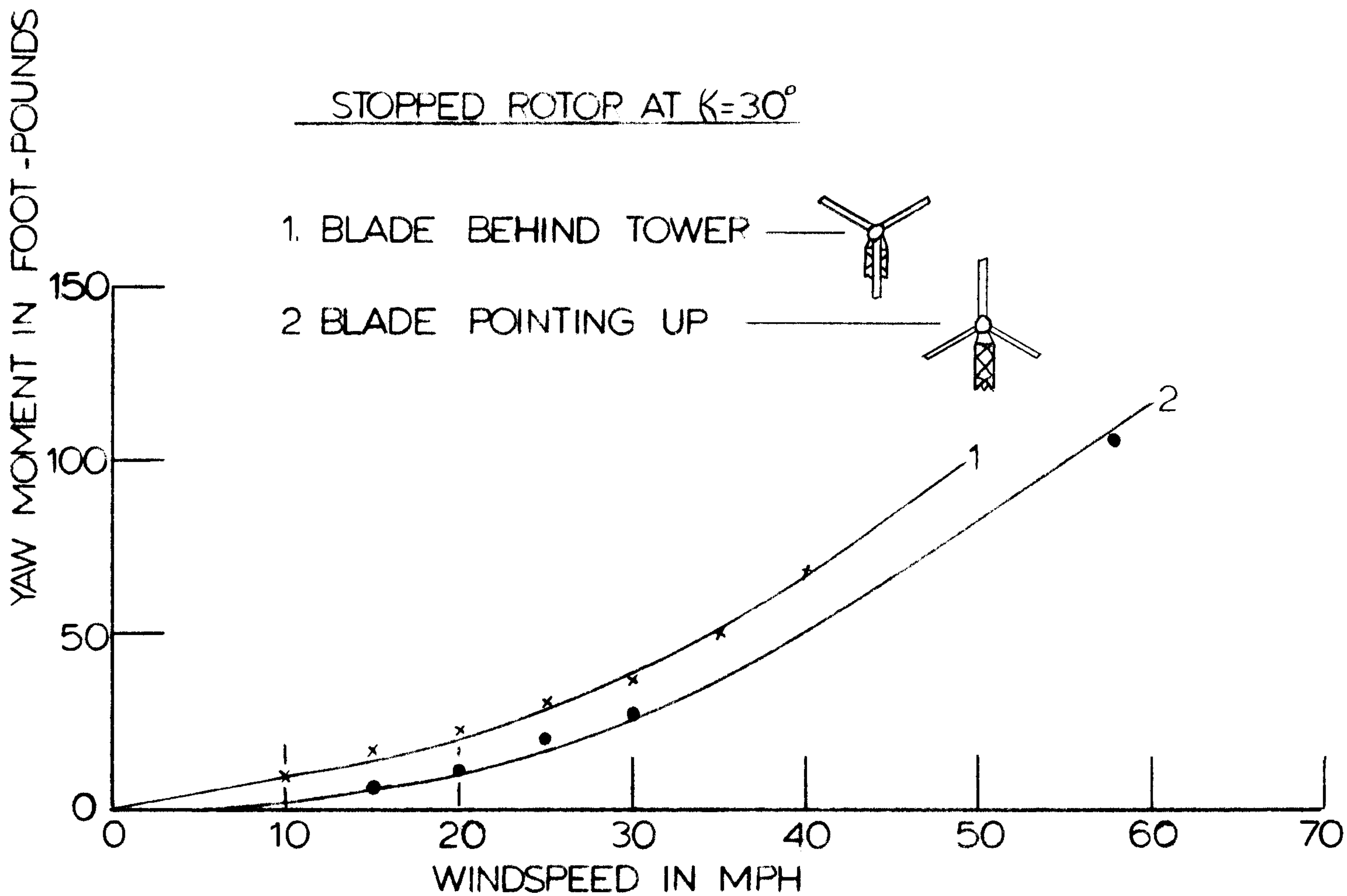


Figure 50

Yaw Moment Test Results - Stopped Rotor

3.5.5 Free Yaw Tests

Following the yaw force tests the E-4000 test model unit was taken to the top of Mt. Washington and set up on the truck (which was guyed down securely) in such a manner that it could yaw freely through 360 degrees. Figures 51 through 54 show the test site and set-up. The temperature ranged from 0°C to 15°C and the winds from 2.2 m/s to 42.5 m/s during the test period, which lasted three days.

Although power output, wind speed and rotor rpm were monitored throughout these tests the purpose of the tests was primarily a qualitative assessment of free yaw behavior under various conditions of wind, bed frame length, and rotor coning angle. Yaw angle with respect to the wind was measured by observing a light streamer attached to the anemometer boom located directly upwind of the machine (see Figures 53 and 54).

The results of these tests were that the machine appeared to track very well, remaining within plus or minus five degrees of downwind at wind speeds from 2.2 m/s to 11.2 m/s at all settings of bed frame length and coning. (Bed frame lengths of 26 in. and 32 in. were tested in combination with coning angles of 0°, 6°, and 12°.) Figure 53 shows typical operation in light winds. As can be seen, alignment with the wind is good.

At higher wind speeds however, the machine consistently operated to the left of the prevailing wind (Figure 54). Yaw in this direction means that the lower blade was advancing. Referring back to the yaw force test data (Figures 47 and 48) it can be seen that the restoring yaw forces are indeed less in this direction than in the other direction. The magnitude of the off-axis yaw appeared to increase with wind speed, ranging from about 10° at 13.4 m/s (30 mph) to perhaps 45° at 35.8 m/s (80 mph). Adjustment of bed frame length had no perceptible effect on yaw behavior with the possible exception that yaw response to sudden changes in wind direction seemed slightly faster with the shortened bed frame length. Coning angle

Figure 51
E-4000 Operating at an Indicated Wind Speed of 85 mph
Mt. Washington, 5/21/80

Figure 52
Mt. Washington Test Site (Instrumentation in Foreground)

Figure 53

Figure 54

did, however, have a noticeable off-axis yaw effect. We estimated that the unit began turning out of the wind at a slightly lower wind speed (9 m/s) and that the angle was perhaps 5° more at higher wind speeds than at 6° coning. Increased coning (12°) has the opposite effect and clearly seemed to improve downwind tracking.

The tendency to turn out of the wind, although not fully understood, does not appear to present any problems for the design of the 15 kW wind machine. This tendency only occurs at higher wind speeds, where it actually serves the desirable purpose of decreasing loads on the rotor. It is probably this effect (combined with the lower air density of the 6200-foot elevation of the Mt. Washington test site) which allowed the E-4000 test model, which was not designed for operation above 26.8 m/s (60 mph), to continue operating without overloading or break-out at wind speeds in excess of 40.2 m/s (90 mph) (see Figure 51).

3.5.6 Tip Brake Tests

Field Testing of Tip Brakes

Early field testing showed that end-mounted tip plates had no detrimental effect on performance (except at wind speeds over 13.4 m/s where the effect was considered desirable--see Figure 41). The original design tip brakes worked well to limit overspeeding, but were found to be subject to premature deployment at normal operating speeds during the 45° yaw force tests. This led to the conclusion that the triggering mechanism for the tip brakes must be completely independent of any aerodynamic forces present on the brakes. Therefore, a new design was developed. This design (see Figures 55 and 56) incorporates a centrifugally operated latch bar which releases the tip brake solely as a function of rotor speed.

Tests showed that the tip brakes reliably deployed at the design rotor speed of 160 rpm (at 140% overspeed) and that any two out of three tip brakes deploying were sufficient to slow the rotor under worst case conditions.

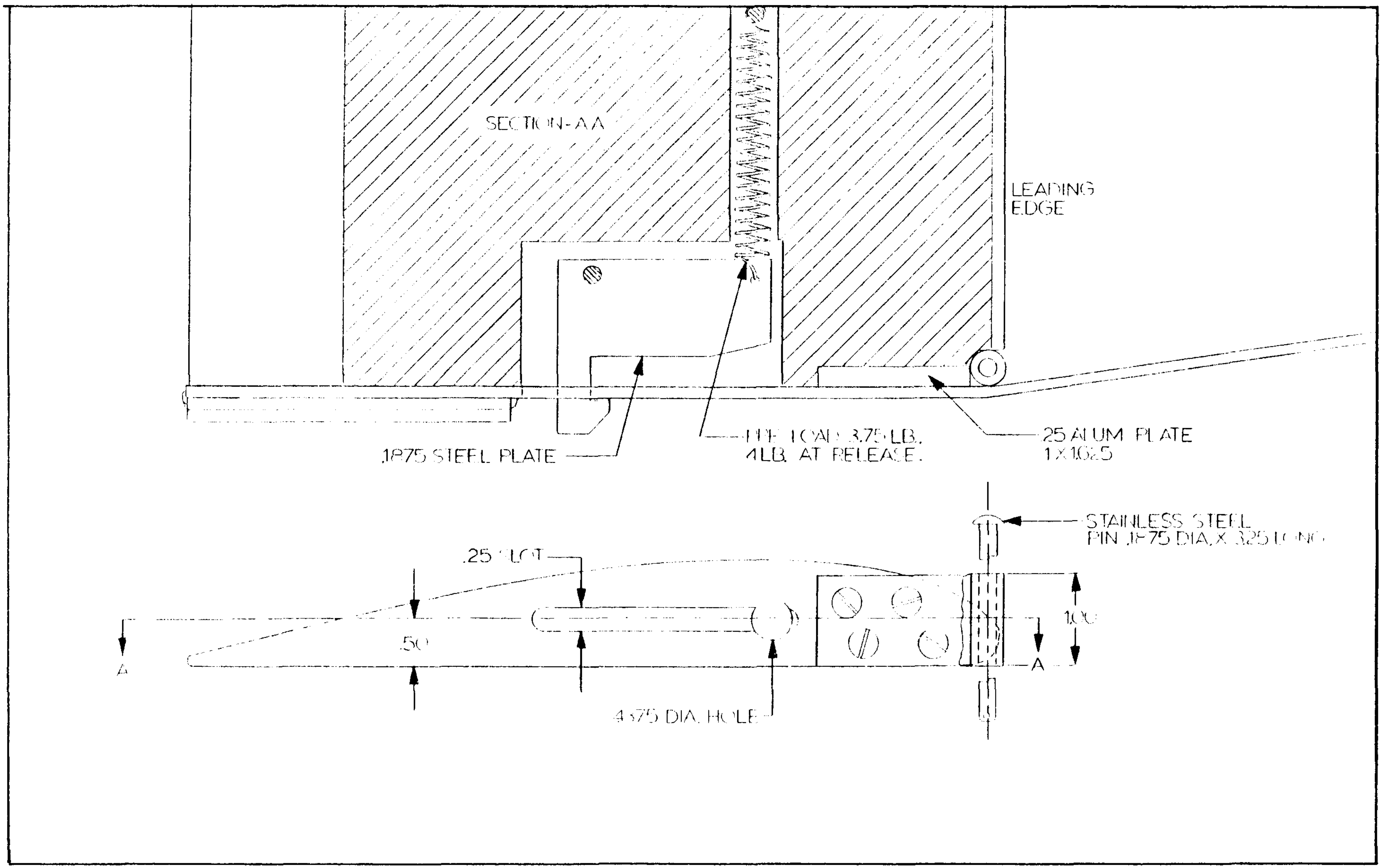


Figure 55

E-4500 Tip Brake Assembly Drawing

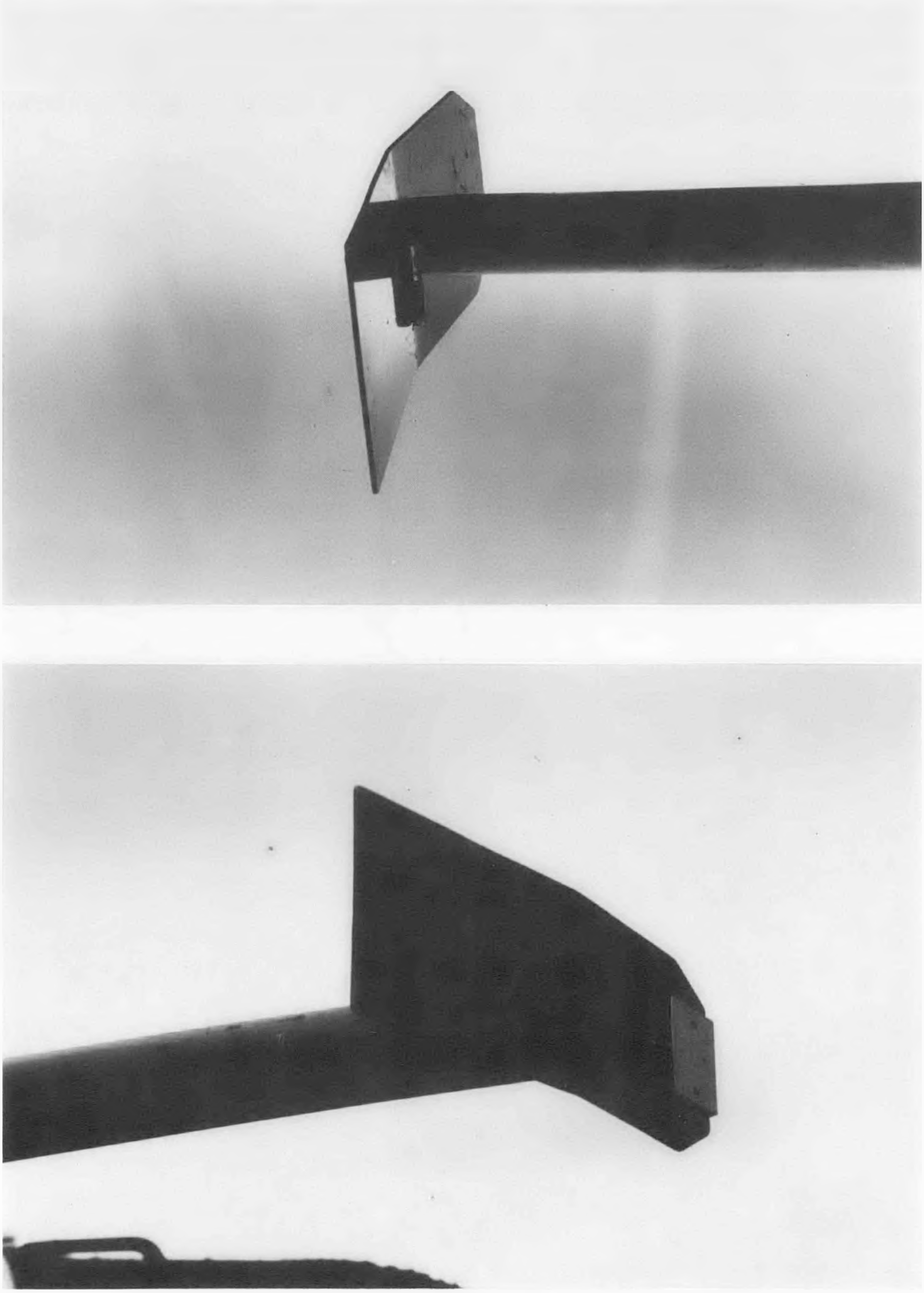


Figure 56

Tip Brakes Used in the Mt. Washington Tests

The tip brakes were tested on Mt. Washington by suddenly unloading the rotor in winds over 31.3 m/s (70 mph). The tip brakes deployed within 2 seconds and the rotor slowed to less than 20 rpm within 15 seconds.

Bench Testing of Tip Brakes Considerable bench testing was conducted on the test model tip brakes to ensure reliable operation. A system was devised whereby the latch spring force, the counterweight and the bend angle of the plate could all be varied independently. This testing greatly increased confidence in the ability to design reliable tip brakes for the 15 kW machine.

1. Counterweight - It was found that adding a counterweight increased the deployment speed, due to increased friction at the latch. A relatively small counterweight will reliably ensure deployment and does not add measureable friction force.

2. Spring Force - Deployment speed increases with spring force as predicted by analysis.

3. Bend Angle - The bend angle is the angle that the end plates are bent inward. Ideally this angle is such that under centrifugal load the plates bend outward, creating a net aerodynamic pressure just prior to deployment that is slightly inward. Testing confirmed that there is a net outward pressure, indicating that the plates need to be set about 4° more inward. Once a certain speed has been reached, (nominally, the speed at which the tips of the plates have positive angle of attack) then they will not deploy since they are held out by the lift of the airstream. This can be seen from Figure 57. At the 12° bend the brakes deploy reliably at 2600 rpm but with a 10° bend they would not deploy.

4. Performance - Performance tests with the end plates at various bends consistently showed a slight improvement in performance over blades with no end plates.

5. Backup System - The aluminum plates are designed to yield and bend outward should they fail to unlatch. This system was tested twice and found to work reliably. The calculated speed of yielding was within 50 rpm of the test speed. It should be noted that backup system deployment is at a higher speed and may be harmful to the rotor.

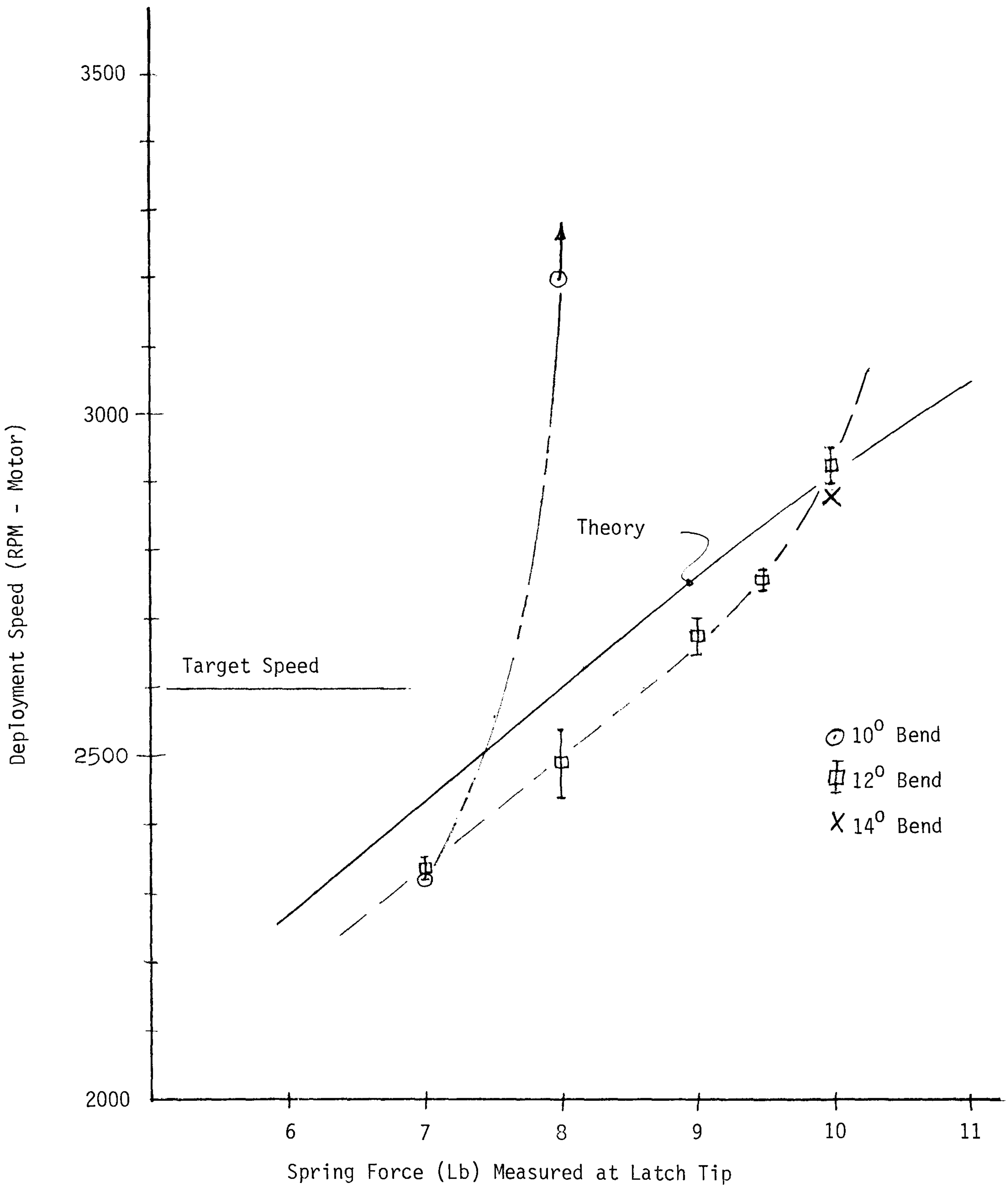


Figure 57
Tip Brake Spring Force Measured at Latch Tip

3.5.7 Cold Temperature Testing of Model Hydraulic Brake

One concern which arose during the changeover from a mechanical to a hydraulic main brake for the 15 kW prototype was the possible adverse effect of cold temperatures on the hydraulic system. Since the hydraulic brake manufacturer was unable to supply any information on operation of their unit below -17.8°C (0°F), a simple experiment was set up on the test model unit in order to obtain experimental data on operation at very low temperatures. (The "standard" design calls for operation down to -30°C, while the "special" design requires operation down to -50°C.)

The experiment consisted of placing a complete Enertech 1800 powertrain (gearbox, motor/generator, and hydraulic brake assembly) inside an insulated wooden box which was packed with dry ice (-68°C). A capillary-type remote-reading thermometer was installed with its sensor located in the hydraulic fluid reservoir and the read-out was placed outside the box. A 115-Vac power cord was run into the box so that the unit could be motored at normal operating speed, and provisions were made for monitoring the current draw to the motor. The test set-up is shown in Figure 58.

Once packed with dry ice, the top of the box was closed and sealed and the temperature of the hydraulic oil was monitored. At various intervals, as the unit cooled down, the motor and brake were energized and the power consumption after a period of 10 seconds was noted. The data collected in this manner is shown in Table XII. It should be noted that the experiment took place in two parts. First the entire powertrain assembly was cooled evenly down to approximately -7°C. Then, to further cool the brake assembly, the box was re-packed with the remaining dry ice concentrated around the hydraulic brake unit.

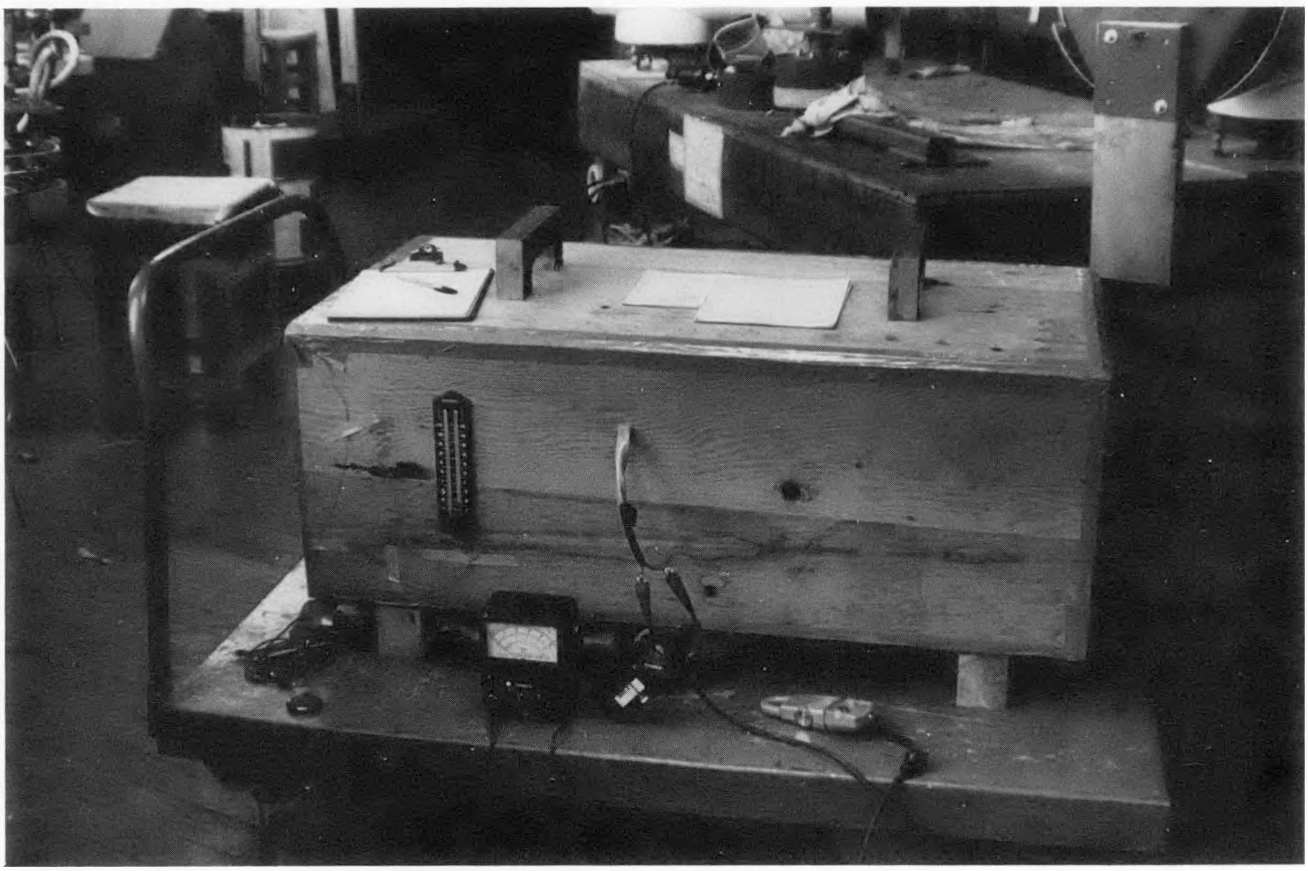


Figure 58
Cold Temperature Test Set-Up - Model Hydraulic Brake
(Instrumentation includes a remote-reading thermometer and ac ammeter.)

Table XII - Current Draw as a Function of Temperature for E-1800 Cold-Test Unit

Temperature of Hydraulic Oil °F	°C	Running Current of motor (incl. brake) after 10 seconds
22°	- 7°	8.5 Amps
12	-12	10
- 3	-19.5	13
-20	-28	17
-30	-34.5	16
-46	-44	15
-52	-47	14

Figure 59 shows the thermometer readings at the two coldest temperatures obtained just after the 10-second test runs (which warmed the hydraulic fluid some 3 to 4 degrees). Figure 60 shows the assembly after the test when the box was opened up. As can be seen, the brake assembly is well frosted up.

This test showed that the hydraulic brake assembly did continue to function without imposing excess drag on the unit at temperatures down to -47°C (-52°F). The major source of drag appears to have been the gearbox as the total drag actually decreased when the brake was cooled seperately and the gearbox allowed to warm up. This explains the slightly lower current draw obtained at the coldest temperatures monitored.

The hydraulic fluid used in this test was Conoco DN-600 gear oil, the same lubricant used in the gearbox. The use of the same fluid in both brake and gearbox simplifies maintenance procedures and decreases the chance for errors. As a result of this test it was decided to use this same approach for the 15 kW machine.

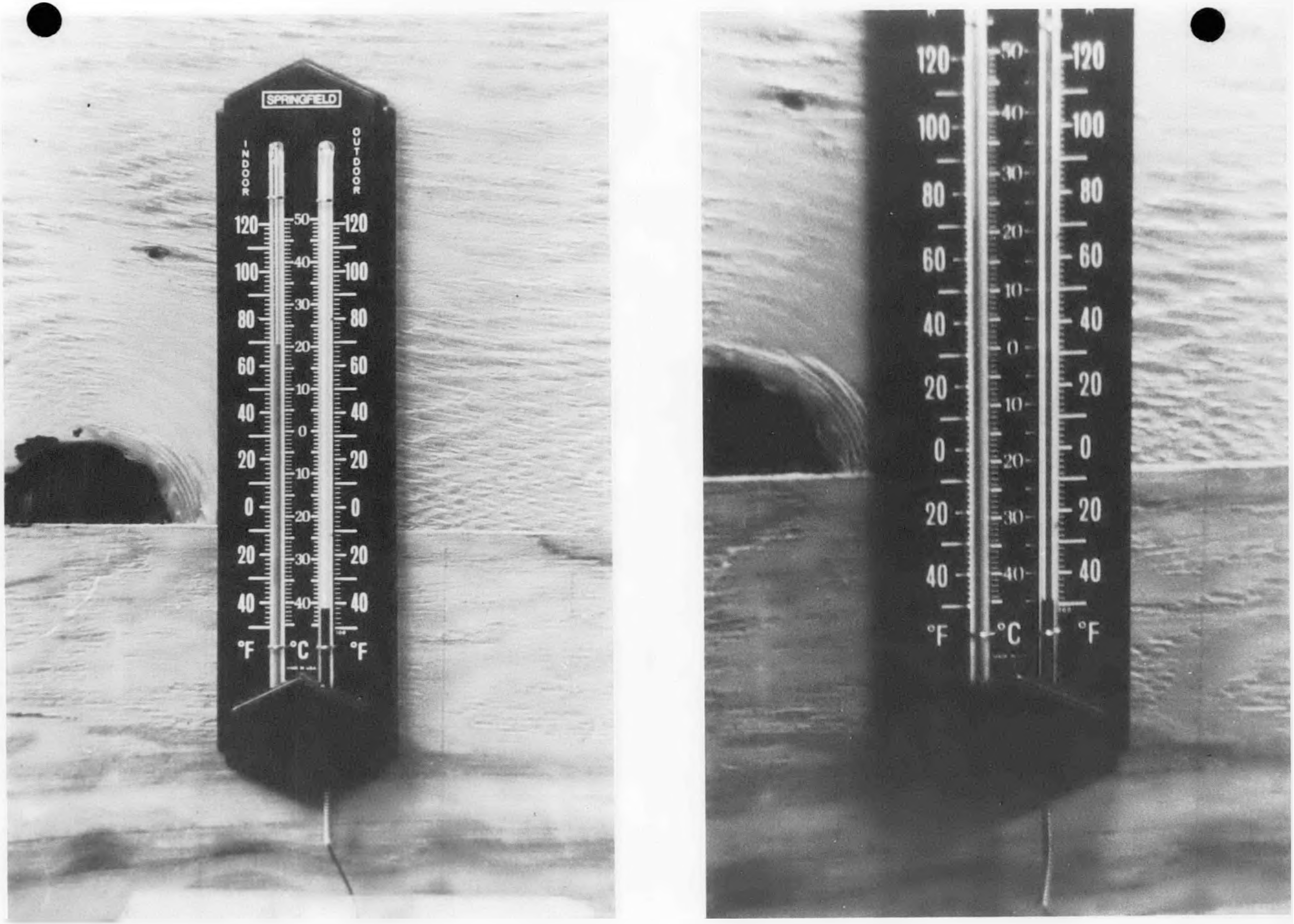
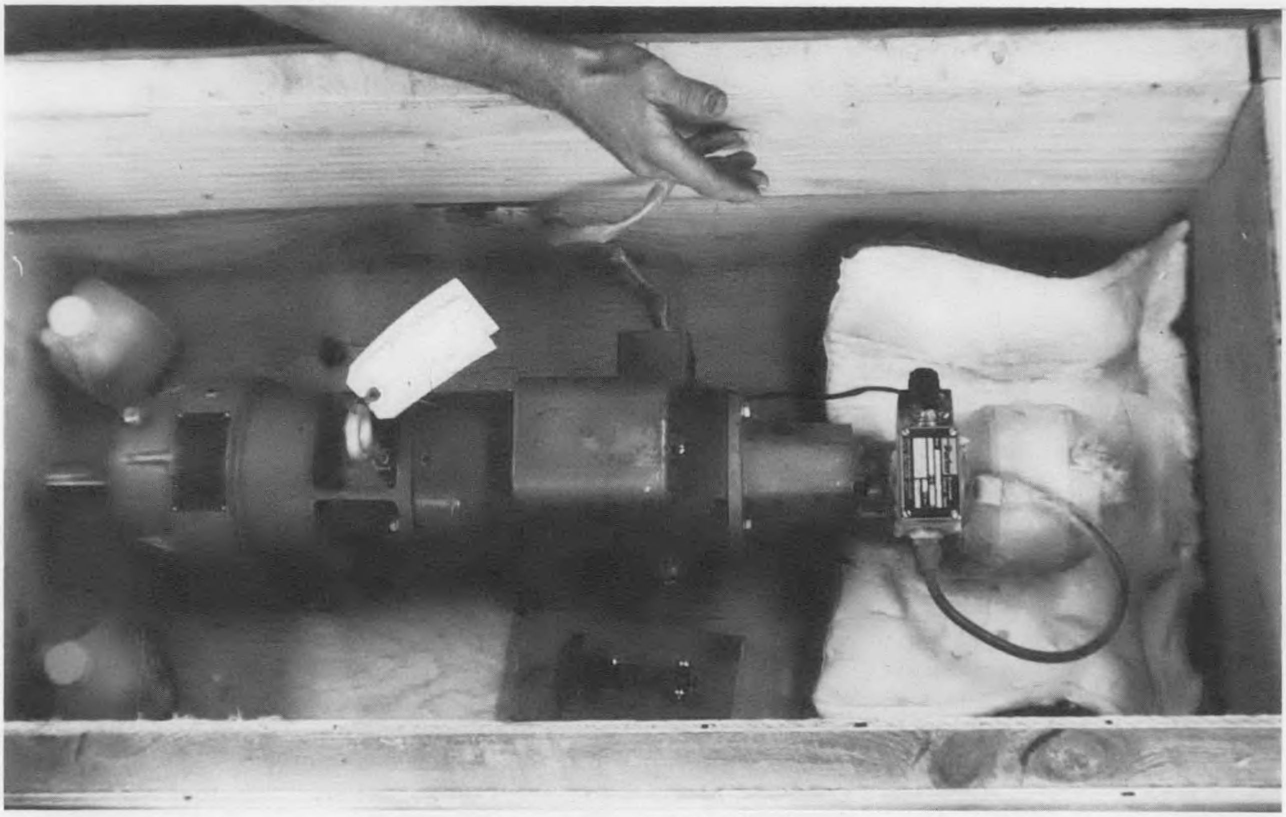


Figure 59
Hydraulic Oil Temperatures After Two Cold Temperature Tests



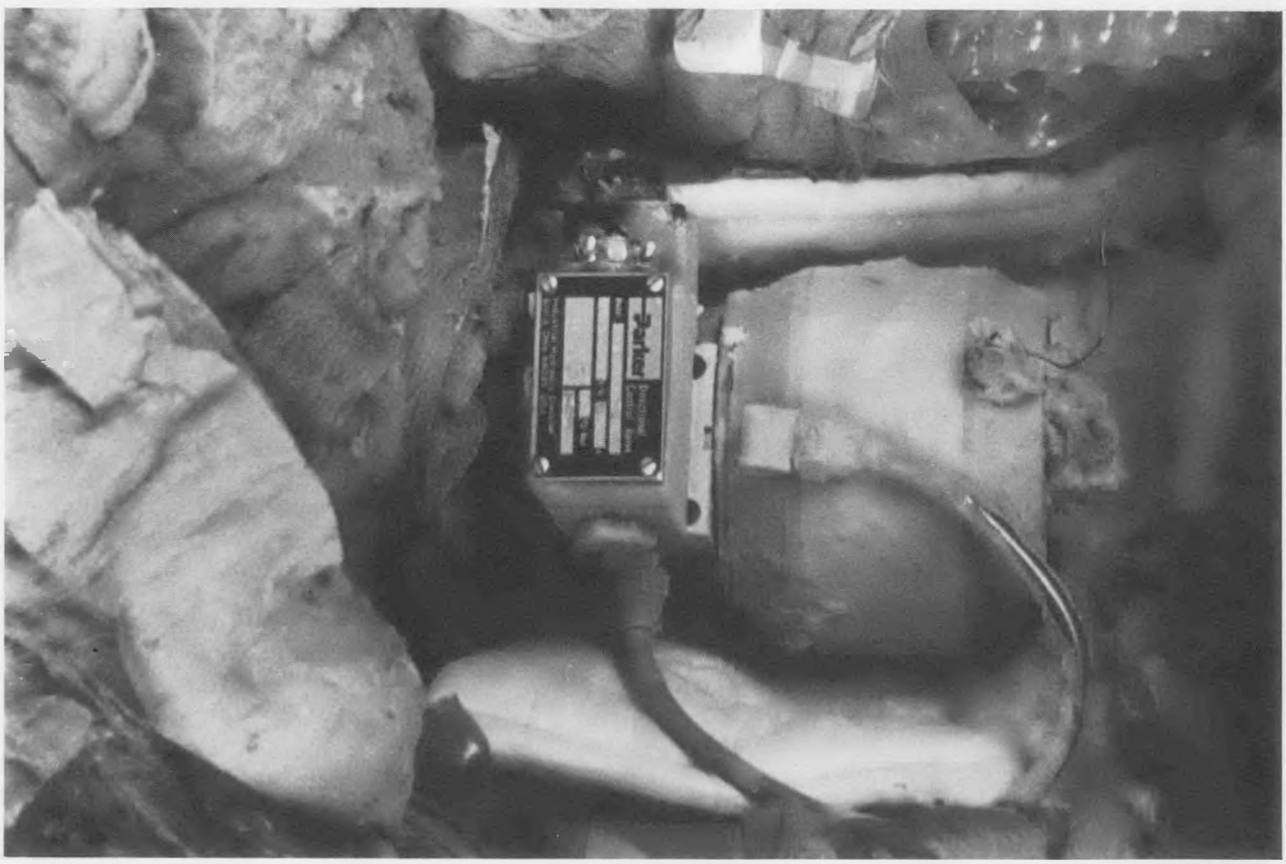


Figure 60

E-1800 Power Train After Cold Testing

3.6 15 kW Induction Generator Testing

3.6.1 Description of Operation

An induction generator test facility was constructed and successfully operated for testing 15 kW generators. Figure 61 shows the physical layout of the facility. An internal combustion engine and transmission provide the power (at the proper rpm) to the generator. This mechanical power is calculated by knowing rpm of the drive shaft (from the tachometer) and torque (from the cradle/spring scale arrangement). Electrical parameters such as voltage, current, and power are sensed in the contactor box and displayed on the monitoring panel (see Figure 62).

A generator is tested by first starting the engine, then adjusting the throttle to approximately 1800 rpm, switching on the contactor, and finally adjusting the throttle to obtain desired power outputs and taking data. The setup works quite well, without excess noise or heat.

3.6.2 Baldor Generators

The first three-phase Baldor generator tested did not perform up to specifications, with a maximum power factor of only .8. That generator was subsequently rewound for single-phase operation by T & L Electric. The results of the initial rewind were encouraging. Efficiencies hovered around 80 percent and the power factor was near .9. However, a "break-out" condition was reached at only 18 kW output. The motor was rewound a second time to improve the breakout point to something over 20 kW. Results improved dramatically with efficiency over 85% in the reversed mode condition and breakout at 26 kW (see Table XIII).

The generator was able to produce 13.5 ft-lb of torque (115 volts, 85 amps) under locked rotor conditions. This is more than adequate for this application.

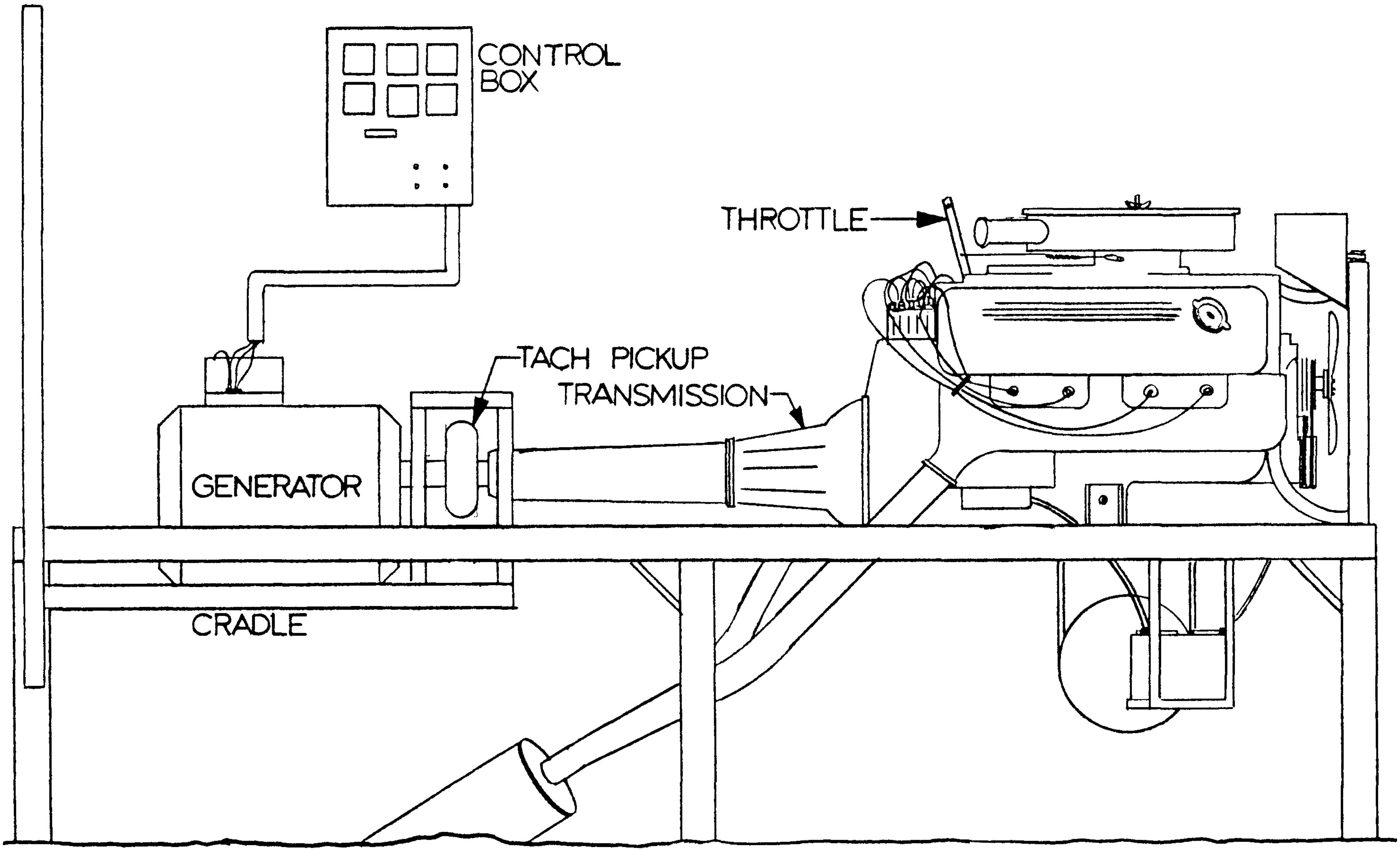


Figure 61
Induction Generator Testing Facility

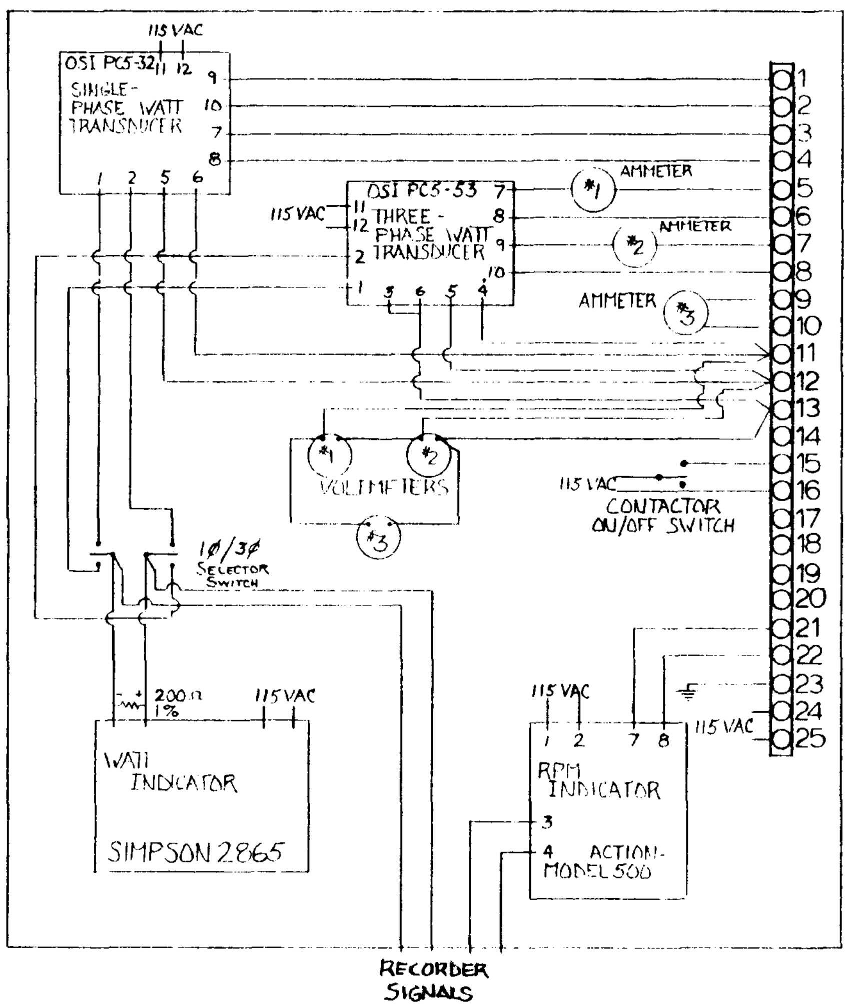


Figure 62

Generator Test Facility Meter Box Wiring Diagram

Table XIII

Baldor 10 Generator Performance Data

Tests Performed by Enertech, October 1980

Power Output (Kilowatts)	Torque (ft-lb)	RPM	Power Input (Kilowatts)	Volts	Amps	Power Factor	Efficiency
2	12	1805	3.1	245	10	.81	.65
4	22.5	1810	5.8	246	20	.86	.69
8	36	1819	9.3	247	35	.93	.86
12	54	1829	14.0	248	52	.93	.86
16	75	1842	19.6	249	73	.88	.82
20	100	1861	26.6	250	98	.82	.75

The effect of "reversing" the motor leads for generation purposes (as opposed to motoring) had the effect of improving system efficiency. An efficiency improvement of approximately seven percent was noted on this machine.

3.6.3 Gould Generator

The Gould generator performance is impressive (see Tables XIV - XVI). It closely approximates the performance predicted at the Critical Design Review.

Of particular interest are the results of the heat run tests. They indicate the three-phase and single-phase generators can be rated at 20 kW and 18 kW output continuous operation. This is better than expected, indicating a somewhat conservative design.

The use of 100 micro farad run capacitance rather than 80 improves high end performance at the expense of low output performance. Since a majority of the machine's energy will be produced at lower windspeeds, the 80 micro farad value is preferable.

The generator design is more than adequate; however, Enertech testing of the Gould generator showed results to be somewhat lower than Gould predicted. This may have been due to the higher testing voltage (250 V) versus the 230 V used by Gould (see Table XVI).

3.6.4 Conclusions

Reversing winding connections significantly improves generator performance. Both the Baldor and Gould generators perform adequately for use on the prototype machine. Particularly promising are their high breakout level (26 kW) and high start up torque (over 10 ft-lb). Additional testing of the Gould three-Phase generator is planned for Phase II.

Table XIV

Gould 1Ø Generator Performance Data

Tests performed by Gould, July 1980

Power Output (Kilowatts)	Torque (ft-lb)	RPM	Power Input (Kilowatts)	Volts	Amps	Power Factor	Efficiency
3.5	16.4	1810	4216	230	15.4	.99	.83
5.0	23.0	1813	5923	230	22.0	.99	.84
10.0	44.2	1826	11464	230	44.5	.98	.87
20.0	99.8	1861	26381	230	101	.86	.76
22.4	119	1881	31795	230	124	.79	.70

Notes: 80 micro farad run capacitance
wired for opposite direction of rotation
data taken after heat run

Table XV
Gould 1Ø Generator Performance Data
Tests performed by Gould, July 1980

Power Output (Kilowatts)	Torque (ft-lb)	RPM	Power Input (Kilowatts)	Volts	Amps	Power Factor	Efficiency
3.5	20.8	1809	5.34	230	27.5	.55	.65
5	26.6	1812	6.84	230	31.0	.65	.73
10	45.8	1823	11.9	230	44.8	.99	.84
15	66.2	1833	17.3	230	66.0	1.0	.86
20	89.0	1850	23.3	230	88.8	.98	.85
25.6	125	1879	33.3	230	130	.86	.77

Notes: 100 micro farad run capacitance
wired for opposite direction of rotation
data taken after heat run
heat run indicated 18 kW continuous rating (Class B insulation)

Table XVI

Gould 1Ø Generator Performance Data

Tests performed by Enertech, July 1980

Power Output (Kilowatts)	Torque (ft-lb)	RPM	Power Input (Kilowatts)	Volts	Amps	Power Factor	Efficiency
2	15	1802	3.8	246	10	.81	.53
4	24	1805	6.2	246	18	.90	.65
8	39	1811	10.0	248	31	1.0	.80
12	57	1819	14.7	250	48	1.0	.82
16	78	1828	20.3	251	68	.94	.79
20	100	1839	26.3	253	89	.89	.76

4.0 DETAIL TOWER DESIGN

4.1 Tower Configuration

The baseline design of the tower is a 3-legged steel self-supporting tower. Because dynamic analysis of sections 6N, 7N, 8N of the Unarco-Rohn SSV series indicated probable dynamic interactions, sections VG1S, 6N, and 7N will be recommended. These sections have been strengthened from the standard Rohn towers. However, as all production jigs and fixtures can be used in manufacturing this design, this is an "off-the-shelf" unit. The tower is made up of three self-supporting sections as shown in Figure 63. The top section is straight with 2.5 feet between legs; the bottom two sections taper to a base leg distance of 6 ft 6.75 in. The legs are made of standard pipe sections and the cross bracing is made of structural angles.

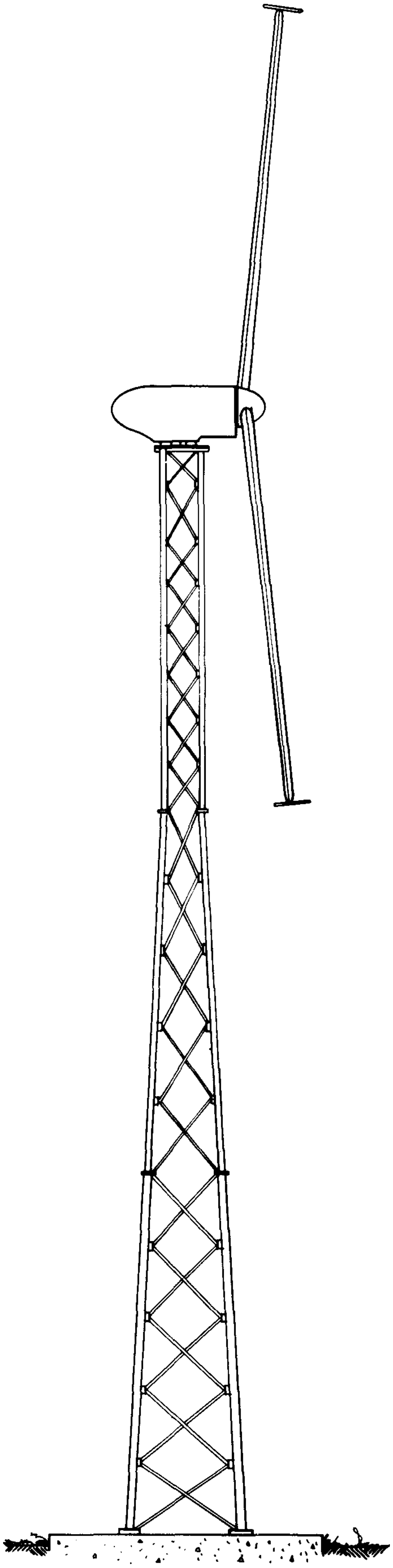
The approximate weight of the standard tower is 3300 pounds. The weight of the tower for the special design is 4518 pounds.

The tower has been designed to safe-life design criteria. Further, the design of the legs and of the angles conforms to the steel construction codes. Dynamic analysis of the structure and operating conditions indicates that interactions of the tower and machine will not occur.

4.2 Tower Structural Analysis

4.2.1 Description of Analysis

The Unarco-Rohn tower sections were analyzed for structural integrity. Tower sections were modeled as pin-jointed truss sections. A computer program was designed to analyze the force in each member. From information on these stresses, the necessary leg sections and cross bracing sections were specified.



Standard

Sections	Legs	Cross Bracing (∠)
VG1S	4" STD Pipe	1½ x 1½ x 3/16
6N	5" STD Pipe	1½ x 1½ x 1/8
7N	5" STD Pipe	1½ x 1½ x 1/8
APPROXIMATE WEIGHT - 3,300 POUNDS		

Special Tower

Sections	Legs	Cross Bracing (∠)
VG1S	6" STD Pipe	1 3/4x 1 3/4 x 3/16
7N	6" STD Pipe	1 3/4 x 1 3/4 x 1/8
8N	6" STD Pipe	1 3/4 x 1 3/4 x 1/8

Figure 63

Final Tower Configuration

The tower was analyzed for both the maximum load condition and for the fatigue condition. For the steel tower, the maximum load condition determined the design. The tower loads for the maximum condition are shown in Table XVII. To determine the wind loads on the tower, the following equation was used:

$$F = \frac{1}{2} \ (\rho) \ (V^2) \ (CD) \ (AS)$$

where F = drag force on tower
ρ = air density
V = wind velocity
CD = drag coefficient
AS = projected area of one face of tower

From tower solidity and Reynolds numbers, the drag coefficient was calculated (Ref 17).

The equations to determine the allowable stress in each member were as specified by the American Institute of Steel Construction (Ref 18 and Ref 19). Table XVII describes the allowable stress in compression for the members. For both tower legs and tower cross bracing, compression stresses drove the design.

4.2.2 Conclusions

The minimum safety factors (SF) for the primary members are as follows:

	SF AISC Code	SF Above Code
VG1S	1.67	1.17
6N	1.67	1.00
7N	1.67	1.25

The sections were specified to minimize the SF above the code to 1. The tower was designed both to safe-life design and to AISC codes.

Table XVII

Tower Loads

I. Maximum Load Condition

A. Moment at Tower Top

Aerodynamic Moment	18900 foot pounds
Weight Moment	8533
Thrust Moment	15800
Ice on Blades	3953
Total Moment	47186 foot pounds

B. Weight Force

Weight of Machine	2870 pounds
Ice on Machine	1309
Ice on Tower	2021
Tower Top Plate	100
Total Vertical Force	6300 pounds

C. Thrust Forces

Thrust on Wind Machine	9900 pounds
Wind Force on Tower	5040

Table XVIII
Equations Used in Tower Design

Summary: All members were designed to withstand compression stresses.

A. Primary members (the pipe leg sections)

(1) $$Fa = \frac{\left[1 - \frac{(K\ell/r)^2}{2C^2}\right] Fy}{\frac{5}{3} + \frac{3(K\ell/r)}{8C} - \frac{(K\ell/r)^3}{8C^3}}$$

where Fa = allowable compression stress

K = length factor

ℓ = length of member

r = radius of gyration

Fy = yield strength of material

C = slenderness ratio dividing elastic from inelastic buckling where

$$C = \sqrt{\frac{2\pi^2 E}{Fy}}$$

E = modulus of elasticity

(2) $$Fa = \frac{12\pi^2 E}{23(K\ell/r)^2} \quad \text{for} \quad \frac{K\ell}{r} > C$$

B. Secondary members (as the cross bracing)

$$Fa = \frac{Fa}{1.6 - \frac{\ell}{200r}}$$

4.3 Tower/Wind Machine Dynamics

The wind machine/tower system was modeled as a lumped-mass system and the first and second tower bending frequencies were calculated. The analysis assumes the following:

1. Tower deflections result from bending moments. Shear deflections are neglected.
2. The bending moment is taken up by the tower legs. The moment of inertia of the tower sections is the moment of inertia of the tower legs about the center of the tower.
3. Static compression loads in the tower legs are neglected. Compression loads are well below the buckling loads and will have little effect on frequency.

While the moment of inertia of a three-legged tower is the same in all directions, the moment of inertia of the wind machine is not. Because the center of mass of the wind machine does not coincide with the yaw axis, transverse motion of the wind machine and tower is accompanied by yawing of the machine. Therefore, first and second tower bending frequencies for both the axial and transverse directions were calculated.

The results of the analysis of tower natural frequencies are as follows:

Direction of Motion	First Bending Frequency	Second Bending Frequency
Along rotor axis	1.89 Hz	11.4 Hz
Transverse to rotor axis	2.15 Hz	11.0 Hz

The wind machine/tower system natural frequencies are shown in Figure 64. The expected rotor excitation frequencies occurring in the operating range (50.5 to 52.8 rpm) are:

1/REV = .842 - .88 Hertz
3/REV = 2.52 - 2.64 Hertz

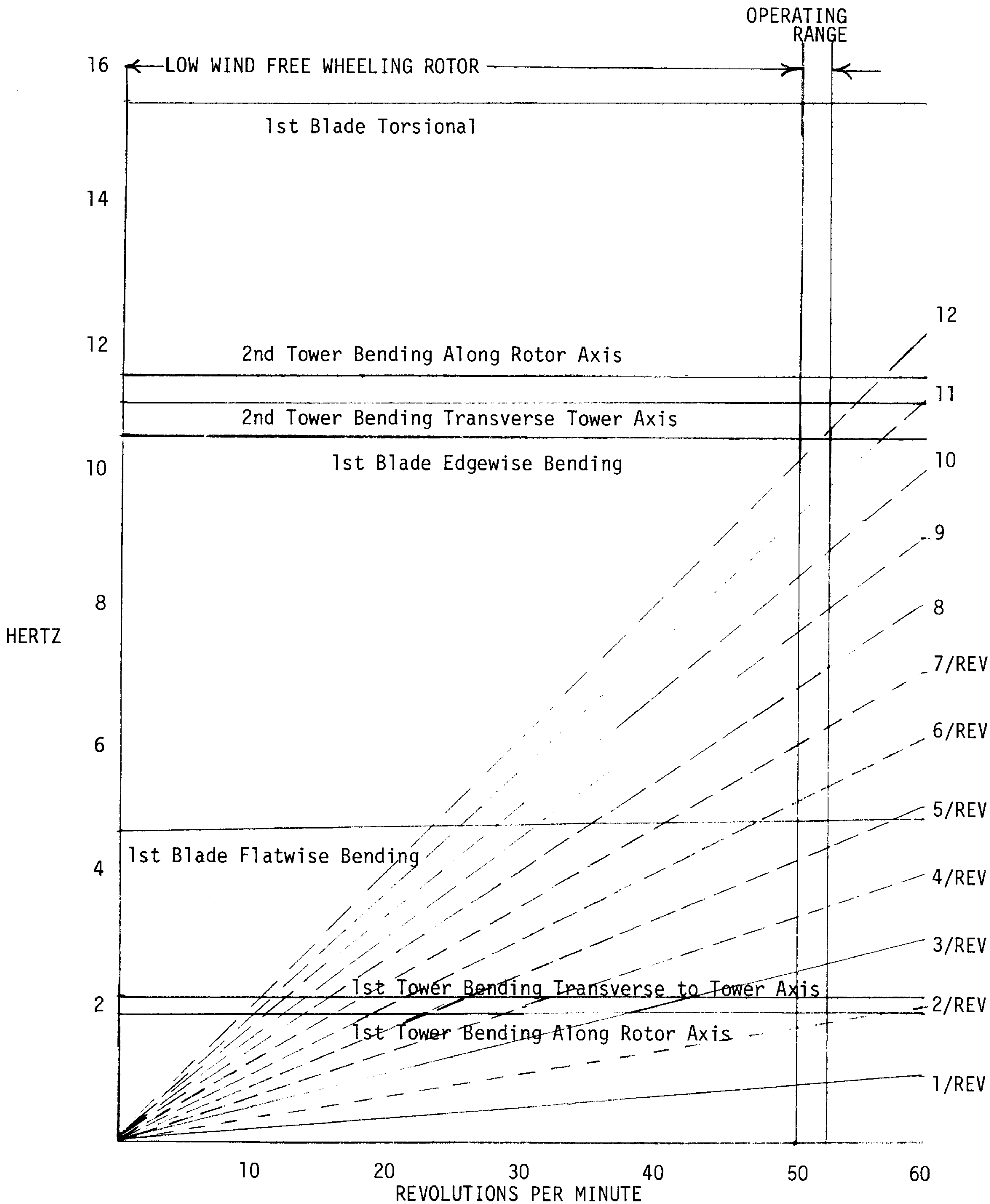


Figure 64

Wind Machine Natural Frequencies

In Figure 64 it can be seen that the two second tower bending frequencies are significantly higher than the excitation frequencies. The first tower bending frequency in the axial direction is about 34% lower than the 3/REV excitation frequency and is not expected to cause any vibrational problems. The first tower bending in the transverse direction is about 18% lower than the 3/REV excitation frequency. If significant 3/REV excitation occurs in the transverse direction, this could cause a vibrational problem. Because the magnitude of this excitation is difficult to predict, the test unit should be observed for possible excitation in this direction.

The first three flatwise blade bending frequencies are as follows:

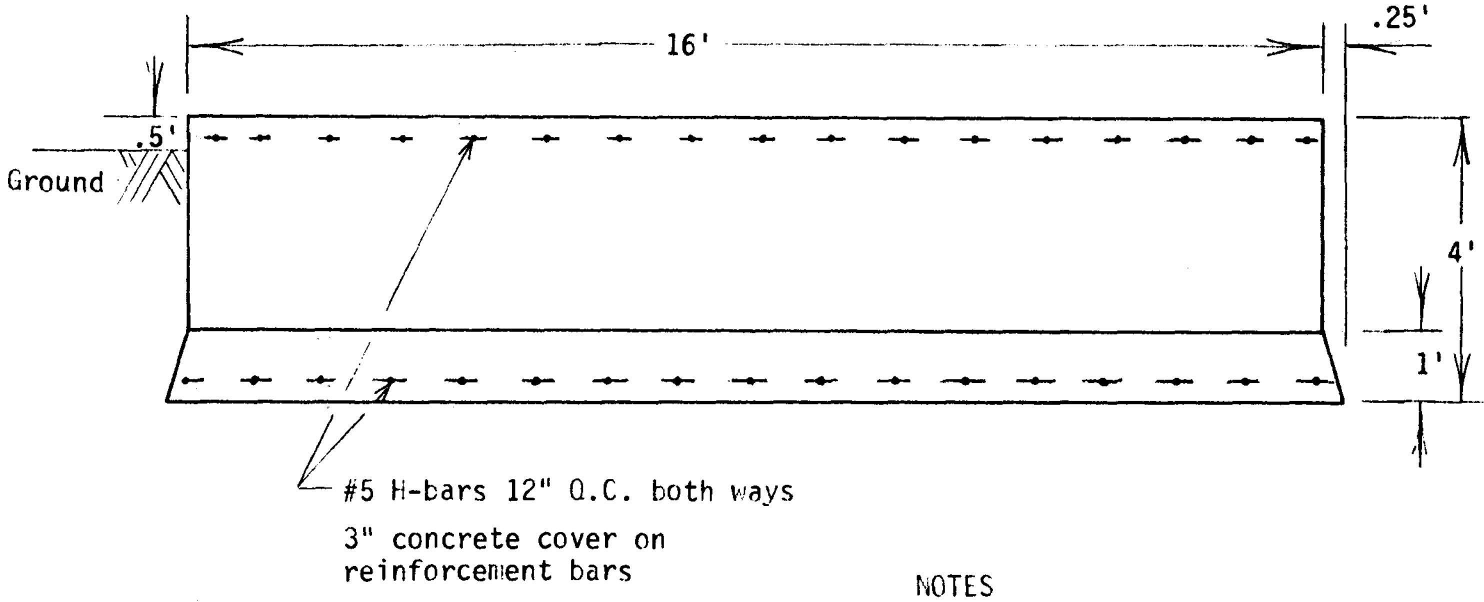
Mode	Natural Frequency (rotor stopped)	Natural Frequency (rotor spinning)
1	4.6 Hz	4.8 Hz
2	16.1	16.3
3	37.5	37.7

The first edgewise frequency of the blade is about 10.5 Hz; the first torsional blade frequency is about 15.5 Hz. Because the blade natural frequencies are well above the rotor excitation frequencies, blade vibration is not expected to be a problem. Although the first blade edgewise frequency is in the range of the second tower bending frequencies, tower/blade interactions are not expected, as excitations of this higher frequency are unlikely.

4.4 Tower Foundation

4.4.1 Description

The recommended foundation for the baseline tower design, the off-the-shelf Rohn tower, is a 16 x 16 x 4-foot deep reinforced concrete mat or slab as shown in Figure 65.



NOTES

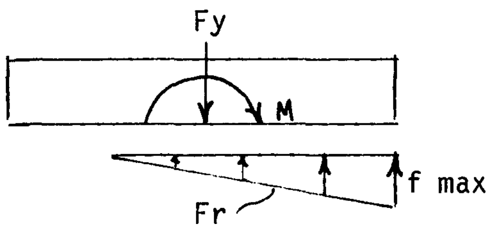
1. 38.2 cubic yards of 3000 psi concrete at 28 days
2. 40,000 psi minimum deformed H-bars
3. Designed for E.I.A. normal soil

Figure 65

Mat-Type Tower Foundation for 60-foot Self Supporting Tower

The foundation was analyzed as a beam structure and designed to comply to standards of the American Concrete Institute. The base of the slab is located 3.5 feet below the ground surface to prevent frost upheavals. Because the foundation must resist an overturning moment by a factor of safety of 1.5, the foundation is sized to provide the necessary resisting weight moment against overturn (as per Ref 20). Deformed reinforcement bars positioned perpendicular to the side faces and located 3-in from both the top and bottom of the slabs resist the tensile stress caused by the internal bending moment in the slab.

The soil is assumed to be an elastic structure and the foundation is designed so the soil can resist both an overturning moment and a vertical weight force. For the analysis, the following free body force diagram is assumed:



where: M = sum of the moments resulting from thrust forces on the tower and wind machine.

Fy = vertical force due to the weight of the wind machine, tower, foundation, and ice load.

Fr = resultant force prism along the base of the foundation.

f max = maximum distributed bearing force per unit lengths on the soil.

The maximum bearing pressure is less than the allowable bearing pressure for a "normal soil" as classified by the Electrical Industry Association in its specifications for towers and foundations for antennas. A normal soil is defined as "a cohesive type soil with an allowable net vertical bearing capacity of 4000 pounds per square foot and an allowable net horizontal pressure of 400 pounds pr square foot per linear foot of depth. . . . Rock non-cohesive soils, or saturated or submerged soils should not be considered normal," (Ref 21).

The foundation options comply with the concrete design specifications and the normal soil condition.

4.4.2 Foundation Options

In the design of the mat foundation, the weight of the concrete structure provides the resistance to overturning loads. Where space and soil conditions permit, three deep foundations (one for each leg) would use both the concrete and the soil to resist overturning. Figure 66 shows two options for deep foundation designs that could be used for a wide-base tower such as that envisioned for a production design (see Section 6.7). The base of the pier and pad foundation is assumed to engage the frustum of an inverted pyramid with sides 30° from the vertical; the base of the drill and bell foundation engages the frustum of an inverted cone.

Deep use of foundations for the narrow-base prototype tower would not be realistic for two reasons. First, the overturning moment is resisted by force couples that are proporational to the tower base width, as follows:

$$Fc = \frac{2}{3} \times \frac{M}{D}$$

where: Fc = maximum resulting force either in uplift or bearing (the force in the other legs is 1/2 of this value).

M = tower base moment resulting from thrust.

D = leg to leg distance.

Therefore, a production design tower (see Section 6.7) with a leg-to-leg distance of ten feet would have less maximum leg force at the base than the prototype tower with a 6 1/2-foot base. Second, in the case of the narrow base prototype design there is not enough space for three separate pads that could resist the bearing loads.

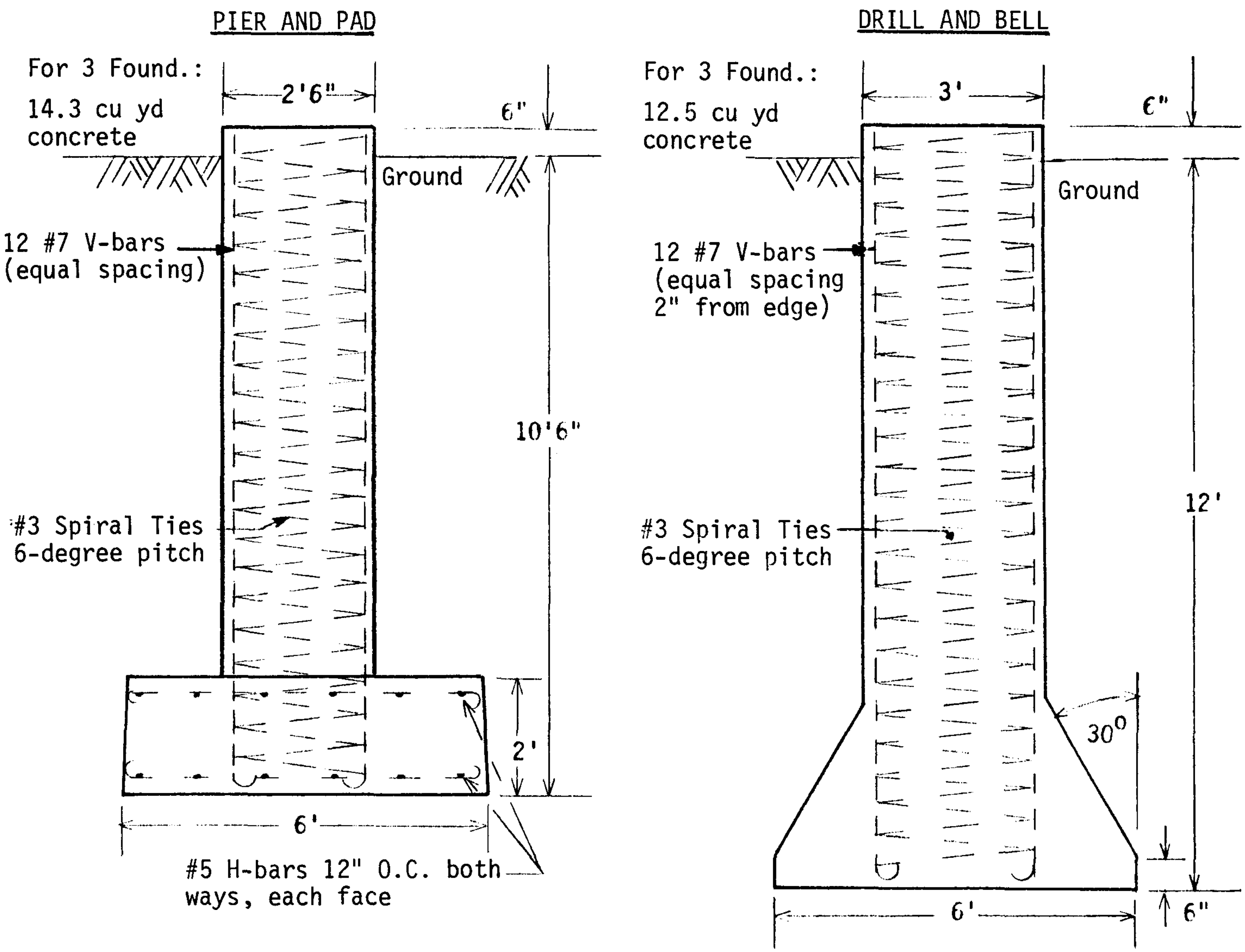


Figure 66

Foundation Options for a Wide-Base Tower

For the production unit design, three foundation options would be available. While the mat foundation represents significantly more concrete than the deeper foundations, the deep foundations would require special equipment and/or large forms. Because depth of normal soil, availability of personnel, cost of concrete and labor, and availability of equipment vary from wind site to wind site, there will probably be cost-effective applications for each design.

5.0 DETAIL INSTALLATION EQUIPMENT DESIGN

5.1 Basic Installation Plan

The contract design criteria specify that the installation be completed using only basic tools and not require the use of a crane.

Accordingly, a gin-pole erection plan was developed for the 15 kW unit. Under this plan, a minimum of three people can install the unit using hand winches, snatch blocks, and two light trucks. Two hand winches are specified for the procedure. An electric winch may be substituted for the main winch. Two snatch blocks are specified, to be used on truck bumbers to guide the tag-lines positively. This is especially important as the base of the specified tower is much broader than the top. Snatch-block locations can be controlled discretely by moving the trucks, which themselves are stable mounting points.

Equipment:

1. Two (2) Beebe W 200-S hand winches - capacity 3000 lb weight; drum capacity 253 ft.; with 3/16" 7 x 19 aircraft galvanized wire rope (allowable load 2000 lb).

2. Two (2) snatch blocks for hand-controlled tag-lines running around bumpers of stationary trucks; four 150-foot tag line ropes.

3. 12-foot aluminum gin pole

4. Work platform

5. Safety belts, hard hats, basic tools, percussion wrenches.

5.2 Detailed Installation Procedure

1. One person climbs the tower with a carabiner and rope attached to the safety belt. A snatch block is attached to the rope and is raised and attached to a tower leg, near the top. The rope is used with the snatch block to raise the two sections of the erection fixture, which are attached to a tower leg, one just below the leg flange at the top section, and the other 7 feet below (see Figure 67).

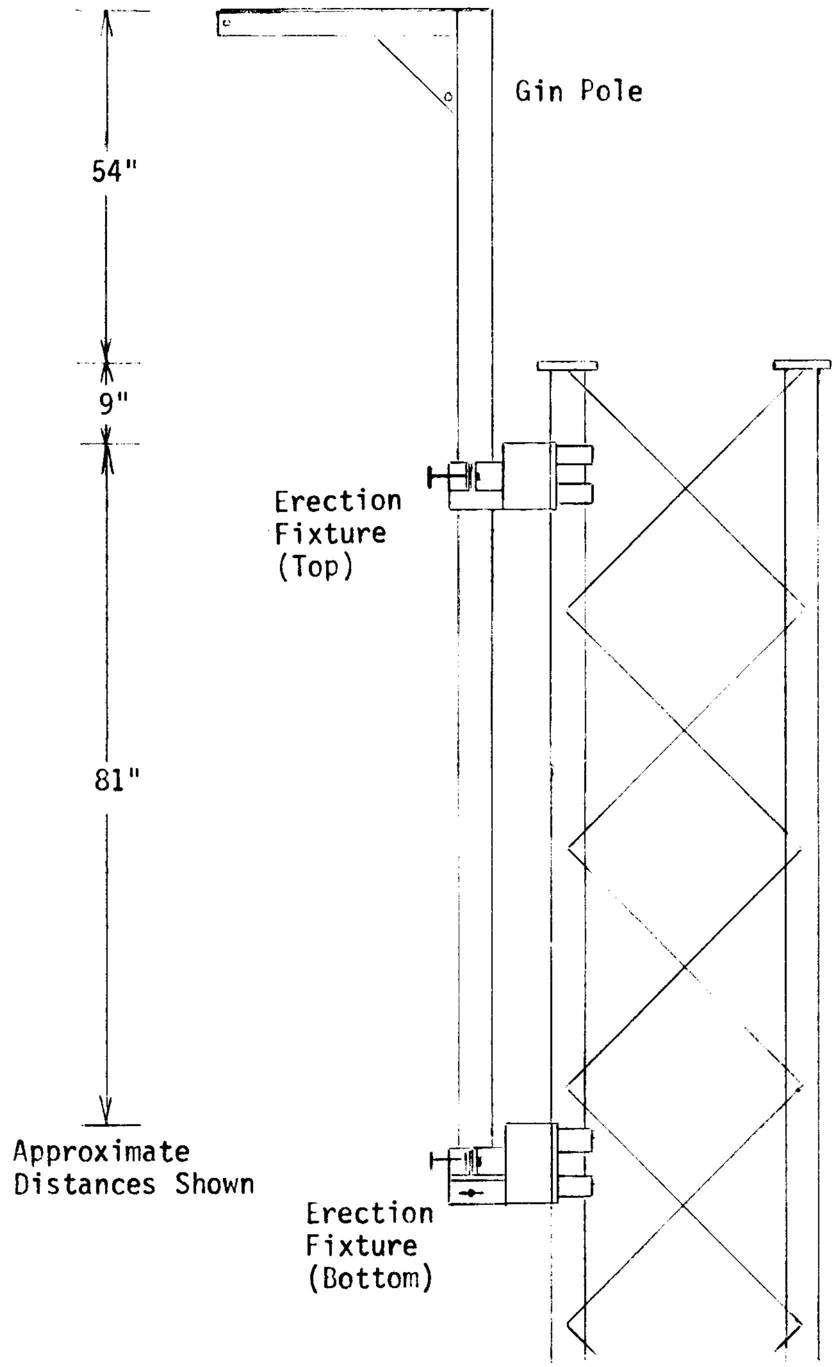


Figure 67

Gin Pole and Erection Fixtures

2. The winches are attached to the tower leg at a convenient working height. The main winch will be on the same leg as the gin pole. Viewed from the top, the auxiliary winch will be on the next leg, clockwise.

3. Tag snatch blocks are fixed to truck bumpers and the trucks positioned as shown in Figure 68. (Positioning is also a function of the surrounding terrain.) Truck wheels are chocked as a precaution. The trucks should not be moved to control an ascending unit.

4. The unit to be raised is positioned about 2 feet in front of the tower face with the winches between the trucks. After the main winch cable and tag lines are connected to the units, the tag lines are tightened to prevent the machine from hitting the tower as it leaves the ground. A second person climbs the tower. As the unit is raised with the main winch, the tag lines are used for control. The person operating the main winch coordinates the operation, and should keep away from under the unit as it is raised. After the unit is raised to the desired height, it is positioned by the two people on the tower. The tag lines are only removed after it is clear that the persons on the tower have complete control over the movements of the unit. The winch cable is removed only after the unit is bolted securely in place.

5. Units are installed in the following sequence:

a. The gin pole is raised and fixed to the erection fixtures with the gib facing out (Figure 67).

b. The work platform is fixed to the face above the two winches and strapped in place with quick-acting ratchets.

c. The tower top plate, with the levelling blocks assembled, is raised with a lifting fixture that raises it horizontally. One lifting point should be attached to the winch cable, so that the overall height of the plate and cable fixture remains low. The tower top plate is bolted to the three flanges on the tower legs.

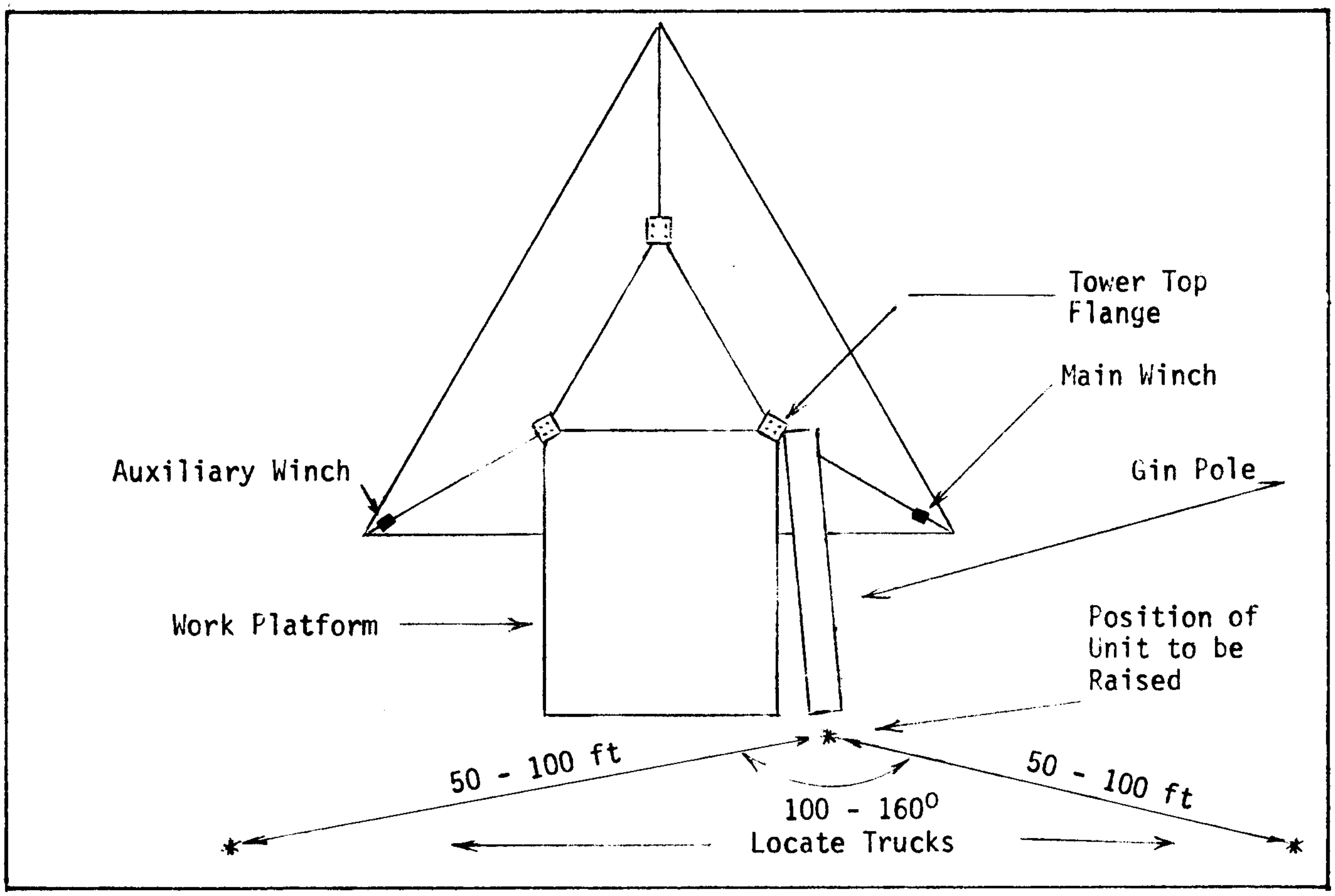


Figure 68

Installation Plan Layout (Top View)

d. The mainframe, including the yaw assembly is lifted into place. An eyebolt is fastened to the mainframe into a predrilled hole at its center of gravity, and the winch cable is attached to it. While the mainframe is suspended, the levelling plate is bolted to the levelling blocks on the tower and tightened with a percussion wrench. The frame should be yawed so that the gearbox mounting plate is over the work platform. The yaw brake is applied.

e. The gearbox and hub are raised together. A lifting fixture which fits on the hub and gearbox, with a single lifting point to be attached to the winch cable, is used to bring the components up horizontally. The gearbox is bolted in place, using percussion wrenches to achieve the desired torque on the bolts. The hub is rotated so that one leg faces directly downward. The auxiliary winch cable is hooked to the right leg of the hub (as seen from behind the gearbox) and the cable fits over a fixture bolted to the left leg of the hub, (see Figure 69).

f. The first blade is raised using the main winch. Two harnesses are used, one near the root for fastening the main winch cable, and the other three feet from the tip for the tag lines. The blade is bolted to the hub, using percussion wrenches. If there is lack of wrench space, the three bolts between the hub and gearbox may be tightened after the blade has been rotated 120°. One tag line is retained on the blade tip.

The auxiliary winch is used to rotate the hub. The fixture on the left leg of the hub is designed to release the cable after it no longer needs to bear against it. This will happen just before the hub has rotated 120°. When the (left) leg reaches the bottom position, this position is held with the auxiliary winch, and the second blade is raised. It is controlled by two tag lines, one attached to a truck snatch block and the other hand held. After the blade is bolted, the hand held tag line is disconnected from the blade.

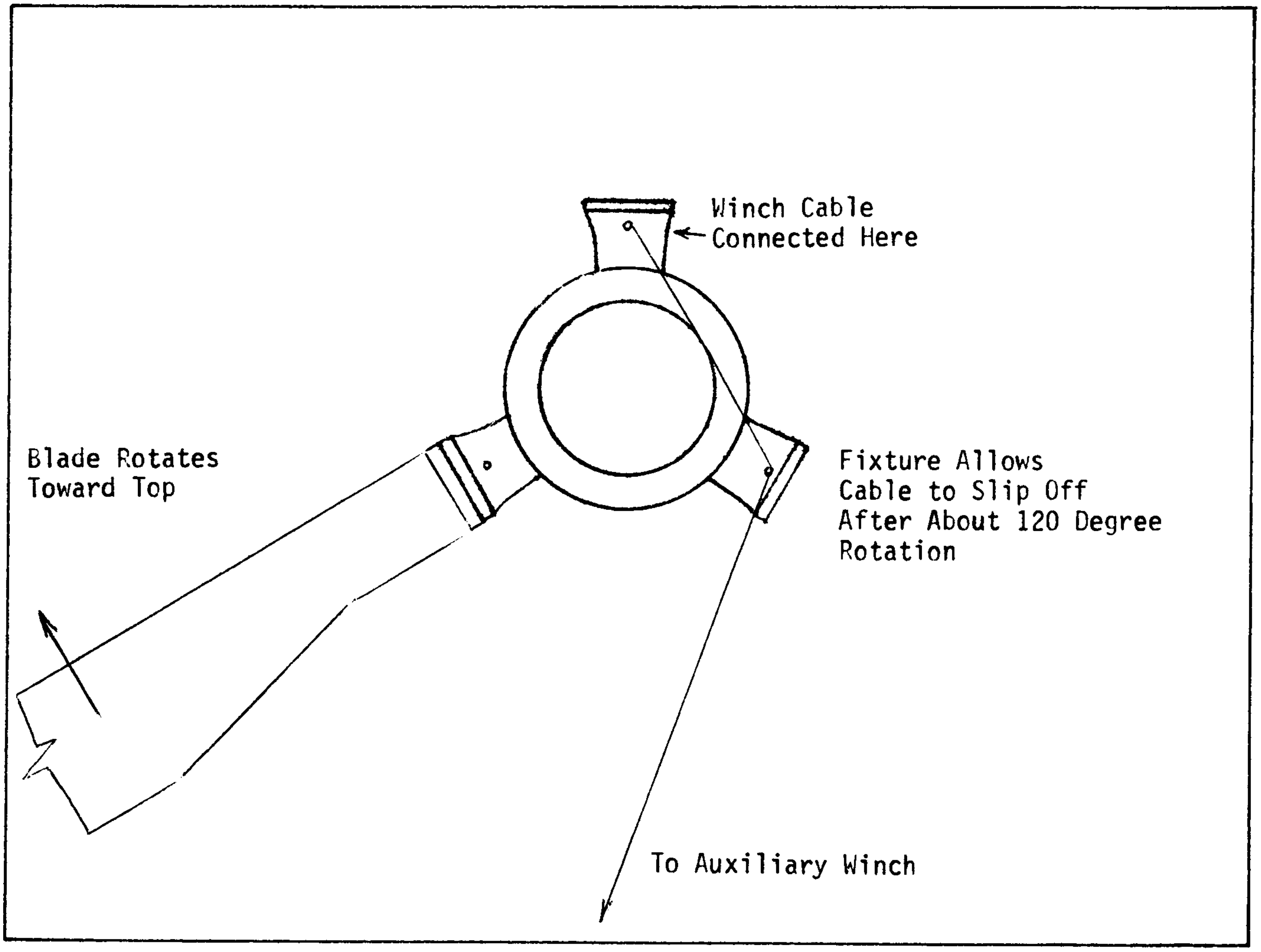


Figure 69

Blade Installation Fixtures

The auxiliary winch is used again to bring the last leg to the bottom. Two persons operate each of the tag lines from the trucks and aid in maintaining stability of the rotor, while the third blade is raised with the main winch and controlled with two tag lines which are handheld. After the blade is bolted, the tag lines at the bottom of the blade are removed. The auxiliary winch cable is retained on the hub.

g. The motor-brake assembly is raised using the main winch and two handheld tag lines. When engaging the motor into the gearbox coupling it may be necessary to rotate the gearbox shaft. This may be done by operating the auxiliary winch and the tag lines on the upper two blades. The motor brake assembly is bolted to the gearbox flange. At this point the tag lines, harnesses, and winch cables may be removed. This may be delayed on the upper blades until electrical connections have been completed and the blades can be motored to a position adjacent to the tower.

h. The spinner is raised and attached to the hub. One person may be positioned atop the gearbox to do this. The nacelle bottom, consisting of two halves, is raised and attached to the mainframe assembly. Finally, the top is raised, installed, and latched.

i. The machine should now be carefully leveled and the electrical connections completed.

5.3 Design of Gin Pole

The gin pole currently used by Enertech on installations is made of 6061 T6 aluminum alloy with yield strength of 40 KSI and ultimate strength of 45 KSI. The ginpole consists of a tube with 3-in O.D. and 3/8-in wall thickness, (see Figure 67).

The heaviest load to be lifted in installation will be the gearbox and hub, taken together at 1100 lbs. This would yield a safety factor of 1.2 over predictable loads. To reach a safety factor (SF) of 3, as is used by installation companies, the material can be changed to 7075-T6 with a yield strength 73 KSI and ultimate strength 83 KSI, or a larger pole can be used. The former solution would keep the weight of the pole to about half of that associated with a larger diameter pole. However, 7075-T6 is not easily available except in very large orders, and can be considered only at a high volume level. The chosen pole size is therefore 4-in O.D. with 1/2-in wall thickness, using 6061-T6 material.

In calculating the stresses on the gin pole, (see Figure 70), the following assumptions are made:

1. The pulleys on the gin pole are frictionless.
2. The force exerted by a person on a tag line does not exceed 150 lbf.
3. Tag lines are inclined to the vertical by 45°.

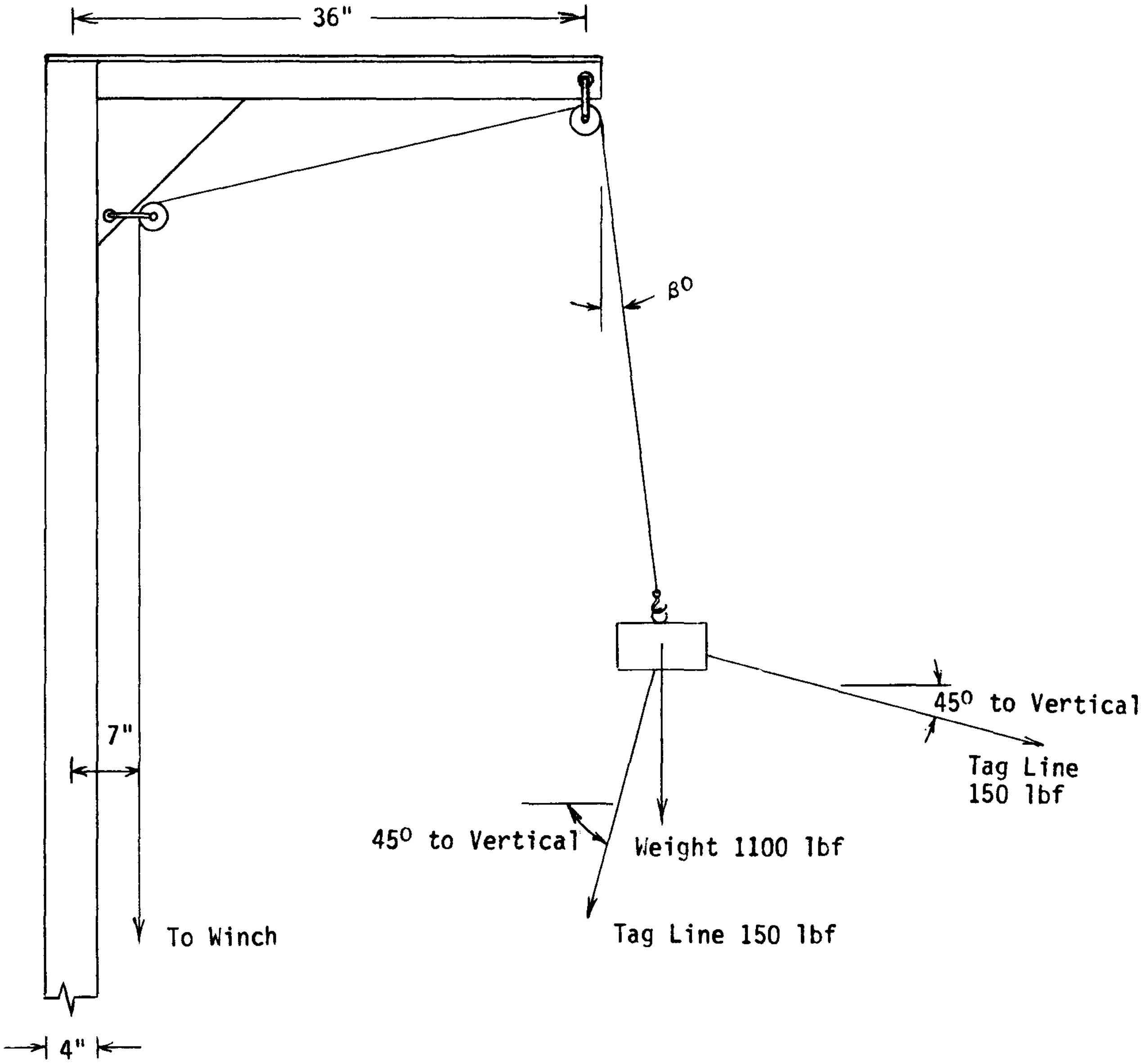
A SF of 3 is used because of the possibility of danger to personnel from a broken or even a yielding gin pole. The gib section of the pole is known to be much more rigid than the tubing wall because of its deep construction.

$W = 1100$ lbf

$T \cos \beta = 2(150 \cos 45^{o}) + W = 1312$ lbf

If $\beta = 10°$, $T = 1332$ lbf

This SF will be adequate for shock loadings, stuck winches or cables, and provide safe design.



Moments on gin pole for weight of 1100 lbf:

Due to lifting weight	=	$T \cos \beta$ (36)	=	1312(36)
Due to winch cable	=	T (7)	=	1332(7)
		Moment	=	48156 lbf/in

$\frac{I}{c}$ for O.D. 4-in I.D. 3-in = 4.30-in^3

Stress = 48156 = 1119 lbf/in^2

Safety Factor = $\frac{40000}{11119}$ = 3.6

Figure 70

Gin Pole Stress Calculations

6.0 SYSTEMS ANALYSES

6.1 Availability Analysis

6.1.1 Introduction

Wind machine availability is the probability that the machine will operate according to specifications at any given time. Yearly availability, Pa, is calculated as follows:

$$Pa = \frac{\text{hours in a year - annual down time}}{\text{hours in a year}} \qquad (1)$$

Machine downtime is a function of active repair time, logistic repair time (i.e. obtaining replacement components), administrative time (i.e. scheduling service personnel), and general maintenance time. Because logistic time and administrative time depend largely upon the organizational service structure, these values can only be roughly predicted. For comparison, availability has been calculated for a site specific case where the wind machine is located 150 miles from an Enertech dealer.

Intrinsic availability (Pa_i) or the ideal availability that can be engineered into the wind machine, is defined as follows:

$$Pa_i = \frac{\text{hours in a year-annual maintenance time-active repair time}}{\text{hours in a year}} \qquad (2)$$

Therefore, design priorities for achieving 95% availability must include 1) a low failure rate for the wind machine, 2) an easy-to-service wind machine.

6.1.2 Analysis

Both site-specific availability and intrinsic availiability have been calculated for the proposed wind machine design. For site-specific availability it is assumed that 1) two days elapse before the failure is detected and 2) except where spare components are available at the wind site, an additional five days are required for logistic and administrative functions before active repair can begin. For intrinsic availability it is assumed the 1) the time elapsed before the failure is detected is nil and 2) that no time is necessary for logistic and administrative functions as all replacement components and all necessary service personnel are available at the wind site.

Availabilities are calculated by summing the individual component down times and substituting the down times into equations 1 and 2. Component down time is calculated as follows:

$$\frac{\text{downtime}}{\text{component}} = \frac{\text{failures}}{\text{years}} \times \frac{\text{time-to-repair}}{\text{component failure}} + \text{annual maintenance time} \quad (3)$$

Calculations for component failure rates are explained in Section 6.3 (Reliability Analysis). Calculations for annual maintenance time are explained in Section 6.4 (Maintainability Analysis).

Results - The results of the availability analysis are as follows:

Site Specific Availability	Intrinsic Availability
Pa = 99.0%	Pa_i = 99.8%

Both these figures are better than the design availability of 95%. It must be remembered however, that these values are preliminary and serve largely as a design tool. A summary of component values is shown in Table XIX.

6.1.3 Conclusions

As a result of the analysis the following conclusions have been reached:

1) Components with high failure rates must be easy to service. For instance, a maintenance adaptor should be available for servicing the drive train on the tower.

Table XIX
Availability Analysis Results

Item	Mean Time to Repair: Failures/Year	Mean Time to Repair: Intrinsic (hrs)	Mean Time to Repair: Site Specific	Annual Repair Time (hr): Intrinsic	Annual Repair Time (hr): Site Specific	Routine Maintenance	Total Downtime: Intrinsic	Total Downtime: Site Specific
1. Rotor								
1.1 Blade	SAFE LIFE DESIGN					1	1	1
1.2 Fasteners	.00028	.25	48.25	.00007	.014	.5	.5	.514
1.3 Cast hub	SLD					.5	.5	.5
1.4 Taper Lock	.00128	8	176	.010	.225	.25	.260	.475
2. Gearbox	.02481	16	184	.397	4.565	1	1.397	5.565
3. Gearbox/Generator Coupling	.00071	6	174	.00426	.124	.5	.504	.624
4. Induction Generator Motor	.11025	7	175	.772	19.294	.25	1.022	19.544
5. Rotor brake	.05983	4	172	.239	10.291	.5	.739	10.791
6. Slip rings								
6.1 Connectors	.00897	.25	48.25	.002	.433	.25	.252	.683
6.2 Wiring	.00769	1	49	.008	.377	.25	.258	.627
6.3 Slip ring case	SLD							
6.4 Connectors	.00897	.25	48.25	.002	.433	.25	.252	.683
6.5 Collector rings	.1282	4	172	.513	22.050	.5	1.013	22.550
6.6 Collector brushes	.02564	1	1	.026	.026	1.0	1.026	1.026
7. Lead in Wires								
7.1 Connectors	.00897	.25	48.25	.002	.433	.5	.502	.933
7.2 Wires	.0077	1	169	.008	1.3	.5	.508	1.801
7.3 Connectors	.00276	.25	48.25	.0007	.133	.5	.501	.633
8. Control System	.0526	4	172	.210	9.047	1	1.210	10.047

Table XIX (Cont'd)
Availability Analysis Results

Item	Mean Time to Repair: Failures/Year	Mean Time to Repair: Intrinsic (hrs)	Mean Time to Repair: Site Specific	Annual Repair Time (hr): Intrinsic	Annual Repair Time (hr): Site Specific	Routine Maintenance	Total Downtime: Intrinsic	Total Downtime: Site Specific
9. Nacelle								
9.1 Nacelle	SLD				.014	.25	.25	.25
9.2 Fastening subsystem	.00028	.25	48.25	.0007	.01351	.25	.25	.264
10. Bed Plate								
10.1 Fasteners	.00028	.25	48.25	.00007	.014	.25	.25	.264
10.2 Steelcomp	SLD					.5	.5	.5
11. Yaw Pivot Assembly								
11.1 Turntable bearing	.00926	16	184	.148	1.704	.5	.648	2.204
11.2 Turntable seals	.0037	16	184	.059	.681	.25	.309	.931
11.3 Bearing fasteners	.00028	.25	48.25	.00007	.014	.25	.250	.264
11.4 Steel Components	SLD					.75	.75	.75
11.5 Leveling System	.00085	.25	48.25	.0002	.041	.25	.250	.291
12. Tower								
12.1 Tower Foundation	SLD							
12.2 Tower	SLD					.75	.75	.75
12.3 Fastener	.00028	.25	48.25	.00007	.014	.75	.750	.764

AVAILABILITY	
Ideal	Site Specific
99.8%	99.0%

2) Components with relatively high failure rates can be placed in parallel; (i.e. the slip-ring brushes).

3) Inexpensive components that are easy to service should be available at the wind site. For the analysis, it was assumed that a spare parts kit located at the wind site included electrical connectors, fasteners, wires, an anemometer, fuses, and slip-ring brushes.

4) Components that require excessive down time when they fail should be designed to last the life of the machine. For this reason, heavy duty bearings that perform throughout the life of the wind machine should be considered in the design.

5) Routine maintenance should include inspection of components that will wear out. Before these components fail, they should be replaced, thereby avoiding subsequent wind machine failures.

6.2 Failure Mode and Effects Analysis

The Failure Mode and Effects Analysis outlines the possible failure modes of the wind machine together with the effects of these failures, the possible causes, and the preventive and corrective actions.

In the Failure Mode and Effects Analysis, the severity of the various component failures has been rated as follows:

Severity Rating Categories

1. Failure causes no immediate loss of generating capability. Some repairs needed during next scheduled maintenance to prevent subsequent failure and system shutdown. No injuries.

2. Failure causes shutdown of wind machine or significant degradation of performance. Immediate repair desirable to prevent subsequent failures and restore system to normal operation. Repair could be completed in one work day. No injuries.

3. Failure causes prolonged wind machine shutdown. Machine cannot be operated without repair. Major element fails requiring more than one day for repair. Possibility of injury or damage to other property.

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
1.	Rotor						
1.1	Blades	Blades roughened	Power output decreased	1	Blade subject to extreme environmental conditions as sand storms, blade struck by foreign object	Blade surfaces are protected by gel coat cover, under gel coat is a layer of fiberglass, blade surfaces are well protected	Repair gel coat finish
		Blades cracked, or broken	Possible rotor imbalance, increased cyclic loads on gearbox shaft, loosening of fasteners possible	2 or 3	Fatigue or maximun loads higher than expected, foreign objects striking blades	Safe life design criteria for blades	Replace blade
1.2	Stud fastener	Loosening of nuts	None immediately, could result in loss of blade	1	Failure to torque nut to specified value	Nut will be checked for tightness during maintenance, locking nut used	Torque nuts to specified value
		Failure of stud	Possible loss of blade	2	Corrosion, loads higher than expected	Safe life design for studs	Replace blade/studs configuration
		Failure of blade/ stud bonding resin	Possible loss of blade	2	Improper bonding of resin	Safe life design	Replace Blade/stud configuration
1.3	Cast hub	Structural failure as yielding or cracking	Loss of blade, vibration, unit shutdown	3	Loads in excess of design loads	Safe life design	Replace hub
1.4	Taper lock bushing	Loss of locking friction	None, retaining nut secures hub	1	Not tightened to specifications	Safe life design, rotor locking nut is back-up to taper bushing	Tighten bushing to specifications

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
2.	Gearbox assembly						
2.1	Cast housing	Crack or hole in casting	Loss of lubrication, could lead to bearing failure, severely damaged housing would lead to unsupported shaft load	1 or 2	Material properties below manufacturer's specifications	Initial inspection of unit, inspection during maintenance and repair	Replacement of unit if functionally damaged
2.2	Helical gears	Gear tooth failure	Increase noise and vibration, may cause jamming of gearbox, decrease life of gearbox	1	Improper assembly of shaft or bearings, improper manufacturer tolerences, loads in excess of design condition	Initial inspection of unit, measurement of initial shaft play, insured quality control of unit, safe life design of gears	Replace gear or replace gearbox
		Stripping of gears	Disengagement of geartrain, disengagement of rotor from brake, no power transmitted, tip brakes deploy	3	Excessive play and backlash of shafts, foreign objects in gearbox, loads in excess of design condition	Inspect unit initially to insure quality control standards, inspect oil following oil change for foreign objects	Replace gear or replace gearbox
2.3	Front seals	Lubrication leaking from seals	Loss of lubrication, could lead to bearing failure	1	Worn out seal, unseating of seal during installation or vibration	Regular inspection of seals, front seals are a redundant subsystem	Replace seal
2.4	Back seal	Lubrication leaking from seal	Loss of lubrication, could lead to bearing failure	1	Worn seal, improper installation	Regular maintenance	Replace seal

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
2.5	Gearbox shafts	Crack or break	Failure to transfer rotor torque, loss of power, uncoupling of rotor and brake, if lowspeed shaft fails, could lead to unsupported rotor	3	Loads in excess of design condition	Safe life design - all gearbox shafts analyzed for both fatigue and maximum force condition	Replace gearbox or replace shaft
2.6	Gearbox bearing	Excessive bearing friction	Excessive heat in gearbox, decrease power production, reduce bearing and gearbox life	2	Improper shaft alignment, broken ball bearing, no lubrication	Provide proper lubrication during maintenance, inspect gearbox initially to insure quality control standards	Replace lubrication, replace bearing if necessary
		Loose bearing	Noise or vibration, decrease bearing life, decrease gearbox life	1	Wearing of bearing race or shaft	Inspect for manufacturer tolerences initially to insure quality control standards	Replace bearing

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
3	Gearbox/generator coupling	Slipping of coupling	Torque transmitted is decreased, decrease power production, heat produced in coupling	2	Worn or improperly installed coupling		Repair or replace coupling
		Broken coupling	No torque transmitted, no power produced	3	Loads in excess of design conditions		Replace coupling
4	Induction generator	Overheating of windings	Blows fuse, decreased life of insulation, unit shutdown and no power is produced	2	Poor ventilation, torque over-loading	Fuse protection, open drip proof generator design (excellent ventilation), generator sized to handle expected torques	Replace fuse, check for blocked air passages
		Short circuit on windings	Circuit breaker trips, insulation damaged, unit shutdown and no power is produced	2	Lightning or wet insulation, overheating of windings	Lightning protection high temperature winding insulation	Check windings and replace as needed
		Bearing failure	Scoring of motor shaft, overheating of shaft, possible shut-down due to vibration	2	Shaft misalignment, lack of lubrication	Lubricate as part of routine maintenance	Replace bearing
		Overspeed	Cause bearing failure or decreased bearing life	2	Failure of control system and failure of tip brakes	Unlikely, tip brakes prevent overspeed	Check bearings, replace as needed

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
4	Induction Generator (cont)	Capacitor failure	Overheating in motor	2	Lightning, current over-load wear out of capacitor, non-conformance to specifications	Fuse protection of motor, lightning protection	Replace capacitor
5	Rotor brake	Failure to apply adequate torque	Overheating of brake, rotor could over-power brake and overspeed, tip brakes would deploy	2	Mechanical or hydraulic failure, excessive wear of brake seals, low fluid level	Brake unit is enclosed, unit will be inspected during routine maintenance procedures, worn brake parts to be replaced	Replace part, check brake operating pressure
		Failure of brake to apply any stopping torque	Rotor would overspeed, tip brakes deploy, no power produced	2 or 3	Mechanical or gear failure, fluid loss, broken generator/ brake shaft	Tip brakes prevent rotor overspeed	Inspect brakes, repair or replace as necessary
		Failure of brake release	No power produced	2 or 3	Brake electrical failure, mechanism failure		Inspect lead-in lines, inspect brake
		Erratic brake drag	Decreased power production, decreased brake life	2	Sticking solenoid or foreign matter on solenoid	Inspection during maintenance should show components with excessive wear	Repair or replace components as necessary
6	Slip ring assembly						
6A	Power slip rings						

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
6.1A	Connectors	Corrosion or foreign matter on connector	Short circuit to ground, loss of power	2	Moisture resulting in corrosion, foreign matter on connector	Connectors are adequately sized so that small foreign objects (unless metal) have little effect	Repair or replace connector
		Loose connector	Intermittent power	2	Vibration	Connectors will be checked during routine maintenance for firm connections	Tighten connector
6.2A	Wiring	Breakdown of insulation	Could cause short circuit to ground, loss of power; or short circuit between rings and brushes opening circuit breaker and failure to produce power	2	Worn insulation with conductor in contact with wire	Inspection of wire's worn insulation	Repair or replace as needed
6.3A	Slip ring case	Crack or break	None initially, moisture and environment in contact with rings resulting in excess wear, or decrease system life	1	Foreign matter in contact with cover, excessive vibration	Case will be inspected	Repair or replace as needed
6.4A	Connectors	(as above)					
6.5A	Power collector rings	Corrosion of slip ring	Heating of slip ring assembly, intermittent power producing open circuit and loss of power	2	Damage to slip ring case and moisture in contact with rings	Periodic inspection of rings, cleaning away small corrosion problems during annual maintenance procedures	Clean rings with crocus cloth, replace if necessary

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
6.5A	Power Slip Rings (cont'd)	Foreign matter in contact with rings and brushes	Short circuit between rings and brushes, opens circuit breaker, loss of power	2	Damage to slip ring case	Periodic inspection during routine maintenance	Repair as necessary
6.6A	Power ring brushes	Misalignment of brushes	Open circuit and loss of power; intermittent power production	1	Improper installation or assembly, loosening of support perhaps due to vibration	Brushes checked for proper alignment during maintenance operations, brushes are in parallel with each other for high reliability of conductor subsystem	Replace damaged brushes
		Corrosion of Brushes or foreign matter on brushes	Short circuit between rings and brushes resulting in loss of power; intermittent contact resulting in intermittent power production; open circuit resulting in loss of power	1	Failure of slip ring case and excessive moisture in contact with slip rings	Small corrosion problems cleaned during routine maintenance, double brushes increase reliability for electrical conduction of power	Replace or repair damaged brush
6B	Brake slip ring assembly						
6.1B	Connectors	Corrosion or foreign matter on connector	Short circuit, brake does not release, no power produced	2	Moisture in contact with connector	Routine maintenance procedures to insure quality of connectors	Replace connector
		Loose connector	Intermittent contact, brake cycles, decreased life of brake	2	Vibration or improper installation	Connections checked for tightness during routine maintenance	Tighten connector

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
6.2B	Wiring	Worn insulation or breakdown of insulation	Short circuit, brake does not release, no power is produced	2	Worn insulation in contact with live conductor	Wires checked for wear	Splice in new wire section
6.3B	Slip ring case	Crack or break	None immediately, possible damage to slip ring components, decrease in subsystem life	1	Foreign object in contact with with case, excessive vibration	Case inspected during routine maintenance	Repair or replace
6.5B	Brake collector rings	Corrosion of rings	Open circuit and brake does not release; intermittent contact and brake cycles, brake could cause motor to overheat and system to shutdown	2	Moisture in contact with rings	Periodic inspection should lead to the clean up of any minor corrosion	Clean rings with crocus cloth
		Foreign matter in contact with rings on brushes	Short circuit and brake does not release	2	Damage to slip ring case	Slip ring case inspected during routine maintenance	Repair as necessary, remove foreign material
6.6B	Brake slip ring brushes	Misalignment of brushes	Open circuit and brake does not release; intermittent contact and brake cycles, perhaps decreasing brake system life	1	Improper installation or assembly, loosening of support brackets due to vibration	Brushes checked for proper alignment, brushes in parallel for high reliability	Re-align brushes
		Corrosion of brushes or foreign matter on brushes	Short circuit between rings and brushes where brake does not release; intermittent electrical contact where brake cycles		Moisture or foreign matter in contact with brushes	Small corrosion problems cleaned during routine maintenance, double brushes increase reliability of connection	Repair or replace brushes as needed

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
7.	Lead-in wiring						
7.1	Connectors	Corrosion or foreign matter on connector	Short circuit - brake will not release, power will not be produced,or instrument data will not be relayed	2	Moisture or foreign matter in contact with connector	Connectors will be inspected periodically for quality assurance	Replace connector
		Loose connectors	Intermittant connection resulting in intermittent power production, brake cycling, or intermittent data signals	2	Vibration or improper installation	Connectors inspected periodically for quality assurance	Replace connector
7.2	Wiring	Broken wire	Open circuit resulting in loss of power, an unreleased brake, or no data transmittance	2	Wire in contact with foreign material, stress on wire	Wires are protected in a conduit	Splice new wire section or replace wire
		Damaged insulation	Short circuit resulting in loss of power, an unreleased brake, or no transmittance of data	2	Wearing of insulation	Wires are protected in a conduit	Splice in new wire section or replace wire
8.1	Generator RPM sensor and associated circuitry	No signal	Low RPM shutdown after startup, brake applied, no power produced	2	Misalignment of coil, poor wiring connectors, faulty slip rings, or circuit failure	Conservative design, frequency(rather than voltage)signal	Troubleshoot wiring and sensor, replace component as needed

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
8.1	Generator rpm sensor (cont'd)	Reads too high	Generator cuts in below synchronous speed. Power consumed in motoring generator. False overspeed shutdown may occur under normal operation, manual reset required	1 or 2	Noise in circuitry	Use frequency signal, use twisted shielded paired instrument cable	Check shield for proper grounding
		Reads too low	Generator may switch on at higher than desired RPM, energy lost Generator may overspeed to higher than desired RPM before shutdown, loads increased				
8.2	Wind speed sensor and associated circuitry	No signal	Wind machine will not start; if running, it will not shut down in high winds until overspeed occurs, loads on machine may be higher than designed for	2	Icing of anemometer head, faculty sensor, anemometer head broken off, faulty slip rings, bad wire connection, circuit failure	Time proven anemometer used, instrument grade slip rings	Inspect and replace anemometer head, troubleshoot wiring or circuit board
		Reads too low	Start-up at higher than desired wind speed: Energy lost	1 or 2	Damaged anemometer cup, faulty circuitry	Time proven anemometer used, conservative circuit design	Inspect and replace anemometer head, troubleshoot circuit board

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
8.2	Wind speed sensor (cont'd)	Reads too low	High wind shutdown at higher than desired wind speed: loads increased				
		Reads too high	Machine starts up when wind speed is too low; wind may be too low to produce power	1 or 2	Noise in signal	Frequency signal used for noise immunity, twisted shielded paired instrument cable	Check shield for proper grounding
8.3	Vibration sensor	Fails open	Does not protect the machine from excessive vibration	1	Failure of sensor, faulty wiring		Replace sensor
		Fails closed	Shutdown, requires manual reset	2	lightning, mechanical failure of sensor	Lightning protection	Replace sensor
8.4	Generator output frequency sensor	No reading or incorrect reading	Shutdown	2	Circuit failure		Troubleshoot, repair circuitry
8.5	Start contactor and control circuitry	Fails open	No start-up	2	Coil failure, faulty wiring, circuit malfunction	Conservative contactor design	Replace coil, troubleshoot wiring

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
8.5	Start contractor and control circuitry (cont'd)	Fails closed	Generator motors to 1800 RPM, run contactor does not close. Possible overspeed because generator provides little counter torque, resulting in shutdown. Normal power output is not obtained	2	Contacts welded Circuit malfunction	Conservative contactor design	Replace contactor repair circutry
8.6	Run contractor and control circuitry	Fails open	No power produced	2	Coil failure faulty wiring circuit malfunction	Conservative contactor design	Replace coil replace contactor repair circuitry
			Generator does not provide counter torque to rotor, overspeed may result				
		Fails closed	Motoring in low winds, power consumed	2	Contacts welded Circuit malfunction		
			During shutdown, motor opposes brake. Heating of brake, blows fuse				
8.7	Brake relay and control circuitry	Fails closed	Brake is inoperative, rotor may overspeed in high winds with tip brakes deploying		Electronics failure Welded contacts	Conservative relay design	Repair circuitry replace relay

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
8.7	Brake relay and control circuitry	Fails open	Brake is not released. Fuse blows when start-up attempted no power produced	2	Faulty wiring Coil failure Electronics failure	Conservative relay design	Repair circuitry replace relay
8.8	Control Panel power supply	Not operating	Shutdown	2	Utility Line failure Component failure		
8.9	Lightning Protection Equipment	Fails open	Does not protect against surges	1	Lightning		
		Fails closed	Burns out quickly, becomes open circuit	1	Lightning		
8.10	Tip brakes	Open at lower RPM than desired	Loss of power output. Brake must be manually reset.	2	Mechanical failure		Inspect components and replace as necessary
		Do not open or open at higher RPM than desired	May allow machine to overspeed to higher speed than desired. Loads may be higher than designed for	1 or 2	Mechanical failure Jamming of tip brakes or release mechanisms by ice or foreign matter		Inspect components and replace as necessary Clean foreign matter from mechanism
		Open at different speeds	Decrease in power output	1 or 2	Jamming of one tip brake Mechanical failure of one tip brake		Clean foreign matter from mechanism Inspect components and replace as necessary

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
8.10	Tip brakes (cont'd)	Open at different speeds	Rotor unbalance: may cause vibration shutdown		Mechanical failure of one tip brake		Inspect components and replace as necessary
9.	Nacelle						
9.1	Nacelle	Cracking or breaking of fiberglass material	None immediately, wind machine unprotected from environment resulting in possible damage to other components, anemometer could be damaged or disconnected	1 or 2	Foreign material impacting the nacelle, stress in excess of design condition		Repair fiberglass or, if necessary, replace nacelle
9.2	Fastening subsystem	Loosening or failure of fastener	Nacelle could be disconnected from unit and possibly go through the rotor	1	Vibration, corrosion of fasteners, fasteners not torqued to design value	Fasteners will be "locked" either by locking nuts or by a locking adhesive, vibration shut-down switch to prevent prolonged vibration	Tighten or replace fasteners
10.	Bed frame assembly						
10.1	Fastening subsystem	Loosening or failure of fastener	Gearbox may be free to move	2	Excessive vibration, loads in excess of design condition, fastener not torqued to design value	Bolts secured by locking nuts, vibration shut-down switch to prevent prolonged vibration	Tighten or replace fastener

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
10.2	Frame beam	Cracking or yielding of beam	Failure to support gearbox at proper orientation, decreased performance; if severe, blades could strike tower	3	Metal not conforming to design specifications, fatigue loads or maximum loads in excess of design condition	Safe life design-beam analyzed for both fatigue and ultimate strength	Replace beam
11.	Yaw pivot assembly						
11.1	Turn table bearing	High bearing friction	Poor yawing characteristics, decreased performance	2	Improper lubrication of bearing, broken or chipped ball bearing, poor alignment of bearing, unseating of seal resulting in loss of lubrication	Lubricate during routine maintenance, bearing loads have been reviewed, machined surface for bearing	Lubricate bearing, check bearing seals, replace bearing if necessary
		Bearing loose on races	None immediately, could lead to decreased life of bearing	1	Excessive wear of race	Bearings checked during periodic maintenance	Replace bearing
		Failure of seal or unseating of seal	Leakage of lubrication, could lead to bearing failure	1	Worn seal, improper installation	Bearing inspected during maintenance	Replace seal or replace bearing as necessary

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
11.2	Steel Components	Cracking or yielding	Slip rings unsupported	2	Loads in excess of design condition	Yaw tube now located in bearing - no longer subject to loads except those of slip rings	Replace component
11.3	Fastening subsystem	Loosening or structural failure of bolts	Loosening of leveling plates	2	Excessive vibration, improper installation of fasteners	Fasteners torqued to specified value, locking fasteners used	Torque fastener to necessary value, replace if necessary
11.4	Tower top adaptor	Cracking or yielding	Loss of support for wind machine	3	Loads in excess of design conditions	Safe life design	Replace tower top adaptor
11.5	Fastening subsystem for tower top	Loosening or structural failure of bolts	Loosening of tower top adaptor	1	Vibration, loads in excess of design condition	Vibration switch to prevent prolonged vibration	Torque fastener to specified value, replace if necessary
12	Tower						
12.1	Foundation	Structural failure	If minor - could cause tower to tilt, decrease performance. If major - could cause collapse of tower	2 or 3	Crushing of concrete, yielding of steel reinforcement	Foundation design so that steel yields before concrete crushes decreasing chance of major failure, safe life design	Move tower and wind machine to different site
		Excessive settling of foundation	If minor - tower would tilt and performance decrease, If major - tower could collapse	2 or 3	Excessive movement of soil	Adequately sized foundation so that local settling effects will be small	Depending on failure, tower/machine could be moved

FAILURE MODES AND EFFECTS ANALYSIS

ITEM NO.	COMPONENT SUBSYSTEM	FAILURE MODE	EFFECT	SEVERITY	PROBABLE CAUSE	PREVENTATIVE ACTION	CORRECTIVE ACTION
12.2	Tower	Structural yielding, buckling, or cracking	Tower could fail to support wind machine	3	Loads in excess of design condition, failure of support welds	Safe life design	Replace tower and re-install wind machine
12.3	Fastener subsystem	Loosening or failure of fasteners	Tower could fail to support wind machine	1	Vibration	Vibration switch should prevent prolonged unit vibration	Torque fastener to design value, replace fastener if necessary

6.3 Reliability Analysis

6.3.1 Introduction

Failure rates for components and component subsystems were predicted based upon generic failure rates, derating factors, and environmental conditions. Component failure rates, (λ COMP) were calculated as follows:

$$\lambda_{COMP} = \lambda_{GENERIC} \times K_a \times K_b \times K_c$$

where λ_{GEN} = generic failure rate
K_a = number of similar components in series
K_b = derating factor
K_c = environmental factor

The failure rate of the wind machine is the sum of the failure rates for the components in series. Figure 71 is the reliability block diagram used for the analysis.

6.3.2 Analysis

Generic failure rates are based on laboratory test data for a large number of components operating under specified conditions. Since the configuration of the control system for the 10,000th unit has not been specified at this time, no attempt has been made to determine its failure rate based on its component parts. Instead, a rough estimate of its failure rate has been used in the analysis. The failure rate used for the control system is on the same order as the highest failure rates for the other components of the system. The steel components have been designed for infinite life or "safe life design" (SLD). Statistical data on material endurance limits is incorporated into the structural analysis.

Because generic failure rates result from testing components at rated specifications, failure rates for components which operate below specifications have been reduced. The induction generator and the transmission system have been derated based upon derating curves which incorporate manufacturer derating information.

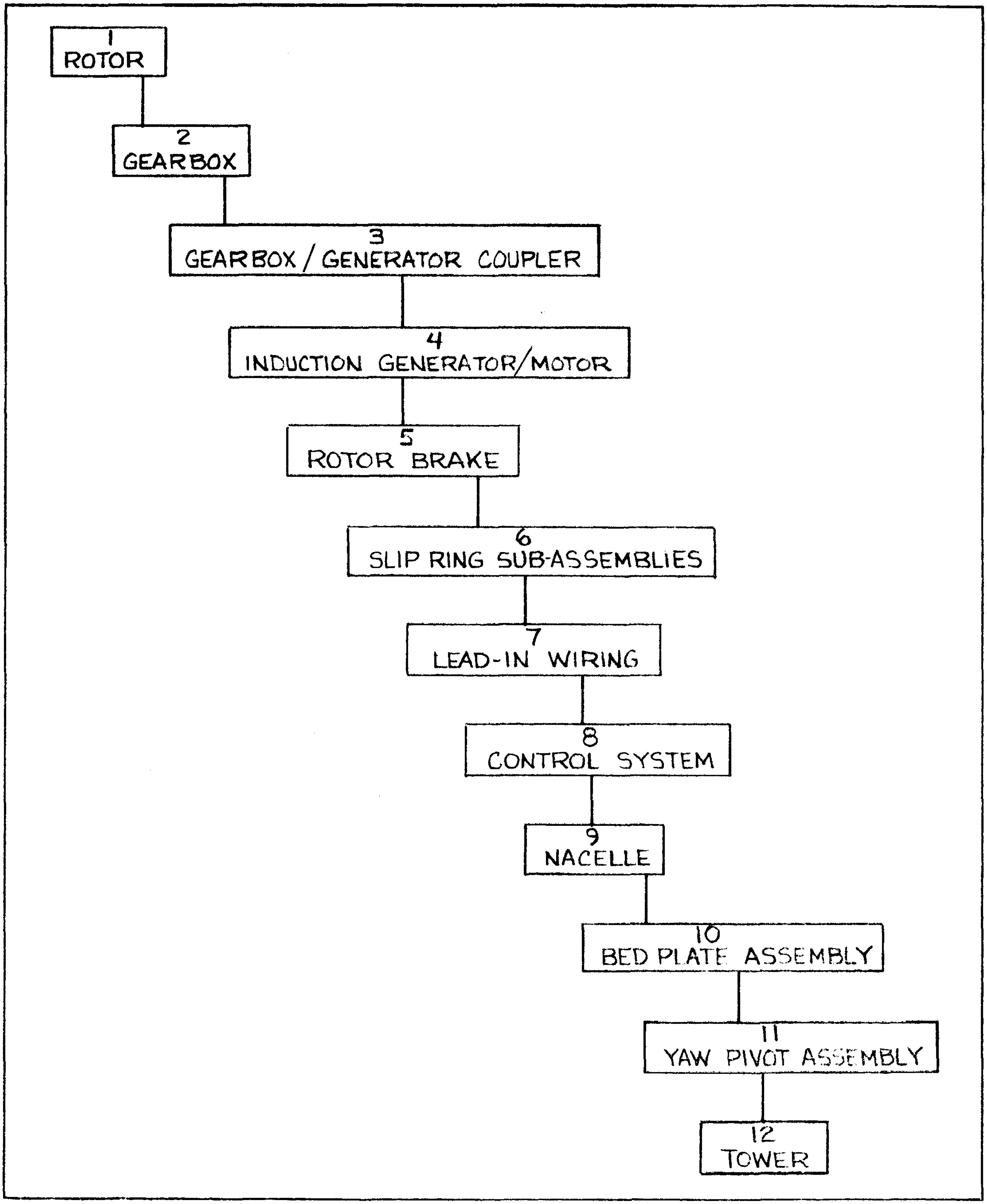


Figure 71

System Reliability Block Diagram

A generic failure, by definition, occurs whenever a component does not perform to its specifications. Not all these "failures", however, would lead to machine failures. A derating factor is used for some components to account for this fact. As an example, the yaw bearing has been derated because 1) some of its generic failures (such as excessive play in the bearing) may not cause the machine to fail and 2) the operating loads on the bearing are below the rated loads. The derating values (for mechanical components) used in the analysis were compared with derating values derived from a standard formula:

$$\text{Life} \quad \alpha \quad \left(\frac{1}{\text{load}}\right)^{10/3} \times \left(\frac{1}{\text{speed}}\right)$$

The values used in the analysis were found to be more conservative than those derived from the formula.

An environmental factor of 3.25, (which falls between the value used for ground vehicles and ship-borne components and the value used for airplanes), is used for all components located on the tower.

6.3.3 Conclusions

The results of the reliability analysis are as follows:

Wind Machine Failure Rate	Mean Time Between Failures
.463/year	2.16 years

The failure rate for each component is shown in Table XX. When these values are used with the maintainability calculations, system availability is estimated to be greater than the 95% required.

Table XX

Failure Rate Estimation

Item	Quantity per Machine	Generic Failure Rate $\#/10^6$ Hours	Derating Factor	Environmental Factor	Failures/Year	Source
1. Rotor						
1.1 Rotor Blade	3	SLD	-	-	-	
1.2 Fastening Subsystem	1	.01	1	3.25	.00028	3
1.3 Cast hub	1	SLD	-	-	-	
1.4 Taper Lock Bushing	1	.045	1	3.25	.00128	1
2. Gearbox						
2.1 Cast Housing	1	.4	.8	3.25	.00912	1
2.2 Helical Gears	4	.002	.8	3.25	.00018	1
2.3 Front Seals	2 (Parallel)	.3	.8	3.25	nil	1
2.4 Back Seals	1	.11	.8	3.25	.00251	1
2.5 Tapered Roller Bearings	3	.02	.8	3.25	.00137	1
2.6 Ball Bearings	3	.02	.8	3.25	.00137	1
2.7 Shafts	3	.15	.8	3.25	.01026	1
3. Gearbox/Generator Coupler	1	.025	1	3.25	.00071	1
4. Induction Generator (20 hp)	1	8.6	.45	3.25	.11025	1, 5
5. Rotor Brake	1	2.1		3.25	.0598	1
6. Slip Rings	9			3.25		4
6.1 Electrical Connectors		.035			.00897	1
6.2 Wiring		.03			.00769	2
6.3 Slip Ring Case		SLD			-	
6.4 Electrical Connectors		.035			.0089	1
6.5 Collector Rings		.5			.1282	3
6.6 Brushes		.1			.02564	1

Table XX (Cont'd)

Failure Rate Estimation

Item	Quantity per Machine	Generic Failure Rate $\#/10^6$ Hours	Derating Factor	Environmental Factor	Failures / Year	Source
7. Lead-in Wires	9					
7.1 Electrical Connectors		.035		3.25	.00897	1
7.2 Wires		.03		3.25	.00769	2
7.3 Electrical connectors		.035		1	.00276	1
8. Control System	1	6	1	1	.0526	
8.1 Generator Speed Switch						
8.2 Anemometer						
8.3 Switch						
8.4 Relays						
8.5 Control Module						
8.6 Fuses						
8.7 Other Electronics						
9. Nacelle						
9.1 Nacelle	1	SLD				3
9.2 Fastener Subsystem		.01	1	3.25	.00028	
10. Bed Plate Assembly	1			3.25		
10.1 Fastening Subsystem		.01		3.25	.00028	3
10.2 Steel Structural Components		SLD				
11. Yaw Pivot Assembly						
11.1 Turntable Bearing	1	.65	.5	3.25	.00926	1
11.2 Turntable Seals	2	.13	.5	3.25	.0037	1

Table XX (Cont'd)

Failure Rate Estimation

Item	Quantity per Machine	Generic Failure Rate $\#/10^6$ Hours	Derating Factor	Environmental Factor	Failures/Years	Source
11.3 Bearing Fasteners		.01	1	3.25	.00028	1
11.4 Steel Components	SLD					
11.5 Leveling System	3	.01	1	3.25	.00085	
12. Tower						
12.1 Tower Foundation		SLD				
12.2 Tower		SLD				
12.3 Fastening Subsystem		.01		3.25	.00028	3

Estimated Failures/Year:	.464
MTBF = 2.16 yr.	
18900 hr.	

Table XX (Cont'd)

Failure Rate Estimation

Sources for Failure Rate Prediction

Sources used for generic failure rate data:

1) Earles, Donald R., and Eddins, Mary F., "Failure Mechanisms," Reliability Engineering Data Series, AVCO Research and Advanced Development Division, April 1962.
Average generic failure rates were used

2) NAVSHIPS 90, Reliability Prediction Handbook

3) Estimated based upon best available field data.

4) Calculated from component failure rate.

The source used for the derating factor was:

5) Von Alven, William H., Reliability Engineering, ARINC Research Corporation, Prentice Hall, Inc., Englewood Cliffs, New Jersey. Pg. 318.

The source used for Environmental factors was:

6) as above by William Von Alven, Pg. 319.

6.4 Maintainability Analysis

6.4.1 Introduction

The maintainability analysis is a two part study to determine the annual maintenance time required on the wind machine and to estimate the mean time to repair (MTTR) each component. These values are used with component failure rate data to determine wind machine availability. The annual maintenance time and the MTTR for each component have been estimated based upon field experience of the Enertech Corporation in the service of wind machines.

6.4.2 Analysis

Table XXI shows a preliminary annual maintenance schedule for the wind machine components. All procedures listed can be performed by one person. Four maintenance procedures are proposed. First, service personnel should examine components that may wear out such as electrical connectors, wires, slip-ring brushes, and pump seals, (the time necessary for these repairs would then be a function of active repair time.) Second, paint and coatings should be repaired so that steel components and blades retain their design characteristics (corrosion may weaken the steel; roughened blades may alter aerodynamic characteristics). Third, yaw bearings should be greased and transmission bearings oiled. Fourth, necessary tolerances should be verified and corrected if necessary (i.e. the brake).

The MTTR for each component is shown in Table XIX of the Availability Analysis (Section 6.1). Repair times were estimated based upon intrinsic availability and on site specific availiability. The intrinsic or ideal repair time is an estimate of active repair time. The site specific repair time includes a week for discovering the failure, for obtaining replacement components, and for scheduling service personnel. Except when spare components are available at the wind site or when failures are discovered during maintenance, a week was added to the ideal repair time for each component in the site specific analysis.

Table XXI

Maintainability Analysis

Item & Maintenance Action	Parts/Items Required	Cost of Materials	Person Hours	Labor Cost (dollars)	Total Cost (dollars)	Down Time (hours)
1. Rotor						
1.1 Inspect blades; sand and use primer paint as needed	Sand Paper, Marine Primer Paint Brush	5.00	1	15.00	20.00	1
1.2 Fasteners - tighten; replace if needed	Set of Wrenches		.5	7.50	7.50	.5
1.3 Inspect casting; sand rust, paint	80 grit sand paper; paint; paint brush	25.00*	.5	7.50	32.50	.5
1.4 Inspect taper lock for firmness; tighten fasteners	Allen Wrench		.25	3.75	3.75	.25
2. Gearbox - Change the oil	Gear oil; bucket; Allen Wrench	15.00	1	15.00	30.00	1
3. Gearbox coupling - check for firm connection; tighten fasteners; align; check and correct for rust	Wrenches; Paint Equipment		.5	7.50	7.50	.5
4. Induction generator/ motor-test			.25	3.75	3.75	.25
5. Rotor Brake - check brake performance; check fluid level; check pressure			.5	7.50	7.50	.5

Table XXI (Cont'd)

Maintainability Analysis

Item & Maintenance Action	Parts/Items Required	Cost of Materials	Person Hours	Labor Cost (dollars)	Total Cost (dollars)	Down Time (hours)
6. Slip Rings						
6.1 Electrical connectors - check firmness; tighten	Wrenches		.25	3.75	3.75	.25
6.2 Wiring - check for proper connections; check insulation			.25	3.75	3.75	.25
6.4 Electrical connectors- as above			.25	3.75	3.75	.25
6.5 Collector rings - inspect for wear; sand with crocus cloth as needed	Crocus cloth	3.00	.5	7.50	10.50	.5
6.6 Brushes - check for alignment and wear	Wrench		1	15.00	15.00	1
7. Electrical Wires						
7.1 Connectors	Wrenches		.5	7.50	7.50	.5
7.2 Wires - inspect full length until ground			.5	7.50	7.50	.5
7.3 Connectors			.5	7.50	7.50	.5
8. Control box/ Control System	Voltmeter, Ohmmeter		1	15.00	15.00	1

Table XXI (Cont'd)

Maintainability Analysis

Item & Maintenance Action	Parts/Items Required	Cost of Materials	Person Hours	Labor Cost (dollars)	Total Cost (dollars)	Down Time (hours)
9. Nacelle						
9.1 Inspect nacelle repair fiberglass as necessary	Fiberglass Repair Kit	15.00	.25	3.75	18.75	.25
9.2 Fasteners - tighten as necessary	Wrenches		.25	3.75	3.75	.25
10. Bed Plate						
10.1 Fasteners - check and tighten	Wrenches		.25	3.75	3.75	.25
10.2 Steel components - inspect; sand rust and paint as needed	Equipment as for hub casting (1.3)		.5	7.50	7.50	.5
11. Yaw Pivot Assembly						
11.1 Turntable bearing- Check yaw motion, lubricate	Grease and Grease Gun	3.50	.5	7.50	11.00	.5
11.2 Bearing Seals. Check for condition and Seating of Seal			.25	3.75	3.75	.25
11.3 Bearing Fasteners - Tighten All Fasteners as Necessary			.25	3.75	3.75	.25
11.4 Steel components inspect for rust, sand and paint as necessary	Equipment as per 1.3		.75	11.25	11.25	.75
11.5 Fasteners - check for firm connection; tighten as needed			.25	3.75	3.75	.25

Table XXI (Cont'd)

Maintainability Analysis

Item & Maintenance Action	Parts/Items Required	Cost of Materials	Person Hours	Labor Cost (dollars)	Total Cost (dollars)	Down Time (hours)
12. Tower						
12.2 Tower - check for rust, etc.			.75	11.25	11.25	.75
12.3 Check fasteners for firm connection			.75	11.25	11.25	.75
TOTALS		66.50	14.0	210.00	276.50	14.0

*Cost of painting materials for all steel components

For high wind machine availability, a spare parts kit should be available at the wind site. For the analysis it was assumed that a kit was available that included electrical connectors, fasteners, fuses, wires, slip-ring brushes, pump seals and an anemometer.

6.4.3 Conclusions

The results of the annual maintenance analysis are as follows:

Person hours required	Cost of annual maintenance
T = 14 hours	C = $276

Because 14 hours of annual maintenance are required, minimum machine down time would occur if maintenance occurred biannually or if two people performed the necessary maintenance on one day.

The results of the MTTR study are as follows:

Intrinsic annual repair time	Site specific repair time
= 2.4 hours	= 71 hours

These analyses do not include time for a major machine overhaul. As specifications on each component become understood, the need for major service during the life of the wind machine may be determined.

Until actual service data and field operation failure rates are known, confidence levels for the numerical values in this analysis are low. However, as a design tool, the analysis has 1) outlined components which should be included in a spare parts kit, 2) defined areas that should be inspected for wear or for tolerance changes, 3) specified routine service on components to maintain design characteristics.

6.5 Safety Analysis

6.5.1 Installation Safety Considerations

Safety has been granted fundamental consideration in the design process of the 15 kW wind machine. The tower and wind machine should be installed by experienced and trained personnel using these guidelines:

1. The crew chief should read the Manual thoroughly and understand the installation procedure before beginning the job.

2. The installation crew should be informed of the extent of the job and safety precautions to be taken.

3. The tower should be inspected per manufacturer's recommendations to insure proper installation before mounting the wind machine.

4. Installation should only be done on a calm, dry day with no possibility of a storm.

5. Installation instructions should be followed in the correct order.

6. All hoisting and safety equipment should be checked before use.

7. The tower crew must wear safety climbing belts; all crew members must wear hard hats. Optional safety climbing devices (belts, restraints, etc.) can be obtained for tower installation to meet additional specified requirements.

8. All personnel should stay clear from under the components being hoisted.

9. The owner of the system and all personnel involved in the installation should be adequately insured.

10. A fence (8-foot high suggested) should be installed around the perimeter of the tower base to deter unauthorized persons from climbing the tower. (An anti-climb shroud may be used instead.)

11. A warning notice should be attached to each tower leg and in conspicuous locations on the fence noting danger to unauthorized personnel climbing the tower. (These signs can be purchased from the tower manufacturer.)

12. No tower installation should be attempted within tower falling distance of buildings.

13. Installation should not be attempted within tower falling distance of buildings.

The installation of the wind machine on a tower can be dangerous if instructions are not followed or safety precautions are not taken. Common sense must be used at all times.

The following specific safety guidelines should also be followed:

1. Hoisting personnel should insure each component is properly rigged according to manual and diagrams.
2. Each component to be hoisted should be securely fastened either permanently or temporarily before beginning the hoisting of another component.
3. Yaw brake bolts must always be applied after nacelle is moved during installation procedure and anytime while people remain on tower.
4. During the time when yaw position is being changed, no personnel should be positioned high enough on tower to be hit by moving assembly.
5. Rotor assembly should be locked at all times when personnel are on the tower.

6.5.2 Operational Safety

The safety of machine and personnel is maintained through appropriate electrical design of shutdown functions. In the event of normal shutdown system failure, rotor speed is limited through the use of blade tip brakes.

The overspeed shutdown function senses potentially damaging excessive rotor speed and institutes unit shutdown. A frequency sensing shutdown function guards against the remote possibility of power feedback to the utility during a power outage.

A vibration switch initiates shutdown, should potentially damaging vibration occur due to ice loadings or other unforeseen conditions.

The six electrical safety features listed below reduce electrical danger to the customer and maintenance personnel. Provisions of the National Electrical Code (NEC) are adhered to.

1. Adhering to NEC motor/generator minimum wire size recommendations prevents the tripping of circuit breakers during high wind conditions.

2. Appropriate size circuit breakers protect wires from overheating.

3. Generator overcurrent protection sized per NEC recommendations prevents winding damage due to excess current.

4. Appropriate motor/generator grounding minimizes shock hazard to personnel. A full size separate ground wire and slip-ring are specified.

5. A locking safety switch is specified for safety to service personnel. This allows the machine to be serviced without the possibility of an inadvertant startup with personnel on the tower.

6. Lightning protection is specified at the generator, contactor and control box. The presence of the wind machine on a tower increases the probability of a lightning strike. The lightning protection specified prevents damage to the customer's electrical service (direct or nearby strikes) and prevents generator damage (nearby strikes only).

6.5.3 Other Safety Considerations

<u>Installation</u> Systems should not be installed on buildings and structures other than specified towers, even if high enough to obtain higher wind speeds and lower turbulence, at least until such time as the machine is proven in the field.

The installation plan and procedure was developed using a full scale model of the tower top section and machine in order to plan the movement and placement of persons working atop the tower. Using basic safe operating methods there should be no danger to personnel associated with the task.

Operation There are no safety problems associated with unattended operation of the wind plant. Icing of the blades can occur; however, any unbalanced condition will cause triggering of the vibration switch and unit shutdown. An icing condition may create safety problems due to the shedding of ice during braking and/or start-up. Loose ice on the leading edge of the blades is likely to fall in an area just below the machine. Anti-icing compounds that will prevent the build-up of ice are being researched.

Though the machine has been designed to withstand winds of 56 m/s (125 mph) it may be advisable to remove the blades when a severe hurricane is forecast. Although the expense may be considerable, it could prevent damage to the machine from flying debris, and prevent structural damage due to winds in excess of design specifications.

Maintenance A conscious design effort has been made to make accessible those parts of the system that could need inspection or service. In order to ensure proper functioning, inspections are recommended as follows:

1. Thirty days after installation
2. Every six months
3. After electrical storms and when winds have exceeded 60 mph.

Inspection procedures are short and simple. The control panel is opened and lightning arrestors and wiring are checked. The machine is turned on TEST and observed for abnormal behaviour or noises. The machine should then be checked for loose wires and fasteners.

6.6 Design Environmental Factors

Concern was expressed at FDR by Rockwell International personnel regarding the ability of the control system to withstand the severe environmental conditions imposed by the design specifications. It was originally intended that the control box itself be installed in-doors; however, the control system can be installed and operated outdoors if necessary. Protection from the elements would be accomplished through the use of a weatherproof enclosure such as the Hoffman A-42H30BLP. Adequate temperature for accurate electronics operation can be assured by the use of thermostatic strip heaters such as those manufactured by Midwest Components or Murata Corporation.

Therefore, even though the control box may not normally be installed in a heated indoor environment, the strip heaters specified will be sufficient to maintain control function stability.

6.7 10,000 Unit Per Year Concept

The production of 10,000 units per year will facilitate significant reductions in the cost of 15 kW SWECS. The basic design concept for the 10,000 unit per year model is to combine the frame assembly and gearbox into a single integral casting, eliminating both the need for frame welding and purchasing of complete gearboxes (see Figure 72). In addition such a design would completely eliminate the need for a full-closure nacelle (unless desired by the customer). The generator would be protected by a small hinged fiberglass shield. By directly bolting the motor to the gearbox casting, the front motor bearing and bearing mounting could be eliminated. The gearbox would be serviced through a quickly removable top plate which provides easy access to internal gears and bearings.

A high production volume 15 kW SWECS would justify the use of more sophisticated high volume hub casting equipment than the prototype, thus reducing costs. A one-piece hub would be used once a suitable casting procedure was developed.

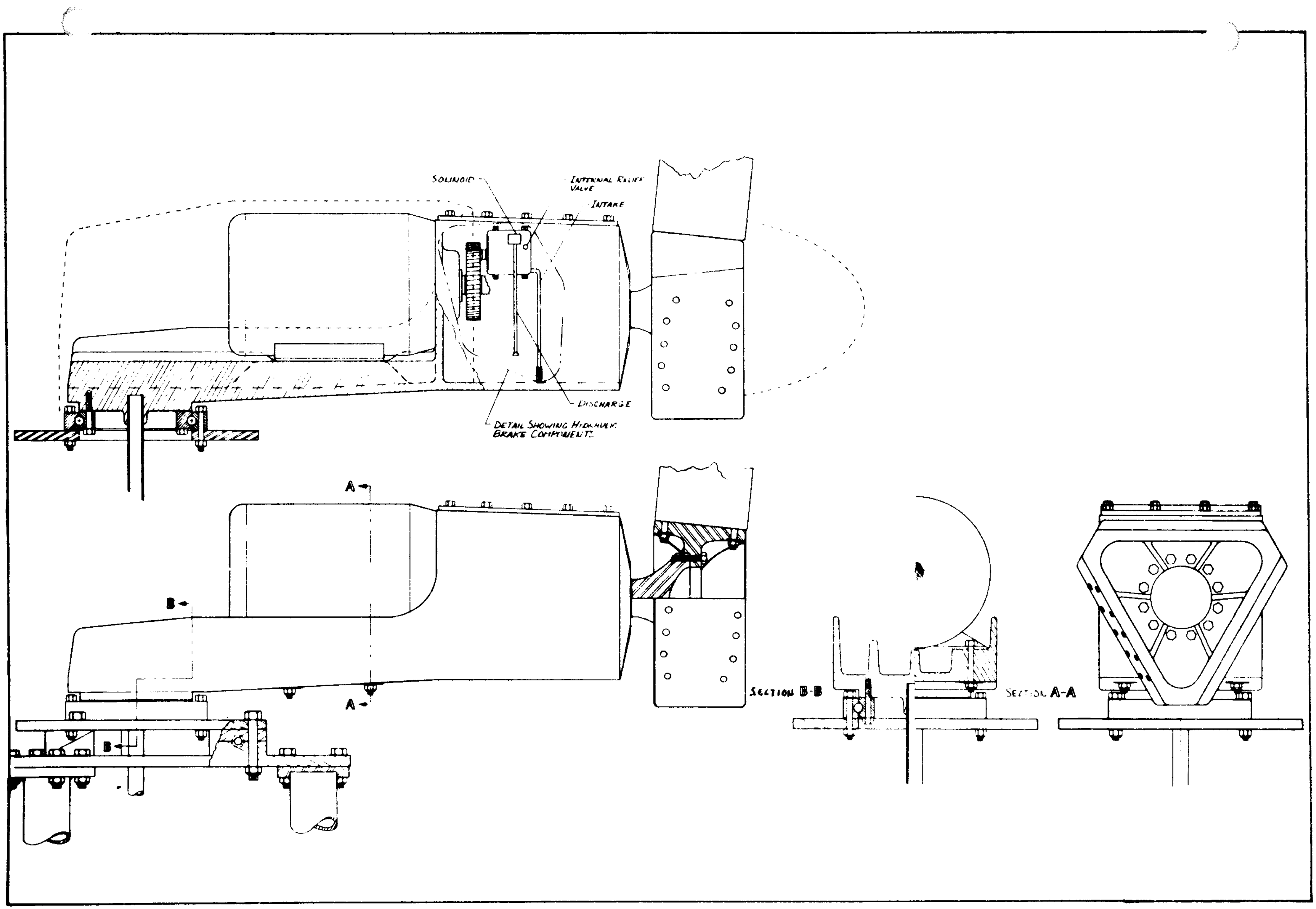


Figure 72

10,000 Unit Machine Concept

The tower for the 15 kW production model would be specially designed to reduce material usage while retaining the necessary strength and rigidity. By providing a wider non-standard base, significant material reductions could be realized. This would result in a tower configuration such as that shown in Figure 73.

The 15 kW production tower-top would consist of a one-piece casting with integral stationary leveling devices. This production method would eliminate practically all machining and/or welding associated with the item.

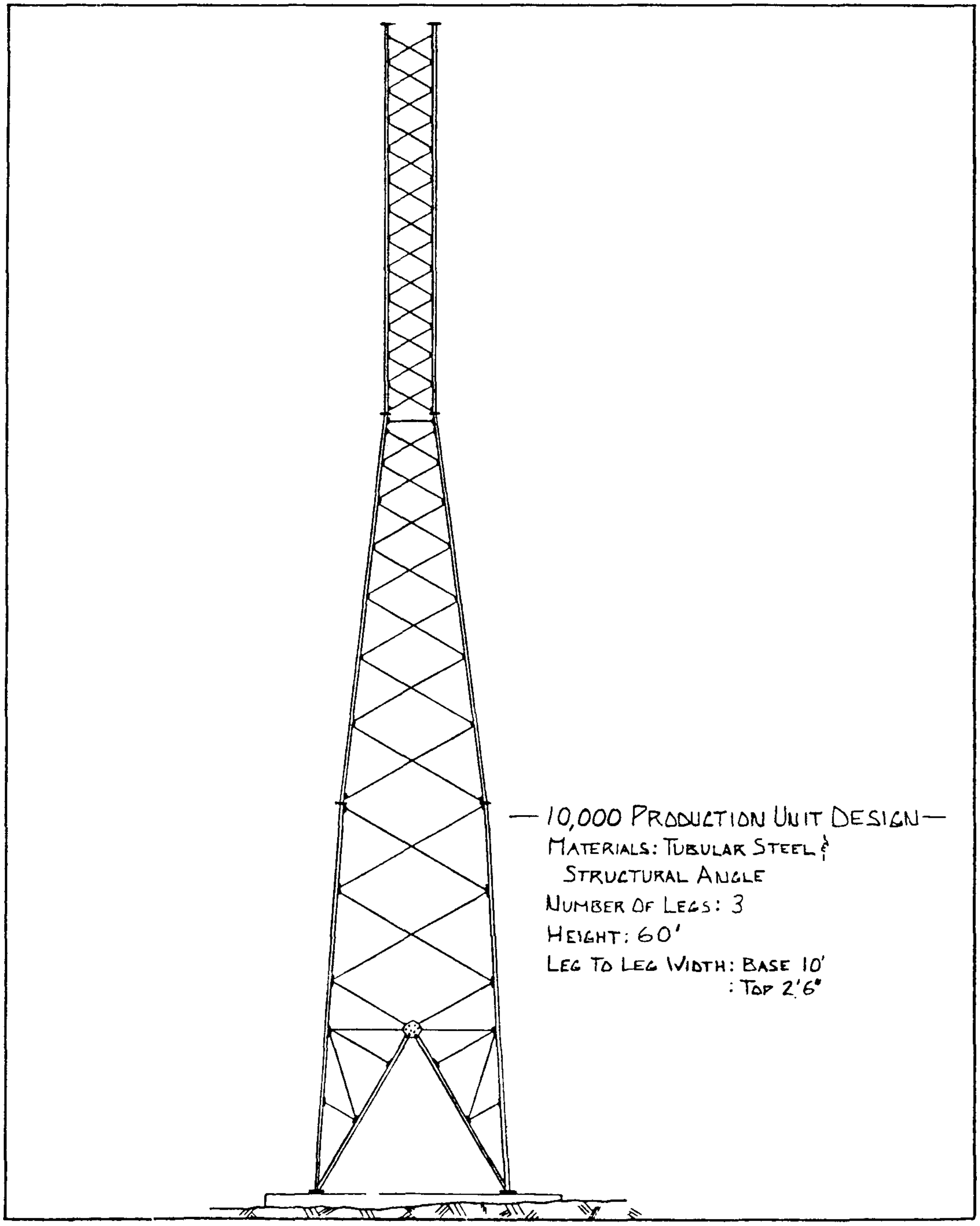


Figure 73

10,000 Unit Tower Concept

In areas where access is not a problem, lowest installation cost would normally result through the rental of a crane for a few hours to first raise the assembled tower and then place the machine on top. Where this is not possible, the safest hand tool installation procedure would involve the building of the tower up from the ground and the sectional installation of the unit on the tower.

The 10,000 unit 15 kW SWECS generator would likely remain a purchased item unless significant quantities of single phase generators could not be procured at acceptable cost. In that case the frames would be purchased and wound in-house.

The production unit hydraulic brake would contain no plumbing and would consist entirely of manifold-valve assemblies thereby reducing brake cost and increasing reliability.

The production control system would consist of optimized integrated/printed circuitry entirely mounted in the nacelle. Thus the only control gear on the ground would be an electrical disconnect switch.

10,000 per year production volume blades would be manufactured in the same manner as the prototype blades, except that highly automated techniques would be employed. The blade subcontractor has indicated that blade production in these volumes would necessitate the building of a new plant utilizing a modular approach to blade construction.

Labor, plant, and equipment requirements are itemized in Tables XXII - XXVI. A drawing of a possible 10,000 unit per year manufacturing facility is provided in Figure 74.

Table XXII

Manufacturing Requirements and Cost - 10,000 Unit Per Year Concept

Manufacturing Space Required	200,000 ft^2
Manpower	640
Hours per Machine	120
Building Cost	$4.0 Million
Machinery Cost	$7.2 Million
Man-Hours Cost per Machine @ $14.00/hr	$1680
Material Cost per Machine	$6856
	$8536
Depreciation - Ten Years per Machine	$ 112
G and A @ 20% per Machine	$1730
Profit @ 18% per Machine	$1556
Total Cost per Machine	$11,934

Table XXIII

Labor Requirements - 10,000 Unit Per Year Concept

Build Blades - Track Pitch Balance		29 Hrs
Assemble Gear Box		3
Assemble Control Box		20
Assemble Brush and Yaw Assembly		1
Assemble Motor and Brake		4
Inspections		2
Crate and Ship		2
Paint and Deburr		2
Accessories and Fiberglass		1
Stockroom Kit Prep		1
Casting		6
Machine Shop -	Gearbox	8
	Hub	7
	Gears-Shafts	1
	Frame	8
Tower Welding Fabrication		25
Total		120

Table XXIV

Manpower Required - 10,000 Unit Per Year Concept

Blades	145	
Assemble Gearbox	15	
Control Box	100	
Brush & Yaw	5	
Motor & Brake	20	
Inspection	10	
Crate Ship & Rec	10	
Fiberglass & Access	5	
Paint	10	
Stockroom Kits	5	
Machine Shop	110	
Casting	30	
Tower	125	
	590	Workers
	30	Supervision
	10	Prod Control
	5	Engineering
	10	Purchase
	645	
Use	650	

Table XXV

Tooling Requirements - 10,000 Unit Per Year Concept

Patterns	Hub
	Main Frame
	Blades
Jigs	Blade Studs
	Gearbox Top
	Turn Table
	Hub Bolts
Fixtures	Main Frame
	Hubs
	Assembly Jigs
	Track & Pitch
	Paint Room Fixtures
	Hyd. Mule

Table XXVI

Manufacturing Space Requirements - 10,000 Unit Per Year Concept

Area		Thousand Ft^2
Blades		70
Casting		15
Mach Shop		12
Stockroom		8
Ship & Rec		4
Track Pitch		10
Paint		5
Inspection		5
Mach Assem		12
Control Box		2
Crate		5
Tower		52
	Total	195

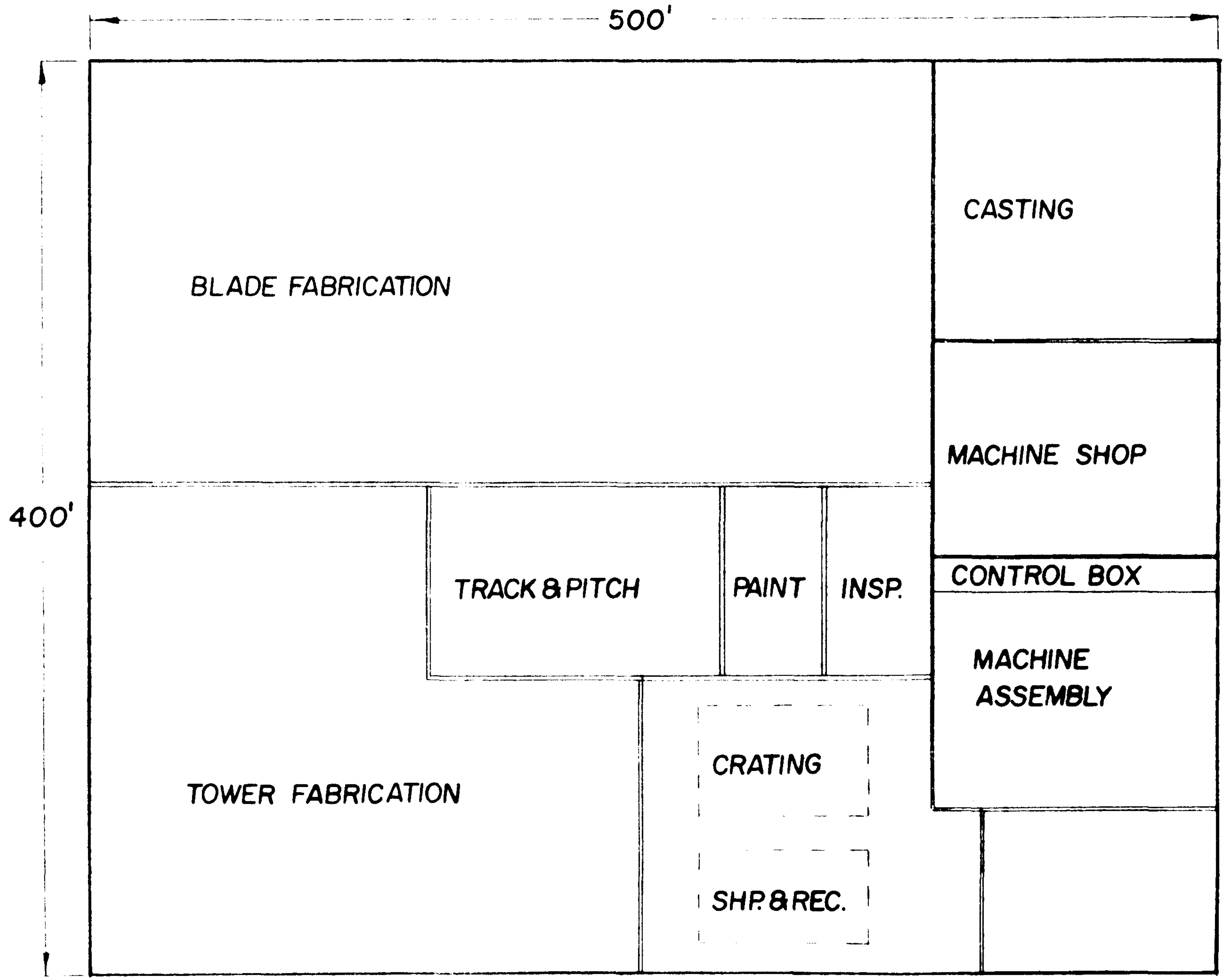


Figure 74

Plant Layout - 10,000 Unit Per Year Production

6.8 Cost Analysis

System cost estimates for both the standard and the special designs are made based on first, 100th, 1000th, and 10,000th production units (see Tables XXVII - XXX). A learning curve of 95% is used for estimating the labor requirements. The material costs are based on either manufacturer quotations or on a dollars per pound basis, obtained from the "1500" machine or other inquiries. In production quantities the estimates are made either by applying the original equipment manufacturer (OEM) multipliers supplied by the manufacturer or by making best estimates based on similar products. For example, the control system cost can be reduced considerably in production quantities, whereas the cost of generators does not reduce at the same rate. This is because the generators are already made in a large quantity and the production methods do not lend themselves to further improvement. On the other hand, the first control system is custom-made and its cost can be reduced considerably in production quantities.

A labor rate of $7/hr. is applied together with a 100% labor overhead. The FOB cost is estimated by using a 20% general and administrative (G & A) cost and a fee of 18% on the total cost. These figures are based on experience gained in the sales of Enertech 1500's and the establishment of a nationwide dealer network. Dealer's commission is 33% of the FOB cost and transportation cost is 7% of the FOB cost. Installed cost is 1.4 x FOB cost + installation cost.

All these cost estimates are based on 1980 dollar figures. In calculating the cost of energy (as specified in the contract), 1978 dollar figures are obtained by using an annual inflation rate of 13% for 1979 and 7.5% for mid-1980.

At FDR concern was expressed by Rockwell International personnel over the realism of the cost of energy (COE) calculations. It was suggested at that time that additional detail be provided and potential improvement areas highlighted. As a result the following information is provided. It must be remembered, however, that this analysis applies only to a mature (10,000 unit per year) configuration, as this is the one to which the COE applies. The analysis will follow the format of the cost summary sheet and will be performed only for the "Standard" design.

Table XXVII

15 kW SWECS Cost/Weight Summary - First Unit

I WIND TURBINE GENERATOR	WEIGHT (lbs)		COST (mhr/$)					
			LABOR (MHRS)				MATERIAL ($)	
	Std.	Special	Std.		Special		Std	Special
Mechanical - Rotor			Hrs	$	Hrs	$		
Blade(s) / Tip Brakes	600	650	16	224	16	224	2800	3000
Hub(s)	420	420	4	56	4	56	754	854
Transmission	-	-						
Bearing(s), Main	35	35	2	28	2	28	575	575
Gearbox w Couplings	690	690	4	56	4	56	2000	2100
Support Struct.	100	100	8	112	8	112	96	96
Shaft(s)	-	-						
Frame	460	460	8	112	8	112	524	524
Strut(s)	-	-						
Nacelle	125	125	4	56	4	56	325	325
Controls	100	100	40	560	40	560	1000	1000
Subtotal (lbs. mhrs. $)	2530	2580	86	1204	86	1204	8074	8474
Electrical - Generator /Brake	335	335	8	112	8	112	2800	2800
Pwr. Condition.								
Slip rings	15	15	8	112	8	112	230	230
Cables	100	100					340	340
Subtotal (lbs. mhrs $)	450	450	16	224	16	224	3370	3370
WTG Subtotal	2950	3000	102	1428	102	1428	11444	11844
II TOWER/AUX. EQUIP - Tower	3314	4518					3700	5180
Guyes/rigging	-	-	-		-		-	-
Subtotal	3314	4518					3700	5180
WGT. Total (WTG & TOWER/AUX.)	6264	7518						
WTG & Tower/Aux - Subtotal ($)			102	1428	102	1428	15144	17024
G&A (20%)			3314		3690			
Fee (18%)			2983		3321			
SWECS TOTAL			22869		25463			
Labor & 100% overhead = $14/hr								

Table XXVIII

15 kW SWECS Cost/Weight Summary - 100th Unit

	WEIGHT (lbs)		COST (mhr/$)					
			LABOR (MHRS)				MATERIAL ($)	
I WIND TURBINE GENERATOR	Std.	Special	Std.		Special		Std	Special
Mechanical - Rotor			Hrs	$	Hrs	$		
Blade(s)	550	600	11	154	11	154	2450	2650
Hub(s)	420	420	3	42	3	42	620	720
Transmission	-	-						
Bearing(s), Main	35	35	1	14	1	14	517	517
Gearbox w Couplings	690	690	3	42	3	42	1385	1485
Support Struct.	100	100	6	84	6	84	90	90
Shaft(s)								
Frame	460	460	6	84	6	84	472	472
Strut(s)								
Nacelle	125	125	3	42	3	42	293	293
Controls	100	100	28	392	28	392	700	700
Subtotal (lbs. mhrs. $)	2480	2530	61	854	61	854	6527	6927
Electrical - Generator	335	335	6	84	6	84	2200	2200
Pwr. Condition.								
Slip rings	15	15	6	84	6	84	175	175
Cables	100	100					306	306
Subtotal (lbs. mhrs $)	450	450	12	168	12	168	2681	2681
WTG Subtotal	2930	2980	73	1022	73	1022	9208	9608
II TOWER/AUX. EQUIP - Tower	3314	4518					3700	5180
Guyes/rigging							115	140
Subtotal	3314	4518					3815	5320
WGT. Total (WTG & TOWER/AUX.)	6244	7498						
WTG & Tower/Aux - Subtotal ($)			73	1022	73	1022	13023	14928
G&A (20%)			2809		3190			
Fee (18%)			2528		2871			
SWECS TOTAL			19382		22011			
Labor & 100% overhead = $14/hr								

Table XXIX

15 kW SWECS Cost/Weight Summary - 1000th Unit

I WIND TURBINE GENERATOR	WEIGHT (lbs) Std.	WEIGHT (lbs) Special	COST (mhr/$) LABOR (MHRS) Std. Hrs	Std. $	LABOR (MHRS) Special Hrs	Special $	MATERIAL ($) Std	MATERIAL ($) Special
Mechanical - Rotor								
Blade(s)	500	550	10	140	10	140	2200	2400
Hub(s)	350	350	10	28	2	28	490	590
Transmission								
Bearing(s), Main	35	35	1	14	1	14	489	489
Gearbox w Couplings	690	690	2	28	2	28	1225	1325
Support Struct.	100	100	5	70	5	70	85	85
Shaft(s)								
Frame	400	400	5	70	5	70	445	445
Strut(s)								
Nacelle	125	125	2	28	2	28	276	276
Controls	80	80	24	336	24	336	560	560
Subtotal (lbs. mhrs. $)	2280	2330	51	714	51	714	5770	6170
Electrical - Generator	335	335	5	70	5	70	1600	1600
Pwr. Condition.								
Slip rings	15	15	5	70	5	70	150	150
Cables	100	100					276	276
Subtotal (lbs. mhrs $)	450	450	10	140	10	140	2026	2026
WTG Subtotal	2730	2780	61	854	61	854	7796	8196
II TOWER/AUX. EQUIP - Tower	3314	4518					3200	4480
Guyes/rigging								
Subtotal	3314	4518					3200	4480
WGT. Total (WTG & TOWER/AUX.)	6044	7298						
WTG & Tower/Aux - Subtotal ($)			61	854	61	854	10996	12676
G&A (20%)				2370		2706		
Fee (18%)				2133		2435		
SWECS TOTAL				16353		18671		
Labor & 100% overhead = $14/hr								

Table XXX

15 kW SWECS Cost/Weight Summary - 10,000th Unit

I WIND TURBINE GENERATOR	WEIGHT (lbs) Std.	WEIGHT (lbs) Special	COST (mhr/$) LABOR (MHRS) Std. Hrs	LABOR Std. $	LABOR Special Hrs	LABOR Special $	MATERIAL ($) Std.	MATERIAL ($) Special
Mechanical - Rotor								
Blade(s)	480	530	8	112	8	112	1684	1884
Hub(s)	310	310	7	98	7	98	217	317
Transmission								
Bearing(s), Main	35	35	1	14	1	14	431	431
Gearbox w Couplings	496	496	16	224	16	224	507	607
Support Struct.	80	80	4	56	4	56	56	56
Shaft(s)								
Frame	330	330	8	112	8	112	213	213
Strut(s)								
Nacelle	10	10	2	28	2	28	50	50
Controls	50	50	20	280	20	280	400	400
Subtotal (lbs. mhrs. $)	1791	1841	66	924	66	924	3558	3958
Electrical - Generator /Brake	280	280	4	56	4	56	1200	1200
Pwr. Condition.								
Slip rings	15	15	4	56	4	56	140	140
Cables	80	80					250	250
Subtotal (lbs. mhrs $)	375	375	8	112	8	112	1590	1590
WTG Subtotal	2260	2300	74	1036	74	1036	5148	5548
II TOWER/AUX. EQUIP - Tower	2651	3614					2464	3584
Guyes/rigging								
Subtotal	2651	3614					2464	3584
WGT. Total (WTG & TOWER/AUX.)	4911	5914						
WTG & Tower/Aux - Subtotal ($)			74	1036	74	1036	7612	9132
G&A (20%)			1730		2034			
Fee (18%)			1557		1830			
SWECS TOTAL			11,934		14031			
Labor & 100% overhead = $14/hr								

6.8.1 Hardware Costs

Rotor Blades Gougeon Bros, Bay City, Michigan, prepared an estimate for the production of 10,000 blade sets per year. This estimate included a 12% fee (profit). Because Enertech would manufacture the blade in-house this profit would not be applicable as a before G&A cost. Therefore, the estimate used is less the 12% (or $1584). In addition, tip brakes must be included in the blade cost. These will cost approximately $30 per tip brake in materials and will require two hours each to fabricate and fit. This figure appears as the 8 hours on the cost summary sheet. Thus, total rotor cost is $1684.

Hub It is estimated that a mature hub can weigh as little as 310 lb. Estimates from Joy Mfg. Company, Claremont, New Hampshire, show that in large quantities, cost per pound of castings would be $.70. Therefore, the hub material cost is estimated at $217. It is further estimated that 7 hours of time would be needed for machining, fitting, painting, etc.

Main Bearing The Kaydon turntable bearing used in both the prototype and mature design is estimated to cost $401 in large quantities - from Kaydon quotes. Labor cost is estimated at 1 hour to unpack, fit, and grease.

Gearbox Because the gearbox and frame are a single piece in the mature design, the cost to produce the mature design version was first figured on a combined basis and then separated into components to be compatible with the cost summary sheet. To figure material costs the $.70/lb casting estimate was used as it was with the hub. Estimated weights for the frame are 403 lb and 496 lb for the gearbox (total 800). Thus the material cost for the frame is $213 and $507 for the gearbox ($300 for shifting gears and bearings). Estimated machining time is 16 hours for the gearbox and 8 hours for the frame.

Support Structure The tower top weight will be 80 lb and thus will cost $56 for the casting. Machining time would be 4 hours maximum.

Nacelle The proposed nacelle for the mature design would consist only of a small fiberglass generator cover at a cost of $50. Hinge fitting would require a maximum of 2 hours labor.

Control System The production of 10,000 control systems per year would allow the use of integrated circuitry. The cost of components is estimated to be $400 while 20 hours would be allowed for control system assembly.

Generator/Brake Mass produced generators, less the front bearing assembly, are estimated to cost $900 (quotes from manufacturers). A mature hydraulic brake is estimated to cost $300 ($150 for the pump, $80 for the solenoid, $20 for pressure relief valve, and $50 for mounting and piping). Assembly time for the two units is estimated at 4 hours.

Slip-Rings In large quantities slip-rings would be assembled in-house from purchased components. Material costs are estimated at $140 with 4 hours required to assemble, mount on machine, and connect wires to machine. The material cost for the wires to connect the rings to the machine is included in these costs.

Cable Quantity discounts should reduce the costs of cables to $250 per set (for 60-ft tower).

Tower Quotes from Rohn for the prototype 60-ft tower indicate that the tower could be purchased from them in large quantities for $1300 each. By the optimization of tower members and the widening of the base it is estimated that this figure could be reduced by 20%. Thus tower cost is estimated to be $2464 if built in-house. Actually this might be reduced further due to the fact that it still includes Rohn's overhead and fee.

As presented on the cost summary sheets (Table XXVII) the above costs total $11,934.

6.8.2 Installation Costs

For the prototype it was estimated that five men would be required for two days. Using a 95% learning curve for each of 50 dealers across country who would each install 200 machines per year, this cost will reduce to $756 at 10,000 units per year.

The foundation cost was calculated as follows:

Excavate for 33 yd concrete	3 hr @ $50/hr	$ 150
Rebar 1000 lb	@ $.30/lb	300
Labor 3 people for	3.5 hr @ $14/hr	150
Concrete 33 yd	@ $40/yd	1320
TOTAL		$1920

Learning curve factors would reduce this approximately $100.
Total = $1820.

Total Installation cost is foundation plus installation.

$1820 + $756 = $2576

As previously explained the FOB cost is multiplied by 1.4 to provide 7% shipping and a 33% dealer margin. These figures are in keeping with past Enertech procedures. Thus the total cost is 1.4 (11934) + $2576 + $19288. This figure is then adjusted back to 1978 dollars and yields $15875. Applying an annual cost factor of .087 and an annual maintenance plus replacement cost $397 and dividing by the predicted annual energy output of 54608 kWh the unit cost of energy is found to be 3.26¢/kWh for a 5.4 m/s (12 mph) average wind site.

6.8.3 Areas For Potential Cost Reduction

Three areas which show potential for cost improvement are 1) Foundation for tower, 2) tower and, 3) control system. The tower foundation as originally specified utilized a large quantity (33 cubic yards) of concrete. However, the installation procedure is simple (a large square hole is required). One method of reducing this concrete requirement is through the use of a pad and pier type of footing. As a result of FDR, Enertech was instructed to investigate other forms of foundations. The results of this investigation appear in Section 4.4.2 (above).

The tower cost is a very significant part of the total system cost (in excess of $2500). It is highly likely that significant strides can be made in tower design which will serve to reduce costs. Composite materials or even wood could be utilized in the development of a "prefab" tower that could be easily assembled in the field. Such an arrangement could significantly reduce tower costs.

The control system--as now designed--utilizes a low voltage starting system to reduce in-rush current. If start-up current does not prove to be a problem it is possible the low voltage circuit could be eliminated, simplifying the control system. In addition 20 hours were allowed for control system assembly. It is possible that through the use of carefully designed and packaged components this time could be greatly reduced.

Concern was expressed at FDR by Rockwell International personnel over the relatively large quantity of concrete (33 yd) specified for the foundation. By spreading the tower base out in the 10,000th design, it becomes possible to reduce this quantity by utilizing three separate piers to support the tower legs. The excavation for the piers becomes more complicated, however, and requires either the full forming of the excavated area or specialized equipment to bore a cylindrical belled hole. The cost differential for the pier and pad foundation vs the single pad foundation is shown in Table XXXI.

Table XXXI

Cost Differential for Pier Pad and Single Block Foundation

Pier Pad		Single Block
Concrete Usage	14.3 yd	33 yds.
Cost @ $40/yd	$512	$1320
Rebar Ft.	1242	510
Cost	$500	$300
Labor hrs	25	10.5
Labor cost @ 14/hr	$350	$150
Excavation hrs	6	3
Excavate @ $50/hr	$300	$150
TOTAL COST	$1662	$1920
C.O.E.	3.22¢/kWh	3.26¢/kWh

The effect of the alternate design on COE is not great. A 3-pier foundation would be justified in these areas where concrete is expensive but not necessarily in all or most installations.

6.8.4 Annual Energy Output Calculations

Given power output vs windspeed data, the annual energy output of the machine can be calculated if the number of hours the machine would operate at each wind speed is known. This can be done using a Raleigh distribution. The probability that the wind speed is equal to or above a particular velocity given by V is,

$$H = 8766 \exp\left[-\frac{\pi}{4}\left(\frac{V}{\bar{V}_z}\right)^2\right] \tag{1}$$

Where, H is the number of hours

$\bar{V}_z$ is mean annual wind speed given by 1/7

Power law as,

$$\bar{V}_z = \bar{V}\left(\frac{h_z}{h}\right)^{1/7} \tag{2}$$

Where, h = 30' (As required in the contract)

h_z = 62' (Hub height)

By differentiating Eq. 1 with respect to V, one can obtain the slope $\left(-\frac{dH}{dV}\right)$ as,

$$-\frac{dH}{dV} = 8766\,\frac{\pi}{2}\frac{V}{\bar{V}_z^2} \exp\left[-\frac{\pi}{4}\left(\frac{V}{\bar{V}_z}\right)^2\right] \tag{3}$$

Using Eq. (2) and (3) and the power vs windspeed data (Table XXXII), the energy output is calculated at annual mean wind speeds of 10, 12, 14, 15, 16, and 18 mph measured at 30 feet (see Table XXXIII, XXXIV and Figure 75).

Table XXXII

Power Output at Various Wind Speeds

V,mph	8	10	12.5	15	17.5	20	22.5	25	27.5	30	32.5	35	37.5	40
P, kW	0	2.5	5.5	8.5	11.6	14.4	17.6	20	22.1	23.5	21.7	17.6	13.2	10.7

Table XXXIII
Energy Output at Various Average Annual Wind Speed
(90% and 95% Availability Factors)

V in mph		10	12	14	15	16	18
Annual Energy in kWh	90% Avail.	41034	54608	68295	73666	78011	83610
	95% Avail.	43314	57641	72088	77758	82345	88255

Table XXXIV

Annual Energy Output

V	P	10 mph		12 mph		14 mph		15 mph		16 mph		18 mph		mph	kW
(mph)	(kW)	dH/dV	E	dH/dV	E	dH/dV	E	dH/dV	E	dH/dV	E	dH/dV	E	V	P
8	0	595	0	468	0	371	0	332	0	298	0	243	0	8	0
10	2.5	591	1478	499	1247	412	1031	375	936	341	852	284	709	10	2.5
12.5	5.5	516	2838	486	2673	429	2360	399	2195	370	2036	317	1745	12.5	5.5
15	8.5	399	3394	430	3655	412	3499	394	3350	374	3181	333	2827	15	8.5
17.5	11.6	277	3217	350	4060	369	4276	365	4235	356	4135	331	3835	17.5	11.6
20	14.9	174	2597	264	3934	310	4625	320	4766	322	4805	314	4681	20	14.9
22.5	17.6	100	1751	185	3264	247	4349	266	4685	278	4900	287	5046	22.5	17.6
25	20	52	1036	122	2435	187	3730	211	4224	230	4601	252	5042	25	20
27.5	22.1	25	545	75	1654	134	2957	160	3538	182	4032	214	4732	27.5	22
30	23.5	11	253	43	1015	91	2148	116	2730	139	3269	176	4136	30	23.5
32.5	21.7	4.3	93	23	508	60	1292	81	1753	102	2215	140	3041	32.5	21.7
35	17.6	1.6	28	12	210	37	651	54	949	72	1270	108	1905	35	17.6
37.5	13.2	0.5	7.0	5.7	76	22	290	35	456	49	669	81	1104	37.5	13.6
40	10.7	0.2	1.8	2.6	28	12	133	21	228	32	347	59	632	40	10.7
Total Energy (kW-hr)		46,217		61,895		78,355		85,114		90,780		98,591			
25% Loss 8-10 mph		462		390		322		293		266		222			
25% Loss 30-40 mph		160		830		2,150		2,970		3,834		5,469			
Net annual		45,594		60,675		75,883		81,851		86,679		92,900			
Annual E at 95% avail. (kW-hr)		43,314		57,641		72,089		77,759		82,345		88,255			

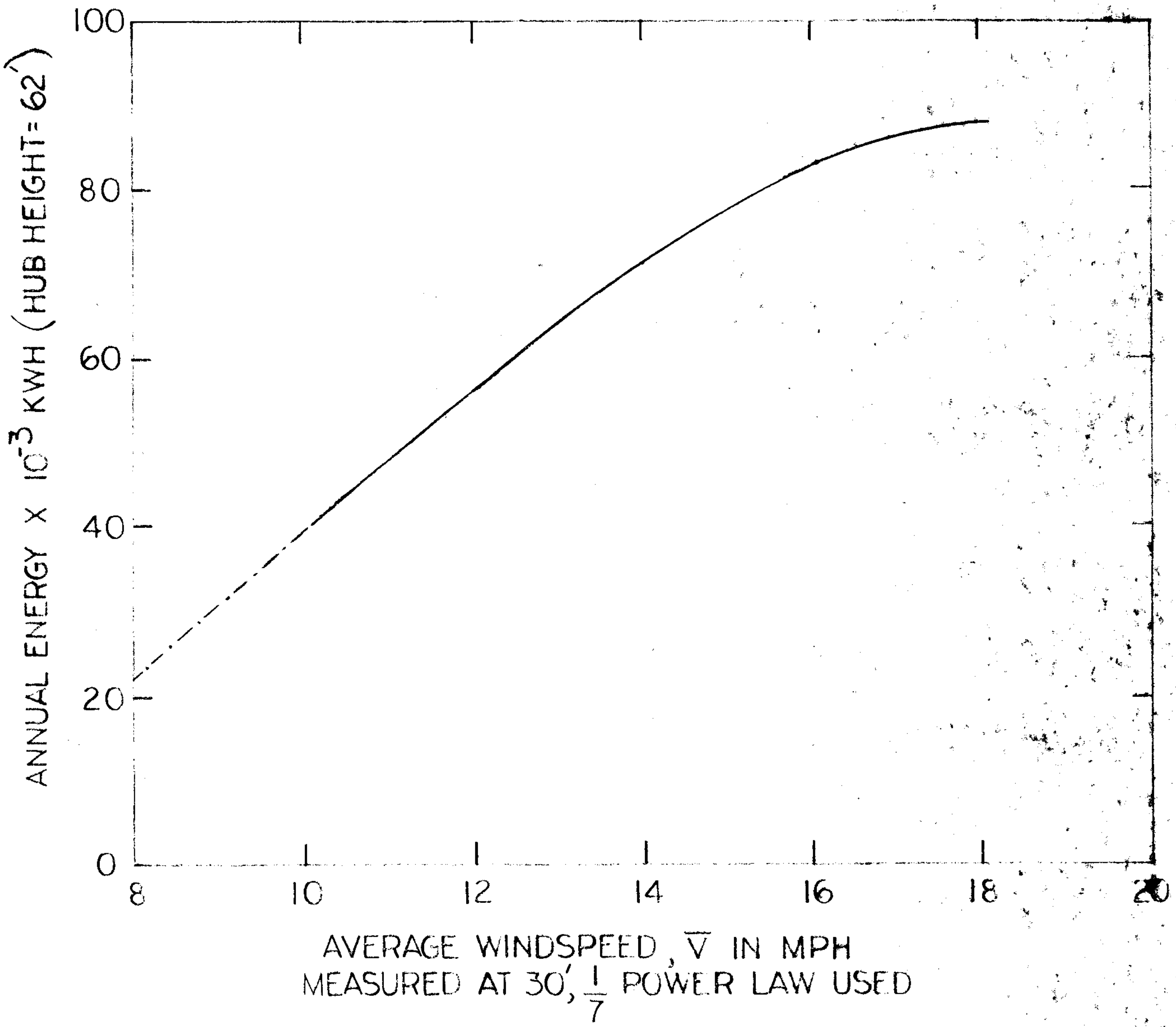


Figure 75

15 kW SWECS Annual Energy Output Curve
(95% Availability)

6.8.5 Cost of Energy Figures*

The COE calculations for the 1st, 100th, 1000th, and 10,000th production units of the 15 kW machine are presented in Table XXXVI. The COE for the standard design ranges from 5.04¢ to 4.40¢ at a 5.6 m/s (12 mph) annual average wind site. The COE for the special design at the same site ranges from 5.36¢ to 3.51¢. A curve showing the reduction of COE with increasing production quantities, as compared with the design goal is provided in Figure 76.

*The fixed charge rate cost of energy calculation method used in this report was specified by Rockwell International to allow comparison of the Enertech 15 kW wind system with other machines developed under DOE funding. The reader should be aware that life cycle costing gives a more accurate cost of energy calculation. A good introduction to this method--as it is applied to wind systems--can be found in SWECS Cost of Energy Based on Life Cycle Costing, W.R. Briggs, Rocky Flats Wind Systems Program, RFP-3261, May 1980 (available from NTIS.)

Table XXXV

Cost of Energy Calculations (V = 5.6 m/s)

	STANDARD				SPECIAL			
	1	100	1,000	10,000	1	100	1,000	10,000
F.O.B. cost in 1980 $	22,869	19,382	16,353	11,934	25,463	22,011	18,671	14,031
Installation/Foundation cost in 1980 $	3,040	2,716	2,646	2,576	3,530	3,206	3,136	3,066
1.4 x F.O.B. cost in 1980 $	32,017	27,135	22,894	16,708	35,648	30,815	26,139	19,643
Total in 1980 $	35,057	29,851	25,540	19,288	39,178	34,021	29,275	22,709
Total in 1978 $ (13% inflation 1979) (7.5% inflation mid 1980)	28,859	24,574	21,025	15,875	32,252	28,007	24,100	18,695
C.O.E. = $\frac{(I.C.*0.087)+ACC}{AKWh}$ x100 ACC = **$397 (Adj. for Infl.)** AKWh = 54608 @ 12 mph 90% Avail. in ¢/KWh	5.32	4.64	4.07	$\boxed{3.26}$	5.87	5.19	4.57	3.71
C.O.E. at 12 mph 95% Avail.	5.04	4.40	3.86	3.09	5.36	4.92	4.53	3.51

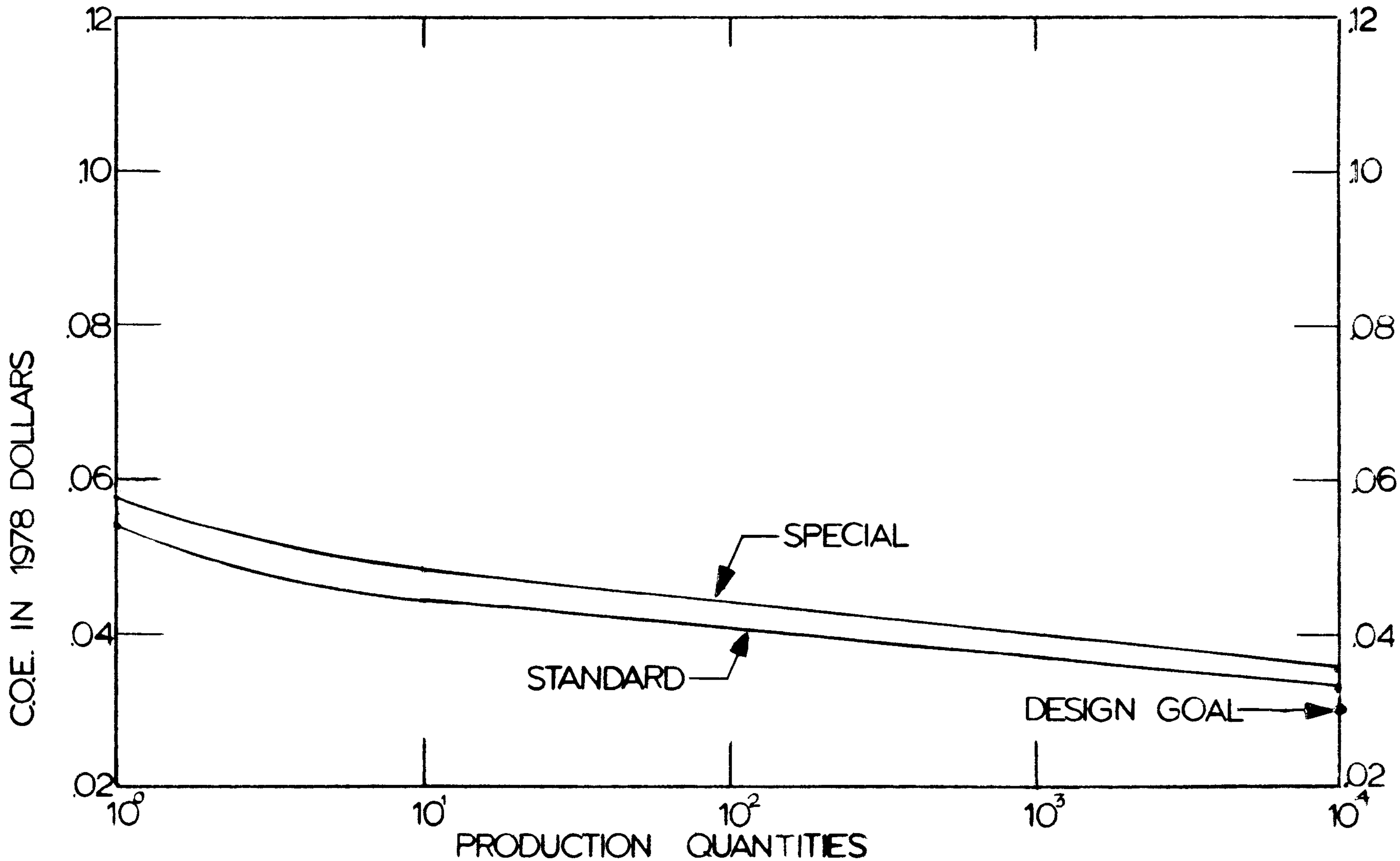


Figure 76

Cost of Energy Projection - Enertech 15 kW

7.0 PHASE II TEST PLAN

7.1 Objective

The overall objective of the Phase II Test Program is to demonstrate satisfactory operation of the complete 15 kW wind system in the "automatic" (unattended) mode for a period of at least two weeks before delivery to the DOE Rocky Flats test center. Satisfactory operation in this case implies that 1) measured performance is at least adequate to meet the output specifications of the 15 kW contact for the "standard" design, 2) that all safety and back-up systems operate as intended, and 3) that there are no unusual, inexplicable and/or potentially damaging system characteristics such as excessive noise or vibration.

An additional purpose of the test program is to provide feedback to engineering allowing further development, refinement and optimization of the design. Depending on time and budget constraints, certain of these changes will be incorporated into the delivered prototype, others will remain as recommendations for future production.

The test program will begin with individual component inspection and testing. It will then progress to testing of the complete system, including tower. The present plan is to install the unit initially on an 80-foot tower at the Enertech test site in Norwich. Later, a 20-foot base section will be removed and the unit will be tested on a 60-foot tower primarily for the purpose of determining vibrational characteristics. In the course of the testing program at least one complete assembly and disassembly operation will be performed without the use of a crane in order to demonstrate the feasibility of this erection method.

7.2 Component Testing

Several components of the wind machine will be tested prior to unit assembly. These components include the following:

Blades: Enertech will test a blade to determine its first bending mode (natural frequency). This will be accomplished by attaching the blade root to a relatively stiff fixture, exciting the blade (by knocking) at the tip and monitoring a vibration pickup output from the blade root on an oscilloscope.

Blade strength and deflection will also be tested by mounting a blade to a reinforced hub fixture in a horizontal position and applying sufficient weight properly distributed along the length of the blade to simulate the worst case design load. Since the blade design includes a 1.5 safety factor, this should not prove a destructive test. Once loaded, blade tip deflection will be measured and compared to calculated values. In addition, the blade stud interface will be fatigue tested at Ft. Eustis, Virginia. Fatigue testing will involve the repeated cycling of a test blade at various conditions of load and overload to determine where the studs fail.

Gearbox: Gearbox testing will consist mainly of detailed inspection. The gearbox will be disassembled, shafts and gearing checked for alignment and all bolts checked for correct torque. All reassembly will be performed using specified torquing values and Locktite® fastener bonding agent.

Hydraulic Brake: Hydraulic brake components will be assembled and installed on the generator test rig. Stopping torque will be measured as a function of relief valve pressure and operating characteristics (such as parasitic drag) will be noted.

Hub: Hub testing will consist of Enertech pattern inspection prior to casting and radiographing of the casted hub parts. This radiography will be done according to ASTM specifications with copies of the x-rays sent to Rocky Flats. In addition, Enertech personnel will be present for certain phases of the machining of the castings.

Tower: Enertech proposes to test the 60 and 80-foot towers to determine their natural frequencies. This will be accomplished before the unit is installed by applying a sinusoidal exciting function to the tower and measuring tower vibration through the use of the unit vibration sensor.

Induction Generator: Induction generator testing is included in the statement of of work and has been ongoing. In addition, development work with T&L Electric, White River Junction, Vermont in the rewinding of the Baldor three-phase generator to single-phase has been conducted to optimize generator configuration. One of the two single-phase units will be selected for use on the prototype.

7.3 System Testing

Once the unit is installed on the tower the following tests will be conducted (not necessarily in this order):

Power/Energy Output: Duration - Four Months

Energy Output: Throughout the test period energy output will be measured and correlated with average wind speed data. Various corrections will be applied to account for air density changes caused by changes in temperature and geometric pressure. Although limited by conditions prevailing at the test site during this period, it is hoped that data can be obtained for winds averaging between 2.2 and 6.7 m/s (5 and 15 mph).

Power Output: In addition to these long term tests, intensive short term measurements of power output versus wind speed will be made during periods of wind activity ranging between 2.2 m/s and 18 m/s. The objective is to obtain data correlating output with wind speed at one mile-per-hour intervals from cut-in up 15 m/s (34 mph)--or higher, if conditions permit. Estimated duration of intensive test: three to four weeks. Data from both these tests will be compared to predicted values and revisions made where necessary.

Yaw Response: Duration - two weeks

Wind direction and wind machine yaw position will be recorded simultaneously, allowing a determination of machine tracking characteristics. Yaw rates will be measured and correlated with wind direction changes. Qualitative observations will also be made by comparing yaw behavior with a visual wind vane reference. These tests will be carried out in light winds 2.2 to 4.5 m/s with both stopped and turning rotor and in higher winds (up to 18 m/s) with a normally turning rotor.

Because it is believed that tower shadow may have a significant effect on yaw angle (especially at higher wind speeds) a portion of the tower will be blocked to determine what effect this has on yaw angle or response characteristics.

Cycling Data: Duration - Four Months

The number of generator start-up cycles, running cycles at operating speed, and brake cycles will be recorded daily. Total generator run time will be recorded and correlated with average wind speed data.

Vibration Monitoring: Duration - Two to Three Weeks

Vibration levels at the gearbox, generator-brake, yaw bearing and tower top will be recorded and correlated with rotational speed data and power output data. Any significant vibration modes will be defined. Response to imbalance will be determined by placing a known imbalance on the rotor and recording the vibration levels. In addition, the tower will be blocked with canvas to determine unit sensitivity to tower shadow.

Freewheeling Characteristics: Duration - One Week

Rotor speed response and yaw response will be recorded with variations of wind speed and direction for the rotor free wheeling mode (brake disengaged - generator disengaged). These tests will be conducted at wind speeds ranging from 2.2 to 6.7 m/s. These observations will aid in control system adjustment.

Tip Brake Effectiveness: Duration - Two Weeks

The effectiveness and deployment characteristics of the tip brakes will be tested (by disengaging the generator-brake and allowing overspeed). Deployment will be cycled several times at wind speeds ranging from 6.7 to 15.6 m/s if possible. In addition, the brakes will be passively tested for the full four months testing period to determine if any spurious deployment occurs.

Control System Testing and Tuning: Duration - Two Weeks

Observation of performance parameters of the machine in a real life environment will allow the proper adjustment of the control system. Optimum cut-in/cut-out rotational speed settings will be determined. The man-machine interface will be evaluated (including machine serviceability). Suggestions will be made for future improvements.

Blade Pitch Optimization: Duration - Two Weeks

To determine the effects of varying blade pitch, the unit will be run in each of three modes for several days. Total variation possible is approximately three degrees. Therefore the unit will be run in a neutral position, a position of less angle of attack and finally a position of greater angle of attack. By correlating energy output and peak power output to windspeed data, an optimum pitch setting can be defined.

System Natural Frequency Determination: Duration - Four Weeks

As proposed under component testing, both the 80-ft and 60-ft towers will be tested to determine natural frequencies. This will be done before the unit is installed to determine tower natural frequencies then after the unit is installed to find system natural frequencies. This information will serve to identify possible vibration problems as well as verify the accuracy of our original analysis.

7.4 Conclusion

The data from the above tests will be reduced and presented in final report form. In addition it is likely that further tests will come to light and be incorporated in the test plan. However, it is felt that the above plan will provide a comprehensive test of the systems capabilities.

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