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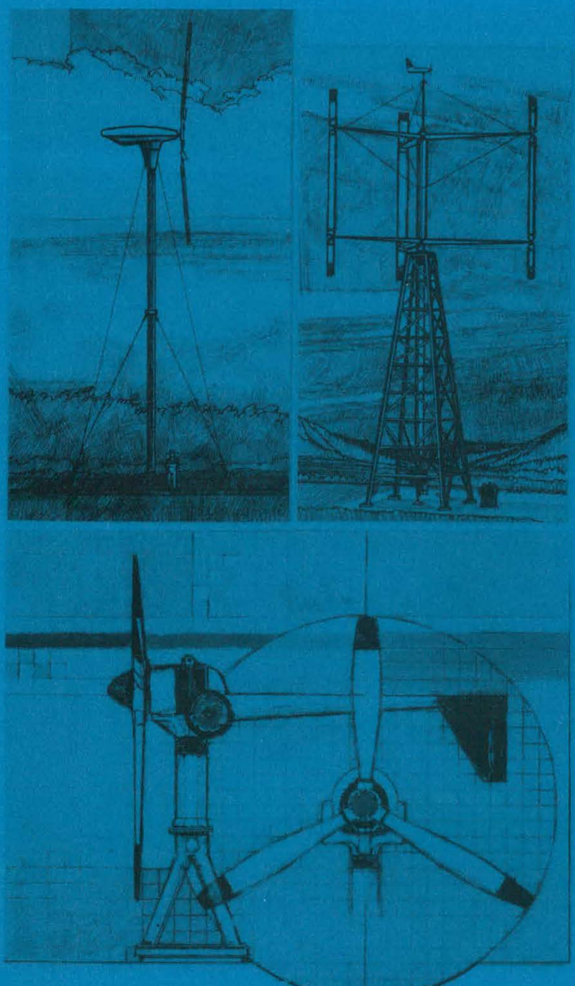
STRUCTURAL COMPOSITES INDUSTRIES 4-KILOWATT WIND SYSTEM DEVELOPMENT

Phase I - Design and Analysis Technical Report

May 1981

N. Malkine
G. Bottrell
O. Weingart



Prepared by
STRUCTURAL COMPOSITES INDUSTRIES
6344 North Irwindale Avenue
Azusa, California 91702

For
Rockwell International Corporation
Energy Systems Group
Rocky Flats Plant
Wind Systems Program
P.O. Box 464
Golden, Colorado 80402

Subcontract No. PF07420C

As a Part of the
U.S. DEPARTMENT OF ENERGY
WIND ENERGY TECHNOLOGY DIVISION
FEDERAL WIND ENERGY PROGRAM

Contract No. DE-AC04-76DP03533

DISCLAIMER

PATENT STATUS

The composite blades and tower described in this report are covered by one or more of the following U.S. Patents issued to Structural Composites Industries, Inc., Azusa, California

U.S. Patent No. 4,260,332

U.S. Patent No. 4,264,278

and other patents pending.

The torque actuated blade pitch control system is covered by the following U.S. Patent issued to Ventus Energy Corporation, Covina, California

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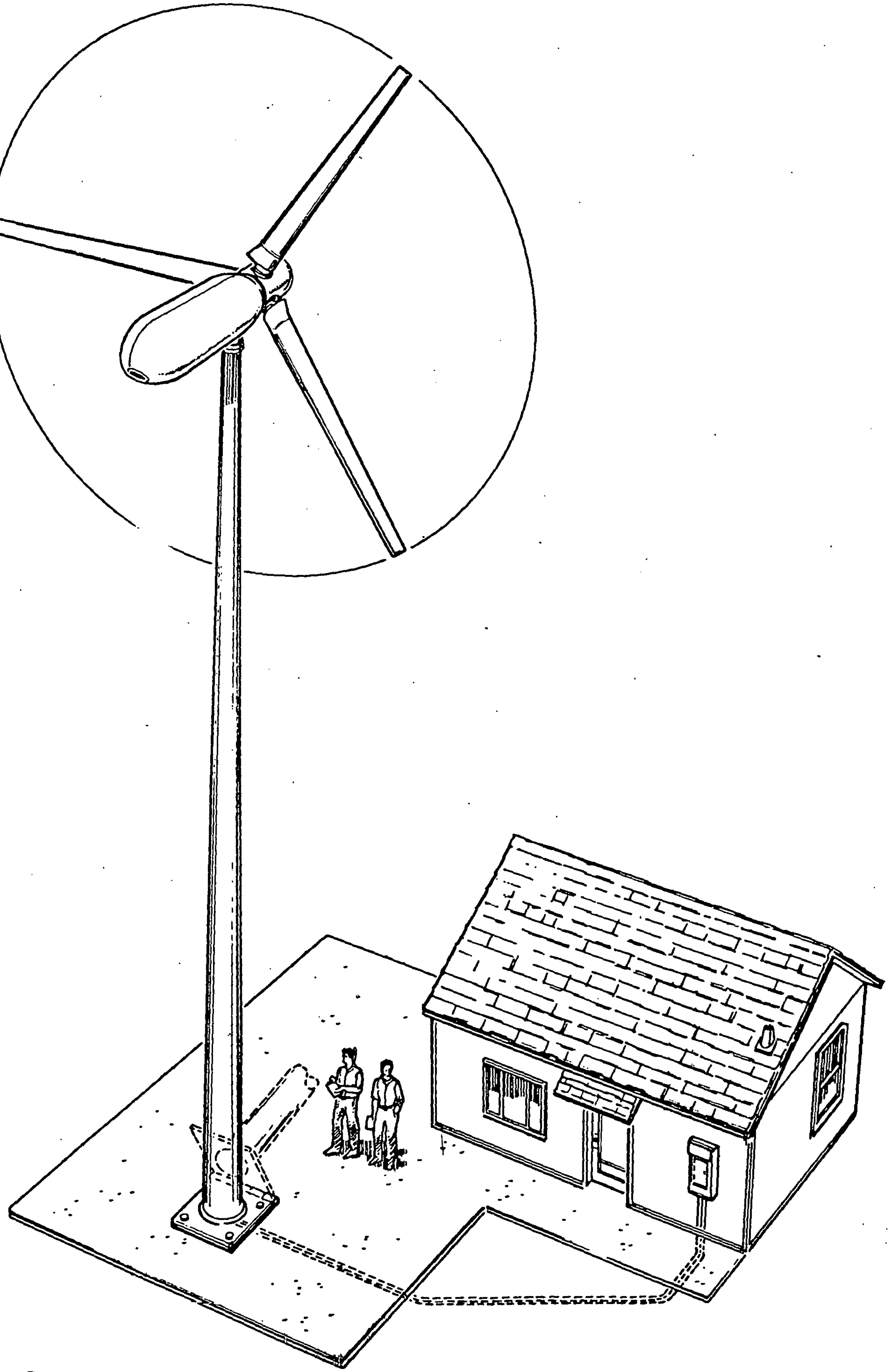

Concept - Structural Composites Industries 4kW Wind System

ROCKWELL INTERNATIONAL NOTE

The contract awarded to Structural Composite Industries, Inc. by the Rockwell International Energy Systems Group to develop the 4 kilowatt wind system described in this report was terminated after completion of the Phase I Design and Analysis effort. Thus, a prototype of the design was not fabricated or tested under federal funding.

Excess cost required to complete the project was the prime reason for termination. Contributing factors were:

1. The inability of the design to meet contract cost of energy goals

2. The extreme complexity of the system; in particular, the control system

3. The lack of technical advancement over existing wind systems

While the design did have several interesting features--such as a new type of pitch control system--these features only contributed to the complexity of the machine and did not result in lowering the cost of energy, increasing reliability, or advancing the state of the art.

ABSTRACT

A 4 kW small wind energy conversion system (SWECS) has been designed for residential applications in which relatively low (10 mph) mean annual wind speeds prevail. The objectives were to develop such a machine to produce electrical energy at 6 cents per kWh while operating in parallel with a utility grid or auxiliary generator.

The Phase I effort covered by this report began in November, 1979 and was carried through the Final Design Review in February 1981. During this period extensive trade, optimization and analytical studies were performed in an effort to provide the optimum machine to best meet the objectives. Certain components, systems and manufacturing processes were tested and evaluated and detail design drawings were produced.

The resulting design is a 31-foot diameter horizontal axis downwind machine rated 5.7 kW and incorporating the following unique features:

- Composite Blades
- Free-Standing Composite Tower
- Torque-Actuated Blade Pitch Control

The design meets or exceeds all contract requirements except that for cost of energy. The target 6 cents per kWh will be achieved in a mean wind speed slightly below 12 mph instead of the specified 10 mph.

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NOMENCLATURE

A	Area
AAWS	Annual Average Wind Speed
AOM	Annual Operating and Maintenance Cost
ac	Alternating Current
C	Chord
^{o}C	Degrees Celcius
CDR	Critical Design Review
c.g.	Center of Gravity
COE	Cost of Energy
Cp	Rotor Power Coefficient
Cpr	Rotor Power Coefficient For a Local Blade Station at Radius r
$\frac{C}{P}$	$\frac{\text{Basic Load Rating}}{\text{Load}}$ (For Bearings Life Evaluation)
DIA	Diameter
dc	Direct Current
DC	Decreasing Pitch Angle
DBT	Double Bias Tape
DLO	Direct Labor Overhead
DOE	Department of Energy
E	Young's Modulus
^{o}F	Degrees Farenheit
FDR	Final Design Review
FMEA	Failure Mode and Effects Analysis
Fx	Force in the Lateral Direction
Fy	Force in the Fore-Aft Direction
Fz	Force in the Vertical Direction
Fr	Force in the Radial (Centrifugal) Direction
ft	Foot
G	Shear Modulus
G.& A	General and Administrative Overhead
Hz	Hertz (Cycles Per Second)

NOMENCLATURE

I	Area Moment of Inertia
IC	Increasing Pitch Angle
in.	Inch
J	Torsional Area Moment of Inertia
k	Stiffness
kW	Kilowatt
kWh	Kilowatt Hour
ksi	1000 lb Per Square Inch
L	Pitch Ball Screw Lead
Lf	Feathering Ball Screw Lead
lb	Pound
lb-ft	Pound x Foot (Moment)
$lb\text{-}ft^2$	Pound x $Foot^2$ (Inertia)
LFT	Longitudinal Filament Tape
LTLT	Linear Taper Linear Twist
mph	Miles Per Hour
m/s	Meter Per Second
Mx	Moment About the Lateral Axis
My	Moment About the Fore-Aft Axis
Mz	Moment About the Vertical Axis
MTBF	Mean Time Between Failures
MTTR	Mean Time To Repair
OPOP	Optimum Taper Optimum Twist
OSHA	Occupational Safety and Health Administration
oz	Ounce
P	Power
PDR	Preliminary Design Review
psi	Pound Per Square Inch
R	Radius of Blade
r	Distance From Centerline of Rotation to Local Blade Station
R_g	Generator Sprocket Number of Teeth
r_s	Pitch Sprocket Number of Teeth
rpm	Revolutions Per Minute

NOMENCLATURE

S	Slip Speed of Center of Contact Area
SCI	Structural Composites Industries
SF	Safety Factor
SM	Safety Margin
SOW	Statement of Work
SWECS	Small Wind Energy Conversion System
t	Thickness
TFT	Transverse Filament Tape (SCI Patented Process)
ULT	Ultimate
UTRC	United Technologies Research Center
v	Wind Velocity
$\bar{v}$	Average Mean Wind Speed
W	Weight
X	Tip Speed Ratio $\frac{\Omega R}{v}$
YR	Year
Zo	Roughness
ZTZT	Zero Taper Zero Twist
β	Blade Pitching Displacement (degrees)
β	Flux Density
ρ	Density (lb/cu ft)
η	Efficiency
ν	Poisson Ratio
σ	Stress (psi)
Ω	Rotor Rotational Speed

1.0 INTRODUCTION

1.1 BACKGROUND

In 1977 the Department of Energy initiated a program for the study and development of Small Wind Energy Conversion Systems (SWECS). As a part of this program, a contract was issued in October, 1979 to Structural Composites Industries (SCI) for the design, development and construction of a 4 kW SWECS. The unique features of the SCI proposed design were:

(a) Composite blades

(b) A simple mechanical speed control system based on rotary motion of the generator housing

(c) A tapered composite tower

1.2 PROGRAM MANAGEMENT

Management of the program for DOE was assigned to the Rockwell International Energy Systems Group, which operates the Rocky Flats SWECS Test Center near Golden, Colorado.

1.3 PROGRAM OBJECTIVES

1.3.1 Phase I - Design and Component Development

1. To develop a complete, detailed design of a SWECS to the specifications in the Statement of Work (SOW), that is capable of generating electrical energy at a cost competetive with alternate energy sources.

2. To develop a special alternate design that would be adequate to withstand more severe environmental conditions.

3. To provide drawings and specifications of the final complete SWECS in sufficient detail to allow a suitably equipped manufacturer to fabricate additional units.

1.3.2 Phase II - Prototype Fabrication and Testing

To construct a prototype of the complete SWECS and, after checkout, to ship it to the Rocky Flats Test Center for test and evaluation.

1.4 PROGRAM REQUIREMENTS

The SWECS was required to produce 3 to 6 kW in wind speeds between 15 and 20 miles per hour, when operating in parallel with a utility grid or an auxiliary generator. The target cost of energy, for the 10,000th production unit, was to be 6 cents per kwh, in 1978 dollars, when operating in a 10 mph mean annual wind speed. Other design criteria are summarized in Table I-1.

Table I-1
BASIC DESIGN CRITERIA

	STANDARD DESIGN	SPECIAL DESIGN
OUTPUT:	* 7,500 TO 15,000 kWh/yr IN A WIND REGIME HAVING A MEAN ANNUAL WIND SPEED OF 4.6 m/s (10 mph) MEASURED AT 9.1 m (30 ft) ABOVE GRADE	SAME
	120/240 ± 5% ac SINGLE PHASE, 60 Hz INTERTIE WITH A UTILITY OR WITH AN INTERCONNECTION WITH AN AUXILIARY GENERATOR (E.G. DIESEL).	SAME
	* DESIGN ENVELOPE: 3 TO 6 kW AT A WIND SPEED BETWEEN 6.7 AND 9.0 m/s (15 AND 20 mph).	SAME
OPERATING WIND RANGE:	CUT-IN: MINIMIZE CUT-OUT: MAXIMIZE SURVIVAL: 56 m/s (125 mph) PEAK GUST	SAME 74 m/s (165 mph) PEAK GUST
OPERATION ENVIRONMENT:	-30°C TO 60°C (-22°F TO 140°F) ICE 1" THICK* RAIN, DUST, LIGHTNING	-50 C TO 60 C (-58 F TO 140 F) ICE 2½" THICK* SAME + SALT WATER SPRAY
OPERATION AVAILABILITY:	95% AVAILABILITY FACTOR	SAME
SYSTEM LIFE:	25 YEARS, MINIMUM	SAME
ENERGY COST GOAL:	6¢/kWh FOR 10,000th UNIT (1978 DOLLARS)	SAME
CONTROLS:	AUTOMATIC STARTUP AND SHUTDOWN. ROTOR OVERSPEED PROTECTION. BRAKE FOR LOCKING ROTOR NOT REQUIRED IF BLADES ARE FULLY FEATHERED.	SAME

* NEED NOT OPERATE WHILE ICE COATED

1.5 REPORTING ACTIVITIES

During the Phase I period, five formal design reviews were convened: Tradeoff and Loads Review (TLR), Preliminary Design Review 1 (PDR-1), Preliminary Design Review 2 (PDR-2), Critical Design Review (CDR), and Final Design Review (FDR). These reviews, together with monthly progress reports, provided DOE and Rockwell International the means to effectively monitor the program progress and technical adequacy.

1.6 SCI PROPOSAL

SCI's original design consisted of a 31-ft diameter, 2-bladed, down-wind machine with variable pitch blades, rigid hub and induction generator.

The blades, tower and nacelle were to utilize high strength composite materials for low cost, light weight, high structural integrity and aesthetically pleasing appearance.

The induction generator was to be a two-speed unit to maximize energy recovery in the very low wind speeds common to a 10-mph mean annual wind speed.

The rotor pitch control system was to be a simple direct electro-mechanical system which utilizes generator reaction torque to sense torque levels and to drive the pitch control mechanism.

The final design differed in some key aspects from the original.

1.7 SCOPE OF PHASE I REPORT

This report documents the effort which began in October, 1979 and ended at the Final Design Review in February, 1981. It covers the sequential and reiterative analyses for optimization of the SWECS in order to minimize the cost of energy while providing high reliability and a 25-year machine life.

2.0 CONFIGURATION OVERVIEW

2.1 DESIGN EVOLUTION

Evolution of major design features, from proposal through Final Design Review, is shown in Table 2-1. Figures 2-1 and 2-2 depict nacelle arrangements for the proposal and final designs, respectively.

Table 2-1 DESIGN EVOLUTION

FEATURE	SCI PROPOSAL APRIL, 1979	PDR-2 JUNE, 1980	CDR SEPT., 1980	FDR FEBRUARY, 1981
ROTOR SIZE & TYPE				NO CHANGE
NO. BLADES	TWO	THREE	THREE	
TYPE HUB	RIGID	RIGID	RIGID	
ORIENTATION	DOWNWIND	DOWNWIND	DOWNWIND	
DIAMETER (ft)	31	32	31	
CONING (DEGREES)	5	5	5	
HUB HEIGHT (ft)	51.25	51.25	51.25	
ROTOR CONTROL			NO CHANGE	NO CHANGE
TYPE	VARIABLE PITCH	VARIABLE PITCH		
SPEED (rpm)	54/107	94		
START UP	MOTOR	AERODYNAMIC		
SHUT DOWN	FEATHER	FEATHER		
BLADE TYPE		NO CHANGE		NO CHANGE
AIRFOIL (NACA)	23012		44xx	
GEOMETRY	ZTZT		LTLT	
CONSTRUCTION	PULTRUDED FIBERGLASS		FILAMENT WOUND	
TOWER		NO CHANGE	NO CHANGE	
LENGTH (ft)	50			48.5
TYPE	FREE-STANDING			FREE STANDING
CONSTRUCTION	TAPERED DIA. TAPERED WALL			TAPERED DIA. TAPERED WALL
MATERIAL	TFT* E-GLASS POLYESTER			TFT* E-GLASS POLYESTER
GENERATOR			NO CHANGE	
TYPE	INDUCTION	INDUCTION		INDUCTION
SPEED (rpm)	1800/900	1850		1862
VOLTAGE (volts)	240, 1 PHASE	240, 1 PHASE		240, 1 PHASE
UTILITY INTERCONNECTION	DIRECT	DIRECT		DIRECT
SYSTEM OUTPUT			NO CHANGE	NO CHANGE
RATED (kW)	5.7 @ 15 mph (1)	5.7 @ 15.7 mph (1)		
PEAK (kW)	6.3 @ 50 mph (1)	6.3 @ 50 mph (1)		
ANNUAL OUTPUT				
@ 100 mph (2) (kWh)	15,000	16,059	15,801	15,676
COST OF ENERGY				
@ 10 mph (2) (¢ per kWh)	6.0	6.8	6.9	8.0
SYSTEM WEIGHT				
NACELLE/TOWER (lb)	1395/1335	1336/1157	1299/1236	1600/1270
CUT-IN WIND SPEED (1) (mph)	6	7.6	8.22	8.84
RATED WIND SPEED (1) (mph)	15	15.7	NO CHANGE	NO CHANGE
CUT-OUT WIND SPEED (1) (mph)	50	NO CHANGE	NO CHANGE	35
SURVIVAL WIND SPEED (1) (mph) STANDARD/SPECIAL DESIGN	125/165	NO CHANGE	NO CHANGE	NO CHANGE

(1) WIND SPEED MEASURED AT 30 ft
(2) MEAN ANNUAL WIND SPEED MEASURED AT 30 ft

ZTZT - ZERO TAPER/ZERO TWIST
LTLT - LINEAR TAPER/LINEAR TWIST
*SCI PATENTED

Figure 2-1 NACELLE CONFIGURATION - Proposal

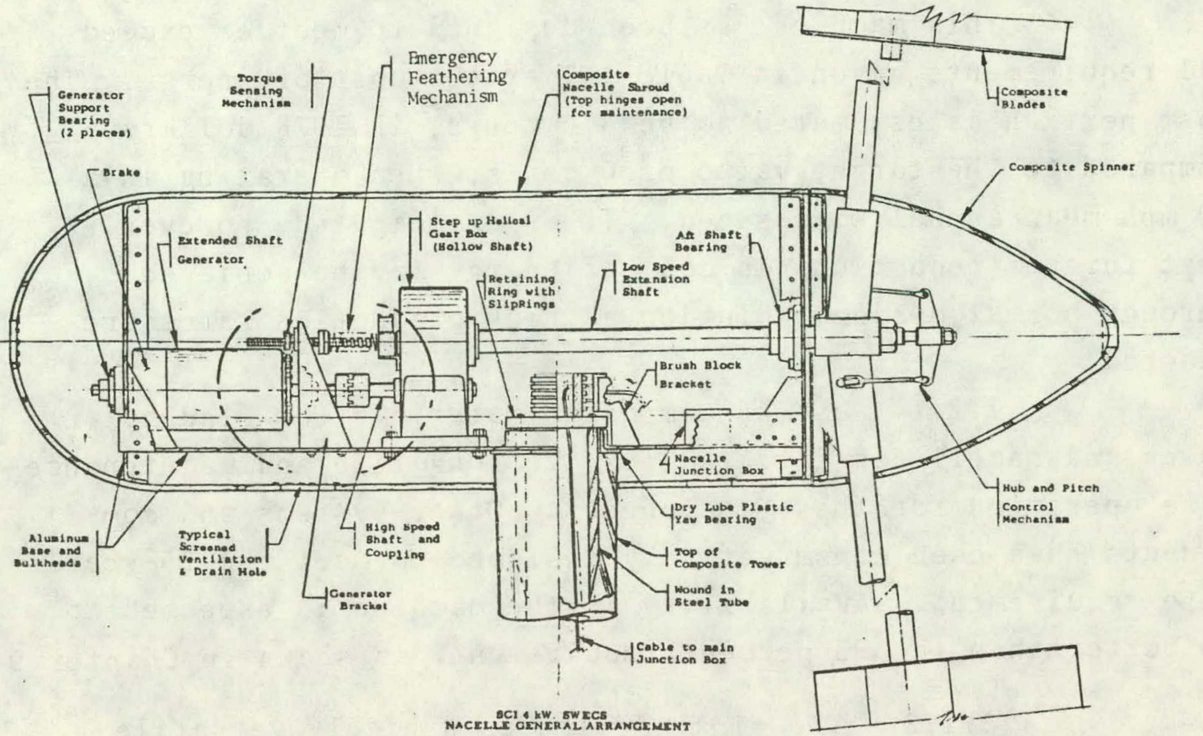
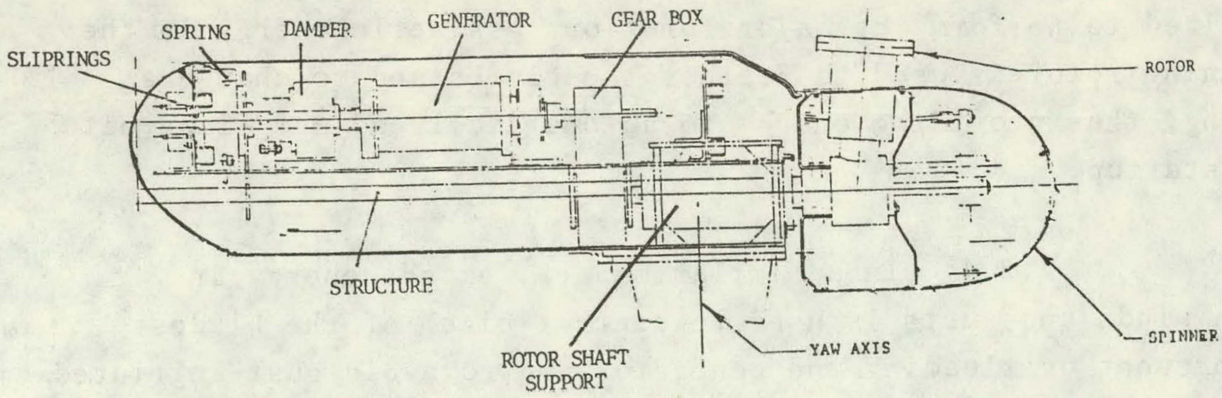


Figure 2-2 NACELLE CONFIGURATION - FDR



2.2 FINAL DESIGN FEATURES

This machine has been designed to meet or exceed all requirements given in Table 1-1 except cost of energy. The cost per kwh is estimated to be 8.0 cents, in 1978 dollars, compared to the target value of 6 cents, when operating in a 10 mph mean annual wind speed. It is anticipated, however, that further reductions in cost of energy may be achieved through normal design evolution as high production rates are reached.

2.2.1 Use of composite materials for blades, tower and nacelle is expected to give long life and maintenance-free operation for these components. Other systems and components have been conservatively designed to meet the 25-year life requirement. Availability of the machine is expected to be better than the 95 percent requirement, as shown in Chapter 9.

2.2.2 The final design is a highly versatile machine which can operate in a wide range of wind regimes at near optimum performance. The machine can be started, operated or stopped at any wind speed between the specified cut-in and cut-out values.

2.2.3 The rotor control system is fully automatic and essentially passive in that virtually no external power is required to perform the major functions. Kinetic energy in the spinning rotor is used to feather the blades and to charge a spring, thus providing energy to automatically adjust blade pitch for startup.

2.2.4 In a similar manner, excess energy in high winds and gusts is used to control pitch of the blades to prevent overloading the generator and to avoid gust-initiated disturbances on the interconnected electric power network. The pitch control system also avoids high steady and cyclic blade and tower loads during normal operation above rated wind speed.

2.2.5 A tower erection system, consisting of ginpole, provisions for a towtruck and hinged tower base, has been incorporated in the design. This will allow low-cost erection of the system without the need for a crane.

2.2.6 The SWECS is designed for direct interconnection with the alternating current utility network or auxiliary generator without the need for batteries or power conditioning equipment. Electrical controls consist of 110-volt electro-mechanical relays mounted in a control panel at ground level. This allows trouble shooting, repair and adjustment of most electrical controls without lowering the tower.

3.0 ROTOR DESIGN

3.1 HISTORICAL SUMMARY

3.1.1 Proposal

A two bladed rotor with pultruded fiberglass 23012 airfoil was chosen initially. The constant 15 inch chord untwisted blade was proposed for minimum cost, assuming the use of the existing UTRC 8 kW blade poltrusion die to make the prototype.

Trade-offs on the number of blades were performed. The highlights are that a two bladed rotor provides a simple hub design and a lower expected cost. There was a concern that a two bladed rotor might have some inherent vibration problems, excited by the gravity forces, causing reversed flexures of the blades. For 3 blades at 120 degrees, the disturbing forces are 240^{o} out of phase so the three forces add to zero.

The downwind rotor was 31 feet in diameter, rigid and with pitch control at the hub through pushrods actuated by the control system. Cut-in wind speed was to be 6 mph with associated rotor speeds of 54-56 rpm. For wind speeds above 11 mph, 107-110 rpm. The bearings for the blades and for the pitch control shaft were assumed to be of the plastic dry lube type.

3.1.2 PDR I and II

There was a significant change between PDR I and II. A rotor trade-off between several airfoils, two or three blades and hybrid, fixed or variable pitch control was performed. A summary of the results is presented in Table 3-1.

Table 3-1 ROTOR TRADE SUMMARY MATRIX

	HYBRID PITCH	FIXED PITCH	VARIABLE PITCH	
NUMBER OF BLADES	3	3[1]	2	3
BLADE GEOMETRY: ZERO TAPER ZERO TWIST	DID NOT SUPPLY ADEQUATE PERFORMANCE	DID NOT SUPPLY ADEQUATE PERFORMANCE	DIA: 31' AkWh: 14,928 COE: .0679	DIA: 31' AkWh: 15,385 COE: .0684 DIA: 32' AkWh: 16,058 COE: .0666
BLADE GEOMETRY: LINEAR TAPER LINEAR TWIST	DIA: 32' AkWh[2]: 15,656 COE[3]: .0744	DIA: 31' AkWh: 15,290 COE .0773 DIA: 32' AkWh: 15,783 COE: .0755	DIA: 31' AkWh: 15,644 COE: .0683	DIA: 31' AkWh: 16,266 COE: .0691 DIA: 31' AkWh: 16,931 COE: .0673
BLADE GEOMETRY: OPTIMUM TAPER OPTIMUM TWIST	DIA: 32' AkWh: 16,119 COE: .0832	DIA: 31' AkWh: 16,508 COE: .0835	DIA: 31' AkWh: 16,624 COE: .0750	DIA: 31' AkWh: 17,168 COE: .077

1 2-BLADED FIXED PITCH MACHINES DID NOT SUPPLY ADEQUATE PERFORMANCE.
2 AkWh AT 95% AVAILABILITY.
3 ADJUSTED COE.

The results shown in Table 3-1 were based on the following conditions:

(a) The SWECS operates at substantially constant speed, with direct connection to the utility grid through an individual generator.

(b) Rotational speed for each design was selected to maximize annual kwh production in a 10 mph mean annual speed.

(c) Cost of energy is based on cost of the 10,000th production unit, operating in a 10 mph mean annual wind speed.

The SCI recommendation at PDR II was for a three bladed rotor, 32 foot diameter, zero taper, zero twist, aerodynamic start up, one speed operation by variable pitch and shut down by feathering.

In order to evaluate an additional alternative for advanced blade fabrication technique, it was decided to utilize a 31 foot diameter rotor with linear tapered linear twisted blades (NACA 44XX series airfoil).

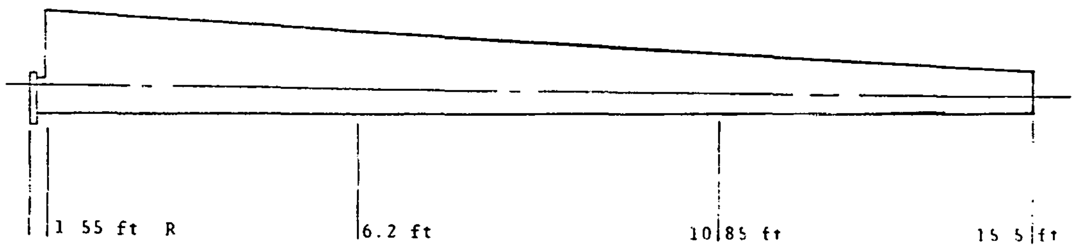
3.2 FINAL DESIGN (FDR)

3.2.1 Rotor Description

ITEM	PROTOTYPE DESIGN
ROTOR TYPE	DOWNWIND
NUMBER OF BLADES	3
BLADE GEOMETRY	LINEAR TAPER LINEAR TWIST
DIAMETER (ft)	31.0
ROOT CHORD (in)	14.97
TIP CHORD (in)	6.00
ROOT TWIST (DEGREES) (1)	4.48^{o}
TIP TWIST (DEGREES) (1)	-1.75^{o}
NACA AIRFOIL	44XX
CONING ANGLE	5^{o}
HUB	RIGID
Rpm	94 (2)
ROTOR SOLIDITY	.048
RATED POWER (kW)	5.7
MEAN ANNUAL WIND SPEED AT 30 ft	4.47 m/s (10 mph)
CUT IN WIND SPEED AT 30 ft	3.95 m/s (8.84 mph)
RATED WIND SPEED AT 30 ft	7.03 m/s (15.72 mph)
CUT OUT WIND SPEED AT 30 ft	15.64 ms (35 mph)
SURVIVAL WIND SPEED AT 30 ft	55.9 m/s (125 mph)
AERODYNAMIC C_p (PEAK)	.426

(1) 0^{o} Reference is at 0.75R

(2) At Rated Wind Speed



.09 .1	r/R	.4	7	1 0
--	AIRFOIL	4418	4415	4412
4.48°	TWIST (1)	2.40°	0 33	-1 75°
14.97	CHORD(in)	11.99	9.00	6.0
7.8	THICKNESS(in)	2.16	1.35	.72
196500	EI lb-ft^2	55610	9994	1281
83240	GI lb-ft^2	26140	4702	339

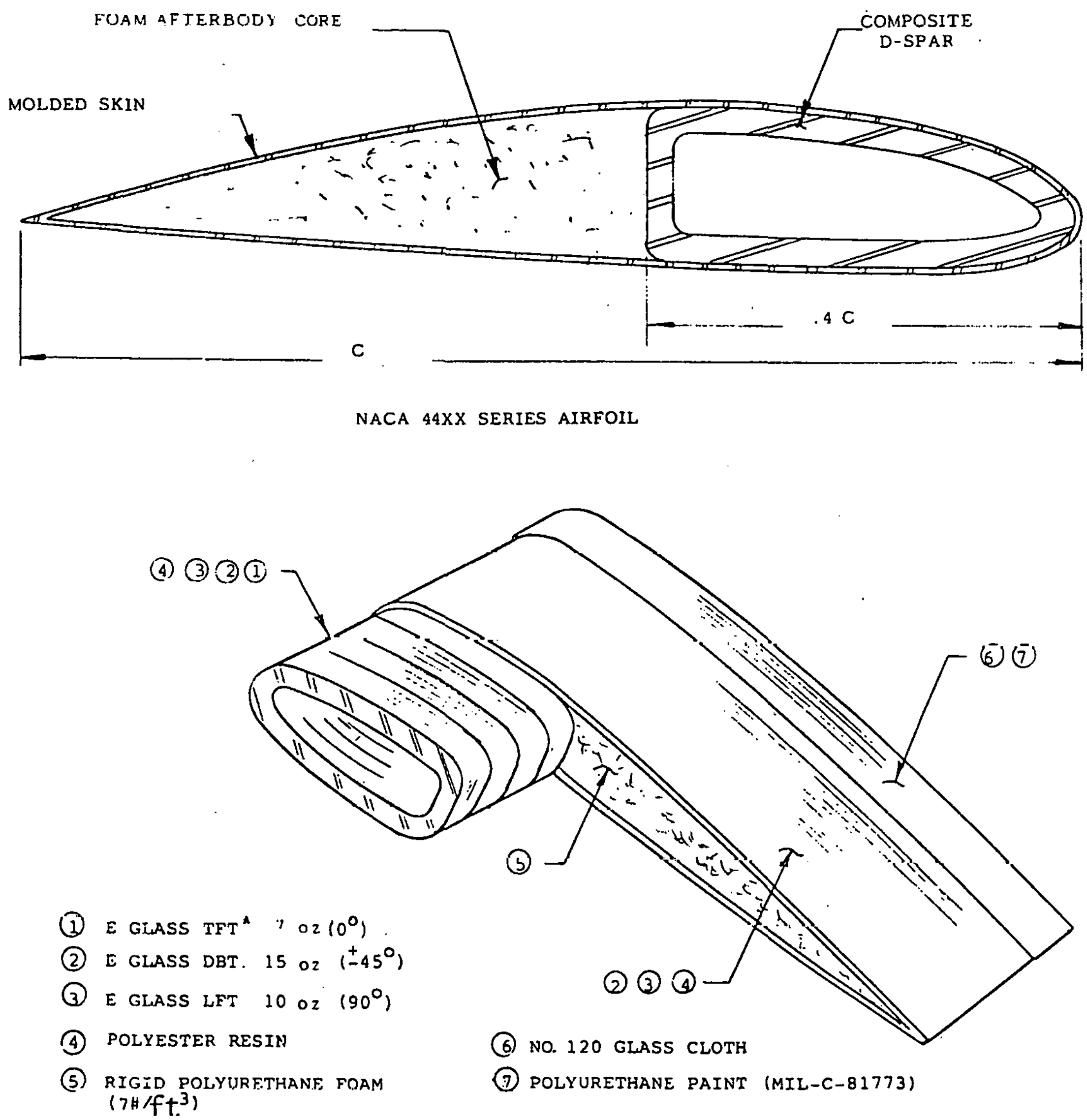
(1) 0^{o} Reference is at 0.75R

Figure 3-1 BLADE GEOMETRY DEFINITION

3.2.2 <u>Rotor Blade Description</u>

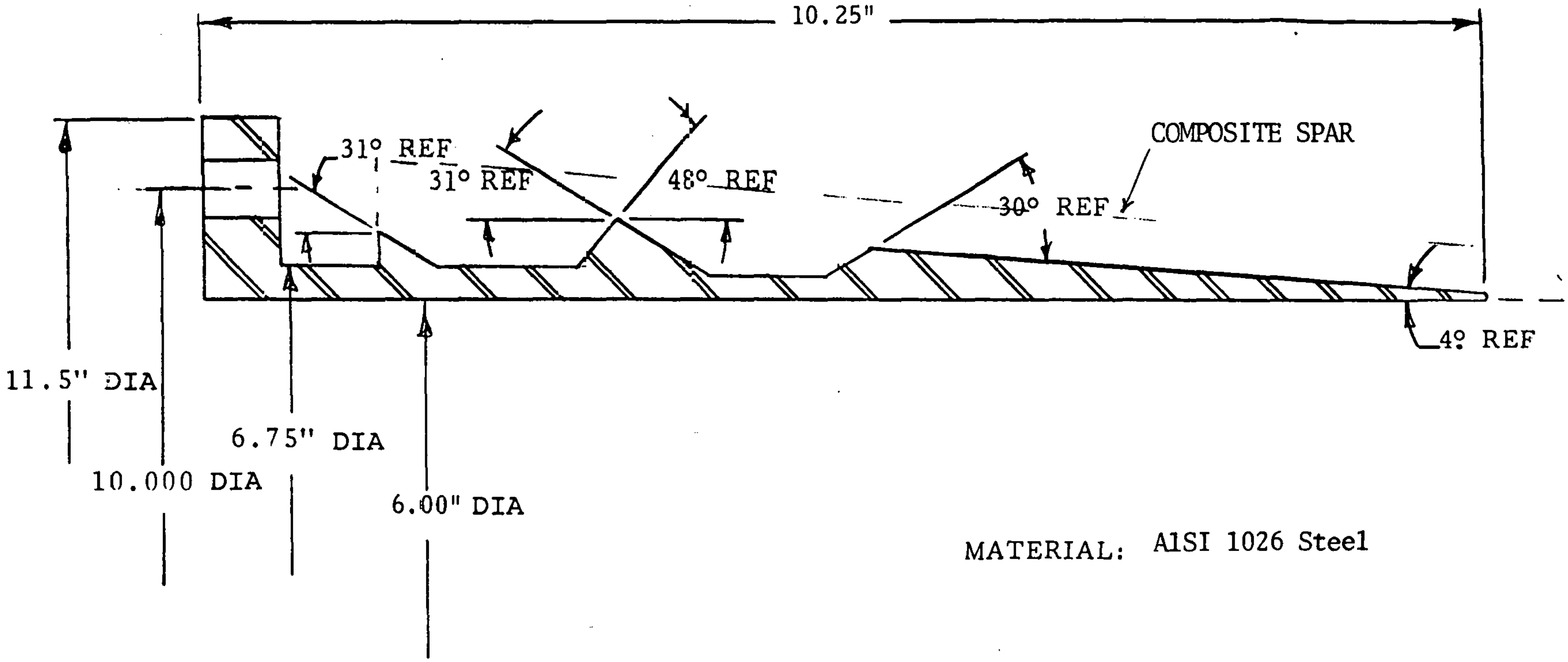
The composite linear tapered linear twisted blade is constructed in the following way (Figure 3-2).

The composite D-spar is wound over a steel mandrel using the SCI patented TFT process (Figure 3-3), and serves as the main structural element of the blade. The SCI patented steel flanged hub fitting (Figure 3-3), is wound into the root end of the D-spar, which transitions from a circular cross section at the hub to a true airfoil D-spar. The D-spar is then compression molded and cured in matched female molds with the polyurethane foam afterbody core and the fiberblass skin. The D-spar mandrel is extracted after cure.



*Patented

Figure 3-2 BLADE DESCRIPTION



MATERIAL: AISI 1026 Steel

Figure 3-3 BLADE ROOT FITTING (Patented)
RETENTION DETAIL

3.2.3 Rotor Mechanical Description

The hub is a welded steel structure mounted on the rotor shaft and holds the three blade supports. The three blade supports are mounted on angular contact bearings which provide low friction and good resistance to axial and radial loads.

A horn, whose position is defined by the pitch control shaft (controlled by the mechanism in the nacelle), actuates the blades to the desired pitch angle. Eight inch displacement of the pitch control shaft provides 88 degrees of rotation of the blades, which is the range needed from start up, through operating range, to feathering. The pitch control shaft and the pitch guide shaft (which turns the pitch horn with the rotor) slide on linear bearings to reduce friction. All the bearings are sealed to prevent atmospheric contamination. The pushrods which serve as torquers are provided with permanent lubricated rod end bearings.

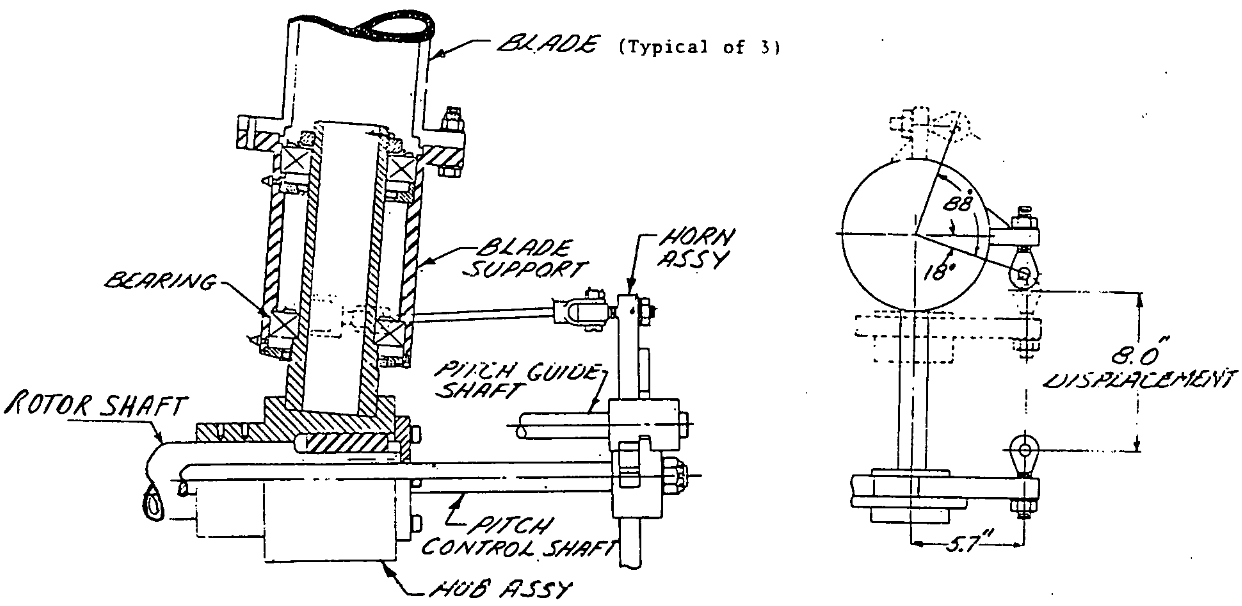


Figure 3-4 ROTOR, GENERAL ASSEMBLY

3.3 SUPPORTING ANALYSES (11) (12)

3.3.1 Fatigue Loads (See Table 3.2)

The fatigue loads were determined from the "PROP" program. A one mode flapping analysis determines the effects of yaw, shear, centrifugal, gravity and flap. For the critical loading conditions of 10^{o} yaw, Z_o = 0.1 m, the steady and oscillating contributions to the flap and lead/lag bending moments are determined. The influence coefficients for the flap wise flapping were integrated in time using a 30^{o} wide tower shadow (30% velocity deficit) to examine the effect of tower shadow.

Table 3-2 BLADE FATIGUE LOADS

X	V_{HUB} fps	Pitch Angle β degrees	Bending Moment At Root M_{FLAT} lb_f-ft	ΔM_{FLAT} lb_f-ft	F_{SHEAR} lb_f	Deflection q_{TIP} (1) (2) inches
3.0	51.38	25.12	-179	± 201	63	-.849 ±.922
3.5	43.93	21.03	-120	± 151	65	-.509 ±.679
4.0	38.37	17.62	-60	± 119	67	-.177 ±.508
4.5	34.06	14.61	+1	± 92	73	+.158 ±.386
5.0	30.62	11.85	68	± 73	78	+.515 ±.293
5.5	27.82	9.09	149	± 58	86	+.941 ±.218
6.0	25.47	5.69	272	± 43	99	+1.575 ±.150
6.13	24.90	3.0	394	± 32	112	+2 194 ±.107
7.0	21.61	3.0	277	± 40	96	+1.650 ±.134
8.0	18.75	3.0	164	± 43	81	+1.114 ±.142

(1) Positive deflection is away from tower. Negative is toward tower.

(2) Blade minimum clearance from tower during operation is 17 in

3.3.2 Yaw Analysis

The yaw analysis considered a rigid nacelle and tower with aerodynamic yaw forces and moments determined from the "PROP" program. The main yaw restoring moment comes from coning. A small yaw force also exists (destabilizing at low tip speed ratio & 2^{o}). The yaw damping is dominated by the contribution due to rotation of the rotor about the yaw axis. Although considered, the yaw force was not significant at any operating condition. Shear causes the rotor to track 3^{o} off the wind (Z_o = 0.1 m). Blade response was determined using the method used for the tower shadow and including the contributions due to yaw rate (aerodynamic and coriolis).

3.3.3 Survival

The survival analysis used three degrees of freedom and an energy method. The tower was given one degree of freedom in flap (crosswind). The nacelle was allowed to rotate about the yaw axis. The three blades were considered to be equivalent to two vertical flapping blades with the mass or the third blade concentrated in the hub. It was found that the nacelle c.g. must be at or ahead of the yaw axis for stability of the system through survival winds. Both blades were assumed to flap with the same mode and phase.

3.3.4 Stress and Stability Analysis

3.3.4.1 Introduction

The blade was analyzed for static stress response under 125 mph wind loading. The statically induced stresses were compared to buckling allowables developed according to Refs. 13 and 14 to derive margins of safety against stability failure of the blade. Margins of safety against a strength failure of the blade design were also determinod.

Section property data were developed from a revised blade thickness profile. Reference was made to the airfoil sections given in Figure 3 1 for geometry data needed to develop critical radii of curvature of the D-spar.

3.3.4.2 Stress Analysis

The static stress analysis of the blade design was performed by updating and analyzing a finite element cantilevered beam model of the blade former design configuration. The model (Figure 3.5) consisted of ten (10) equal length tapered beam elements with eleven (11) associated concentrated mass points.

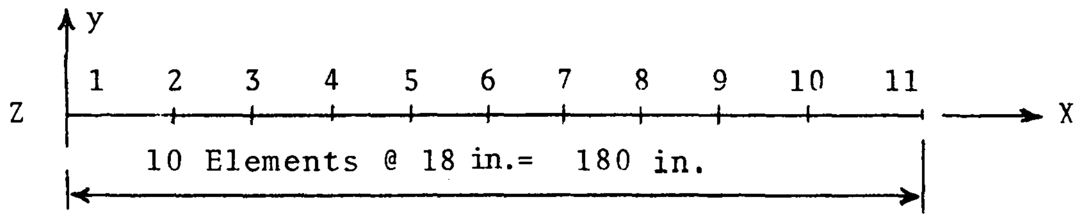


Figure 3.5 Finite Element Model Of The Blade

The blade thickness, t, for the design configuration is given according to Ref. 13. The node numbers refer to the mathematical model given in Ref. 14. The data given for the area (A) and moments of interia (I_{FLAP} and I_{Chord}) are the same as that reported in Ref. 14. The section properties at STA 0 represent those pertaining to a 6 3/8 inch diameter by 3/8 inch thick blade steel hub adapter to the nacelle. These same properties are expressed as an equivalent TFT hub adapter in Ref. 14.

As in Figure 3-1, the properties at STA'S 0.1, 0.4, 0.7 and 1.0 were determined by approximating the full scale blade sections at these stations by a set of 11, 12, 10 and 10 linear elements, respectively. The design thickness values, t, given in Table 3-3 were used in the calculations. These thicknesses represent the sum of the D-spar and skin thickness of the blade at each section location.

The bending stiffness and cross sectional area of the afterbody lamimate and foam afterbody core were neglected in the area and moments of inertia calculations. This was done since these blade components represent soft (low modulus), non-load bearing blade constituents in the primary bending and torsion directions. The afterbody weights were included, however, in the blade weight (w) and center of gravity (c.g.) calculations. The material properties are given in Table 3-3.

The results of the stress analysis including margins of safety against strength failure are summarized in Table 3-4. The minimum margin of safety against a strength failure of the blade was found to be + 1.32. The maximum tip deflections of the blade was 21.5 inches.

3.3.4.3 Stability Analysis

The blade was then analyzed for lateral stability under the applied loading condition. This was done by computing the allowable buckling stress along the blade and comparing this stress to the static stress response to derive margins of safety against blade buckling.

Using the TFT elastic constants given above, and equivalent elastic modulus E_e, was determined. The flapwise and chordwise moments of inertia were associated with bending about the Z and Y axes, respectively.

The model was analyzed for static stress response under lateral (Y axis) wind loading (p = 46 psf) and simultaneous dead weight gravity loading applied in the + X direction. The wind and gravity loadings were developed as concentrated forces applied at the nodes. A summary of the section property data and loadings for the model are given in Table 3-3.

Table 3-3 <u>Section Properties Of The Blade</u>

STA	NODE	t (in)	A (in^2)	I_{Flap} (in^4)	I_{Chord} (in^4)	$J_{Torsion}$ (in^4)	W Weight (lb/ft)	c.g (% C)
0	1	0.38	7.51	38.15	38.15	76.31	27.04	50.0
.1	2	0.40	6.03	7.67	23.04	19.65	5.57	29.0
.2	3	0.37	5.24	5.84	18.30	15.16	4.69	30.0
.3	4	0.35	4.45	4.00	13.57	10.66	3.81	30.0
.4	5	0.32	3.66	2.17	8.83	6.17	2.93	31.0
.5	6	0.29	3.06	1.58	6.65	4.48	2.62	31.0
.6	7	0.26	2.46	0.98	4.48	2.80	2.30	30.0
.7	8	0.23	1.86	0.39	2.30	1.11	1.99	30.0
.8	9	0.20	1.49	0.29	1.66	0.78	1.59	31.0
.9	10	0.18	1.12	0.18	1.03	0.45	1.19	32.0
.0	11	0.15	0.75	0.08	0.39	0.12	0.79	33.0

The material properties used in the model to represent the TFT blade are as follows:

$E_x = 3.82 \times 10^6$ psi $\qquad \rho = 0.065$ lb/cu. in.

$E_y = 1.89 \times 10^6$ psi

$G_{xy} = 0.58 \times 10^6$ psi

$\nu_{xy} = 0.265$

$\sigma_{u(0^o)} = 20.000$ psi

The Equivalent Elastic Modulus $\quad Ee = 2.2 \times 10^6$ psi

Laminate Construction $\quad$ 72.57% at 0^o

16.04% at 90^o

11.39% at $\mp 45^o$

Table 3-4 <u>Summary of Stress and Stability Analysis Results For The Blade</u>

STA (r/R)	NODE	STATIC STRESS (psi)	ULT STRENGTH (psi)	SM ON STRENGTH	BUCKLING ALLOWABLE (psi)	SM ON BUCKLING
0	1	4,120	20,000	3.85	-	-
.1	2	8,620		1.32	15,210	0.76
.2	3	7,750		1.58	14,600	0.88
.3	4	7,340		1.72	14,500	0.98
.4	5	8,340		1.40	13,780	0.65
.5	6	6,640		2.01	12,720	0.92
.6	7	5,590		2.58	11,740	1.10
.7	8	6,230		2.21	10,610	0.70
.8	9	2,980		5.71	10,430	2.50
.9	10	900		21.22	10,640	10.82
0	11	0	20,000	-	10,350	-

The buckling allowables and the associated margins of safety against stability failure of the blade are given in Table 3-4. It should be noted that a knockdown factor of 0.4 was applied to the theoretical buckling equation to derive the buckling allowables given in Table 3-4.

The minimum margin of safety against buckling of the blade was found to be + 0.65.

3.4 PERFORMANCE CHARACTERISTICS

The performance characteristics of the rotor is given in Tables 3-5 and 3-6.

Table 3-5 Rotor Performance Characteristics

X	C_{Pr}	β Degrees	P_{OUT} kW	P_{ROTOR} kW	Ω_{ROTOR} rpm	η (1)	V_{HUB} mph	V_{HUB} m/s
2.78	.04602		7.410	9.53	95.093	.7779	37.78	16.89
3.00	.05533	25.12	7.184	9.13	94.966	.7868	35.03	15.66
3.50	.08185	21.03	6.767	8.44	94.734	.8014	25.95	13.39
4.00	.1159	17.62	6.455	7.96	94.560	.8106	26.16	11.69
4.50	.1582	14.61	6.213	7.61	94.425	.8168	23.22	10.38
5.0	.2099	11.85	6.020	7.33	94.317	.8212	20.87	9.33
5.5	.2718	9.09	5.863	7.11	94.230	.8243	18.96	8.48
6.0	.3449	5.69	5.732	6.93	94.156	.8267	17.36	7.76
6.135	.3669	3.00	5.700	6.89	94.097	.8272	16.97	7.58
6.89	.4040	3.00	4.378	5.11	93.373	.8360	15.00	6.71
7.53	.4213	3.00	3.391	4.10	92.82	.8269	13.64	6.10
8.31	.4261	3.00	2.292	3.02	92.21	.7580	12.27	5.49
9.29	.4123	3.00	1.142	2.05	91.57	.5558	10.91	4.88
10.56	.3637	3.00	0.285	1.21	91.09	.2347	9.55	4.27

(1) η = (Generator Eff) x (Gearbox Eff)

Table 3-6 Annual Energy Output

$\bar{V}$ (1) mph	Annual Energy (2) kwh
8	8,732
9	12,207
10	15,676
12	22,041
14	27,156
16	30,740
18	32,801

(1) At 30 ft Elevation

(2) Corrected For 95% Availability & Control System Losses

4.0 CONTROL AND MECHANICAL SUBSYSTEM

4.1 HISTORICAL SUMMARY

4.1.1 Proposal

4.1.1.1 The Basic Principle

A mechanical speed control system, based on the cradle dynamometer principle is utilized (Figure 4-1). The generator (an induction motor) housing, is allowed to rotate against a spring and serves simultaneously as a torque sensor for the rotor and as a torquer, to actuate and adjust the pitch angle of the blades.

4.1.1.2 Explanation of the Principle

An induction motor, when driven slightly above its synchronous speed, generates directly into the power system to which it is connected (and from which its exitation is received). The frequency and wave form of the generated power, exactly matches that of the power system so no problem of synchronization exists.

The generator driving torque (and the power produced), changes considerably for a slight speed change. Thus the generator acts as a brake, allowing the turbine speed to exceed only slightly its rated speed so long as the generator breakdown torque is not exceeded. The rotation of the generator housing against the preset spring is used to sense the torque and adjust the pitch of the wind turbine blades, so that the torque and speed error is reduced to a minimum.

4.1.1.3 Proposed System

A two speed generator was proposed so that generation could begin at about 6 mph on the 900 rpm generator winding and be switched to 1800 rpm generator winding as wind speed exceeds 11 mph. Then rotor speed was to be maintained between 107 and 110 rpm by means of pitch angle adjustment.

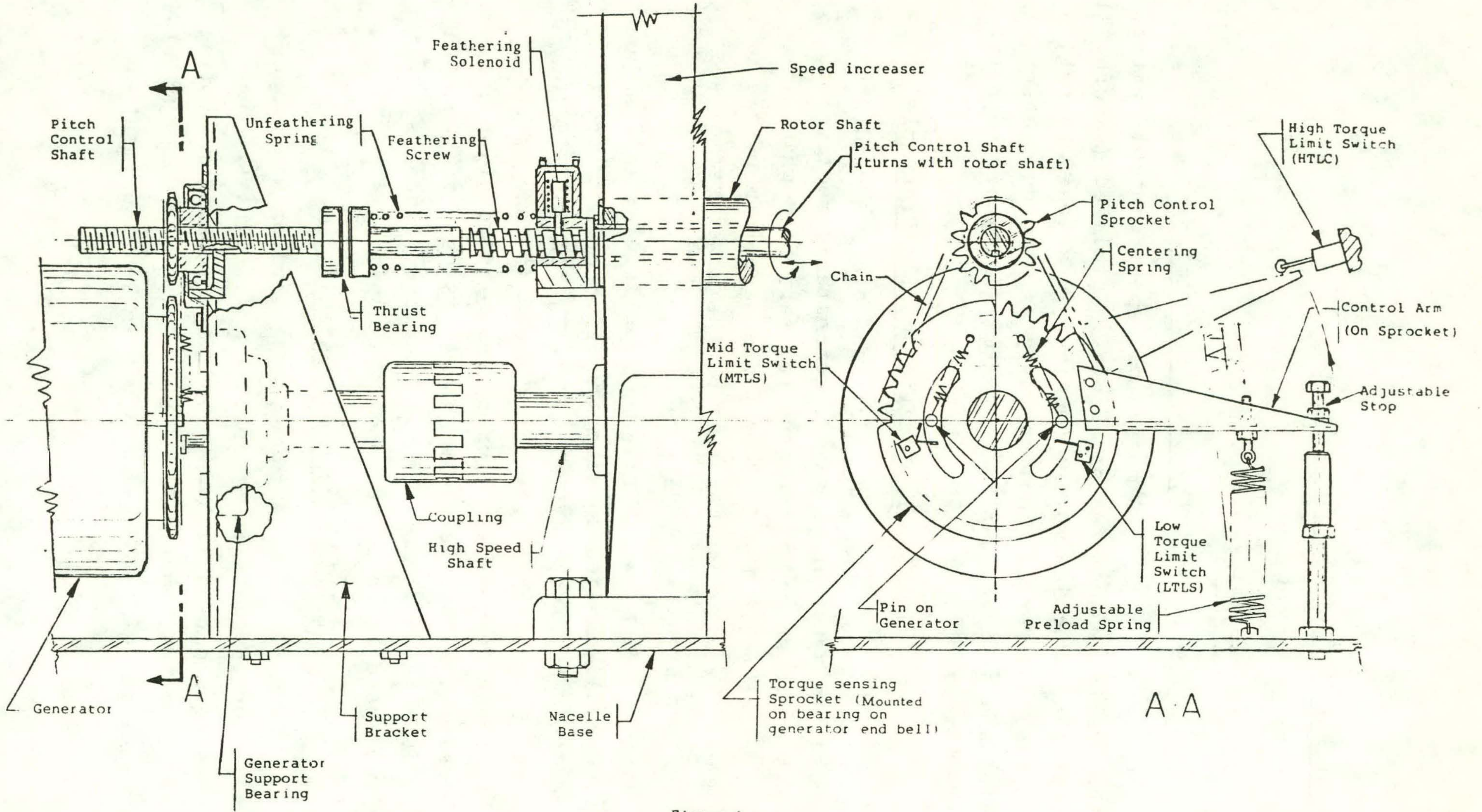


Figure 4 1

TORQUE SENSING AND EMERGENCY
FEATHERING SYSTEMS Proposal

Main Additional Control Features Were:

a. Start up of the system by unfeathering and motor start when minimum wind speed was reached.

b. Switching between the two generator speeds according to wind conditions.

c. Shut down of the system by feathering when:

  c.1 Excessive torque limit is reached (high wind speed).

  c.2 Utility power fails.

  c.3 Excessive vibration occurs.

  c.4 The rotor overspeed.

  c.5 Insufficient torque is developed (low wind speed - motoring instead of generating).

  c.6 Manually shut down.

4.1.2 PDR I

The control system was basically the same except for the start up method. Instead of fast unfeathering and motor start up, a pitch ramp gearmotor was added to control the unfeathering speed and allow an aerodynamic startup to reduce voltage flicker on the utility.

4.1.3 PDR II (Figure 4-2)

Same control principle but several changes were made in order to simplify the design for lower cost of energy.

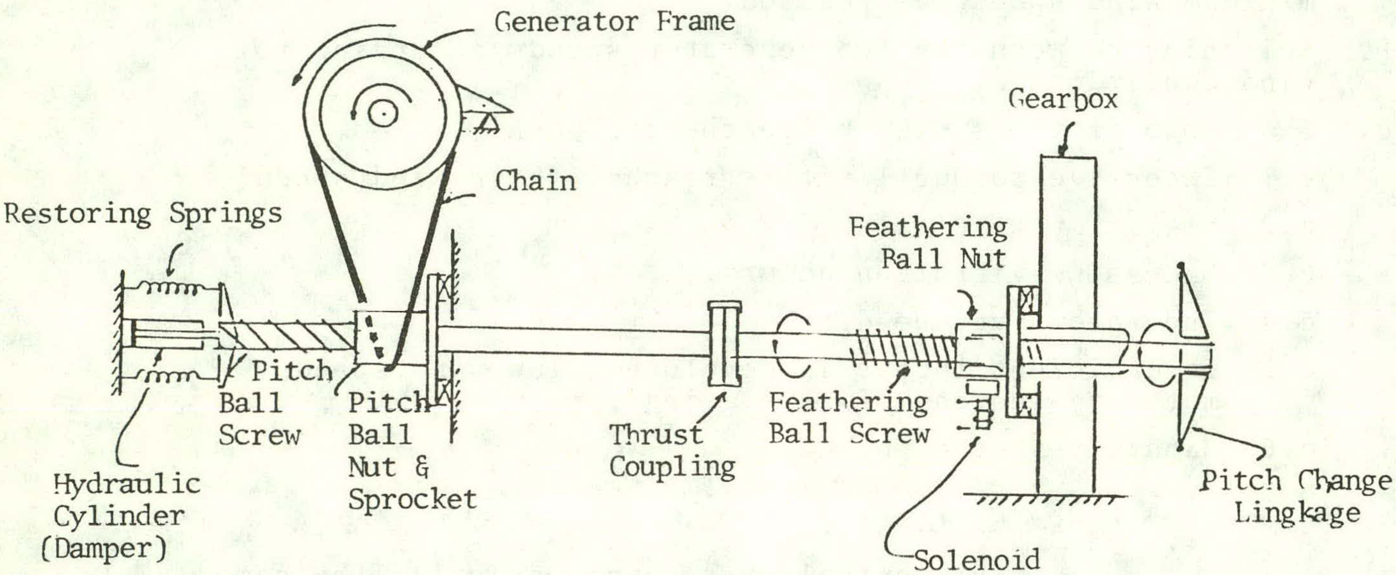
(a) The three phase (run as single phase) two speed generator was changed to single phase, one speed. Accordingly, all switching devices for two speed control were eliminated.

(b) The start up ramp rate was to be controlled by a double acting hydraulic cylinder serving also as control system damper thus replacing the ramp gearmotor.

(c) The restoring springs were relocated along the pitch control shaft.

(d) The generator rotation was defined to 140^{o} for the control phase and a full turn (360^{o}) for feathering.

Figure 4-2 CONTROL SYSTEM SCHEMATIC - PDR II



4.1.4 CDR (4)

4.1.4.1 Scope

A preliminary mechanical design of the nacelle was made and main design considerations were presented (18). The nacelle main design criteria were:

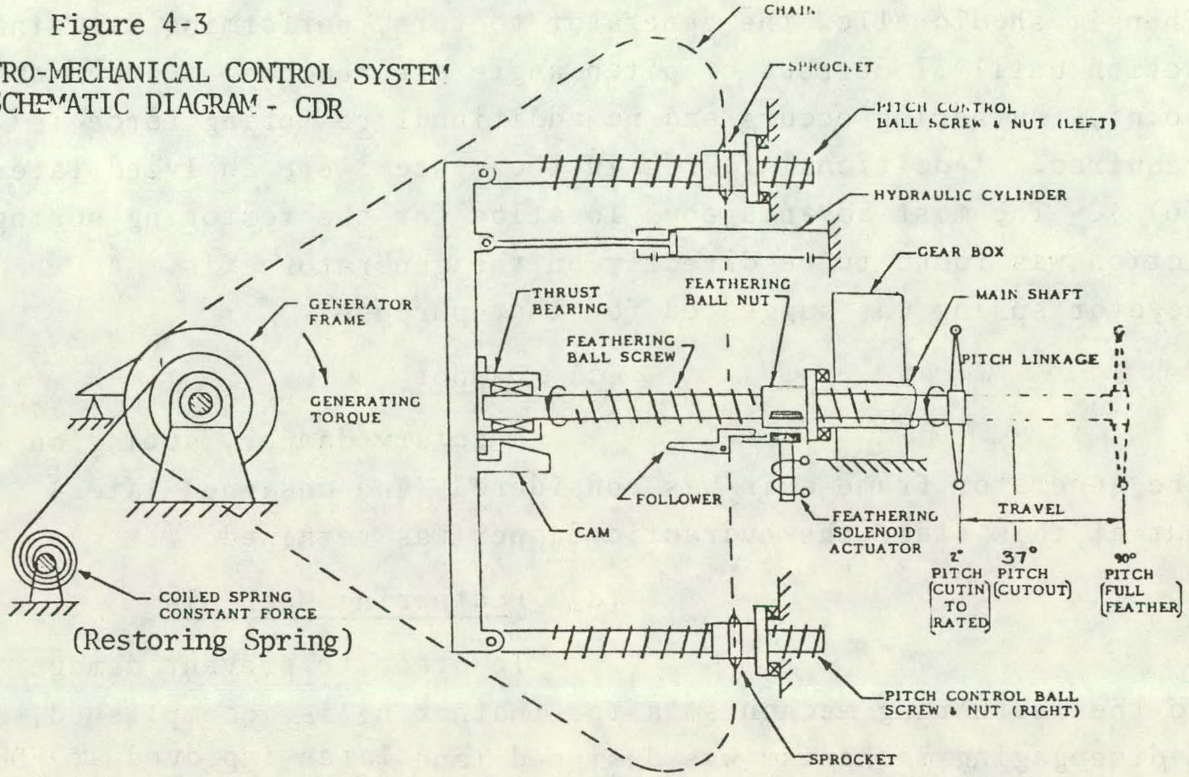
(a) Lightweight rugged structure.
(b) Minimum length, flexibility for yaw axis location versus center of gravity.
(c) Minimize friction and hysteresis in control mechanisms.

4.1.4.2 Control System - Nacelle Configuration

In the design presented at PDR II there were three elements in line (pitch ball screw, feathering ball screw and damper), performing a linear displacement of 7 inches so the displacements and mounting spaces (dead zones, fittings, etc.) added up to a considerable length.

A design, relocating all tracks in parallel was presented (Figure 4-3 and 4-4).

Figure 4-3
ELECTRO-MECHANICAL CONTROL SYSTEM
SCHEMATIC DIAGRAM - CDR



4.1.4.3 Main Design Considerations Control System

(a) Main Control Parameters

Pitch rod travel - 8 in.

Feathering ball screw lead $L_f = 1$ in.

Pitch ball screw lead - $L = 1$ in.

Pitch sprocket - $r_s = 14$ teeth

Generator sprocket - $R_g = 112$ teeth

Overall ratio $i = \frac{r_s}{L \times R_g} = 0.125 \left[\text{inch}^{-1}\right]$

$$\frac{\text{Generator frame angle}}{\text{Pitch Angle}} = 4.1$$

(b) Restoring Spring

The restoring spring holds the generator against a stop until nominal torque is reached. Then it should allow the generator to turn, performing pitching action until 37 degrees of pitch angle are reached. Beyond that point, feathering occurs and no additional restoring force is required. (Additional forces in the system were analyzed later (6)). The most advantageous location for the restoring spring action was found to be directly on the generator axis. A negator spring was suggested for that purpose.

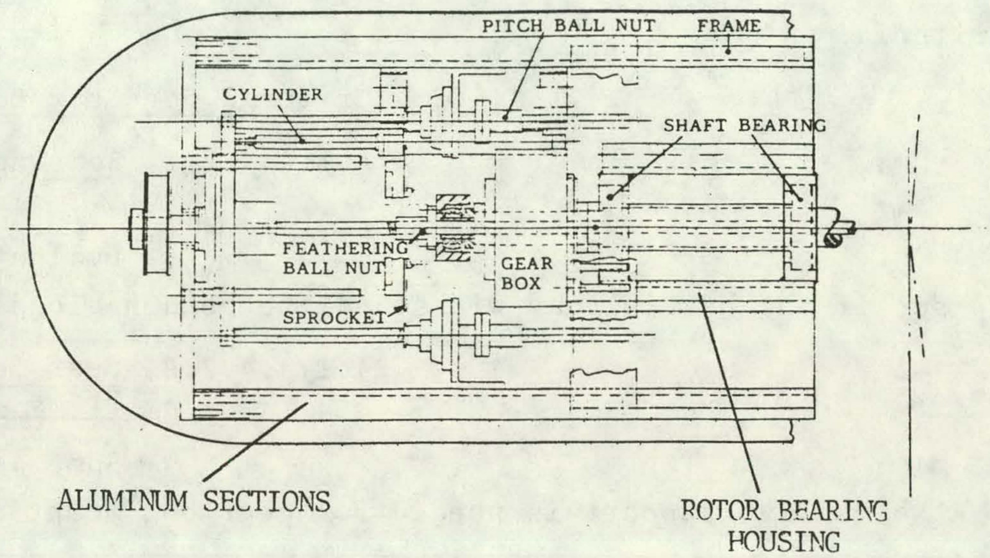
(c) Damper

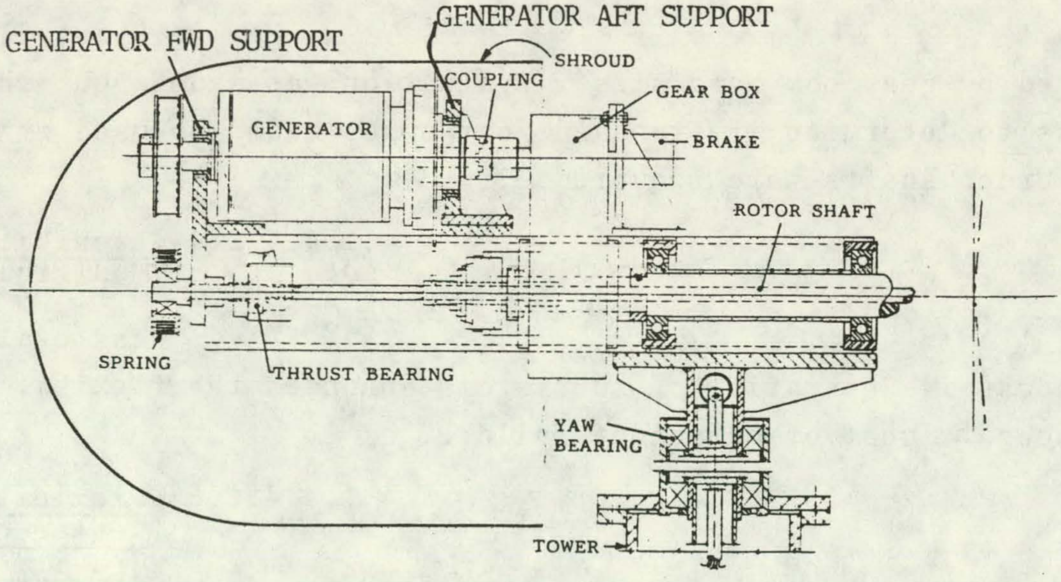
A rotary damper, acting on the generator frame axis was considered (and designed later) but at this stage the hydraulic damper was retained.

(d) Feathering Mechanism

In order to prevent damage to the feathering mechanism after feathering is accomplished, a disengaging mechanism was designed (and later improved at FDP).

Figure 4-4 NACELLE CONFIGURATION - CDR





4.1.4.4 Gear Box Structure Trade Offs

4.1.4.4.1 Scope

A hollow shaft is needed in gear box. Standard gear boxes are designed to withstand considerable torques (as rated) but only small overhung loads. The bending moment acting on the rotor shaft was evaluated to be 5100 ft-lbs. Eight cases were considered and evaluated against the same criteria.

4.1.4.4.2 Criteria For Gear Box Choice

4.1.4.4.2.1 Bearing Loads

Evaluation was made to see if the bearing can withstand the bending loads.

4.1.4.4.2.2 Gear Box Frame Loading

Same as for bearings. Also massive support is needed for gear box mounting.

4.1.4.4.2.3 Shaft Stiffness

Shaft diameter is limited by gear box bore size. Evaluation was made about shaft stiffness to determine if standard gear boxes could be used or if special order shafts were required.

4.1.4.4.2.4 Flexibility For Modifications

Was evaluated for each case. Separation of subsystems enhances the flexibility and reduces the cost of component failure.

4.1.4.4.2.5 Flexibility For Center of Gravity Location

Was evaluated for each case - such flexibility is important in prototypes.

4.1.4.4.2.6 Price/Special Order/Standard

For small quantities special order will be very expensive and is to be considered only if major advantages are realized.

4.1.4.4.2.7 Maintenance Access

Ease of access considerations were evaluated for each case.

4.1.4.4.3 GEAR BOX TYPES CONSIDERED
(See Sketches in Table 4-1)

Type a. Standard Hollow shaft gear box hub mounted on shaft located into the bore. Gear box fastened by its frame, supporting all bending loads.

Remarks 1) Neither bearings nor gear box frame can withstand the bending loads.
2) Shaft size is limited by bore.

Type b. Modified gear box hub mounted directly on one piece extended shaft. Gear fastening same as a.

Remarks 1) Same as a.1.
2) Special order.

Type c. Standard gear box with rotor shaft supported by two bearings directly to yaw housing. Gear box mounted on rotor shaft with rotation prevented by a torque linkage.

Remarks 1) Structural stresses concentrated in small space.

Type d. Modified gear same mounting as c but using one piece shaft as in b.

Type e. Standard gear box, basically same as c but supplied on both sides.

Remarks 1) Limited shaft diameter.
2) Complicated assembly.

Type f. Modified gear box, same as e but with integral shaft.

Type g. Standard gear box, foot mounted, connected with a coupling to shaft. Rotor shaft mounted as in c.

Remarks 1) High flexibility for modifications.
2) Alignment problems because of pitch rod.
3) Complex mounting.

Type h. Special design foot mounted gear box providing sufficient frame and shaft stiffness for withstanding external loads.

Remarks 1) Best option.
2) Due to high tooling cost and schedule, this option is to be considered for production only.

4.1.3.4.4 Summary

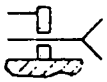
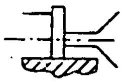
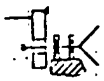
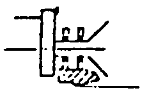
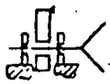
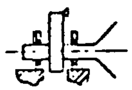
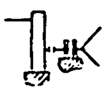
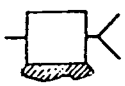
After evaluating pros and cons, type c was chosen for prototype. Spur gear was selected because of availability problems. Only h looks better but is too expensive and not recommended for prototype.

Gear box will not be used as a structural element.

4.1.4.5 Structure (Figure 4-4)

A rugged structure was selected, based on longitudinal aluminum sections, fastened by intermediate supports (generator supports) and the rotor bearings housing. The yaw bearing is fastened close to the bearings housing providing a central compact rugged structural element for the nacelle. This also provides flexibility for yaw axis location. (The sliprings were located in the tower near the base, providing access when the tower is lowered).

Table 4-1 GEAR BOX TRADE OFF SUMMARY MATRIX

TYPE	CRITERIA / DESCRIPTION	Gear Box Bearing Loads	Gear Box Frame Load	Shaft Stiffness	Frame	Flexibility For Mod & C.G.	Price Special Order/ST	Maintenance Access And Adjustment	REMARKS
a		Very High	Very High	Low	Complex & Massive	Good	Std. Cheap	Good	Unaccep-table
b		Very High	Very High	High	Complex & Massive	Poor	Sp. Order Expensive	Fair	Unaccep-table
c		None	Low	High	Simple & Light	Very Good	Std. Cheap	Good	Chosen For Prototype
d		None	Low	High	Simple & Light	Poor	Sp. Order Expensive	Fair	
e		None	Low	Low	Complex	Good	Std. Cheap	Poor	
f		None	Low	High	Complex	Poor	Sp. Order Expensive	Poor	
g		None	Low	High	Complex & Massive	Good	Sp. Order Expensive Frame	Poor	
h		Moderate	Moderate	High	Simple & Light	Poor	Sp. Order Very Expensive	Good	To Consider For Production

4.2 FINAL DESIGN (FDR)

4.2.1 Geometrical Configuration (Figure 4-8)

After the main components have been sized, it was found that there is no more justification for parallel configuration as presented at C.D.R. since the nacelle length was dictated by the upper level including the generator, the damper, the spring and the sliprings.

On the lower level, the pitch and feathering ball screws and accessories were located as a "dust protected compartment" to insure good performance of ball screws.

4.2.2 Control Operation Description (See Figure 4-5)

4.2.2.1 Start Up Operation

4.2.2.1.1 Start up begins from feathered position.

4.2.2.1.2 Signal To Start Conditions

(a) Following normal stop, a signal to start is received whenever abnormal conditions are automatically corrected (eg - overheated generator cools adequately) and wind speed is maintained between cut-in and cut-out range for a preset period of time.

(b) Following emergency stop, a signal to start is received in the same manner, except a reset pushbutton must first be manually actuated.

4.2.2.1.3 Solenoid actuators are energized, pulled back, thus releasing feathering ball nut, (if not already mechanically disengaged) and ratchet on pitch ball nut. That position is maintained as long as the system is in operation.

4.2.2.1.4 Charged spring drives system from full feather to normal operating pitch in several steps controlled by action of solenoid actuator which allows multiple 15 degrees pitch changes, each followed by a preset delay period.

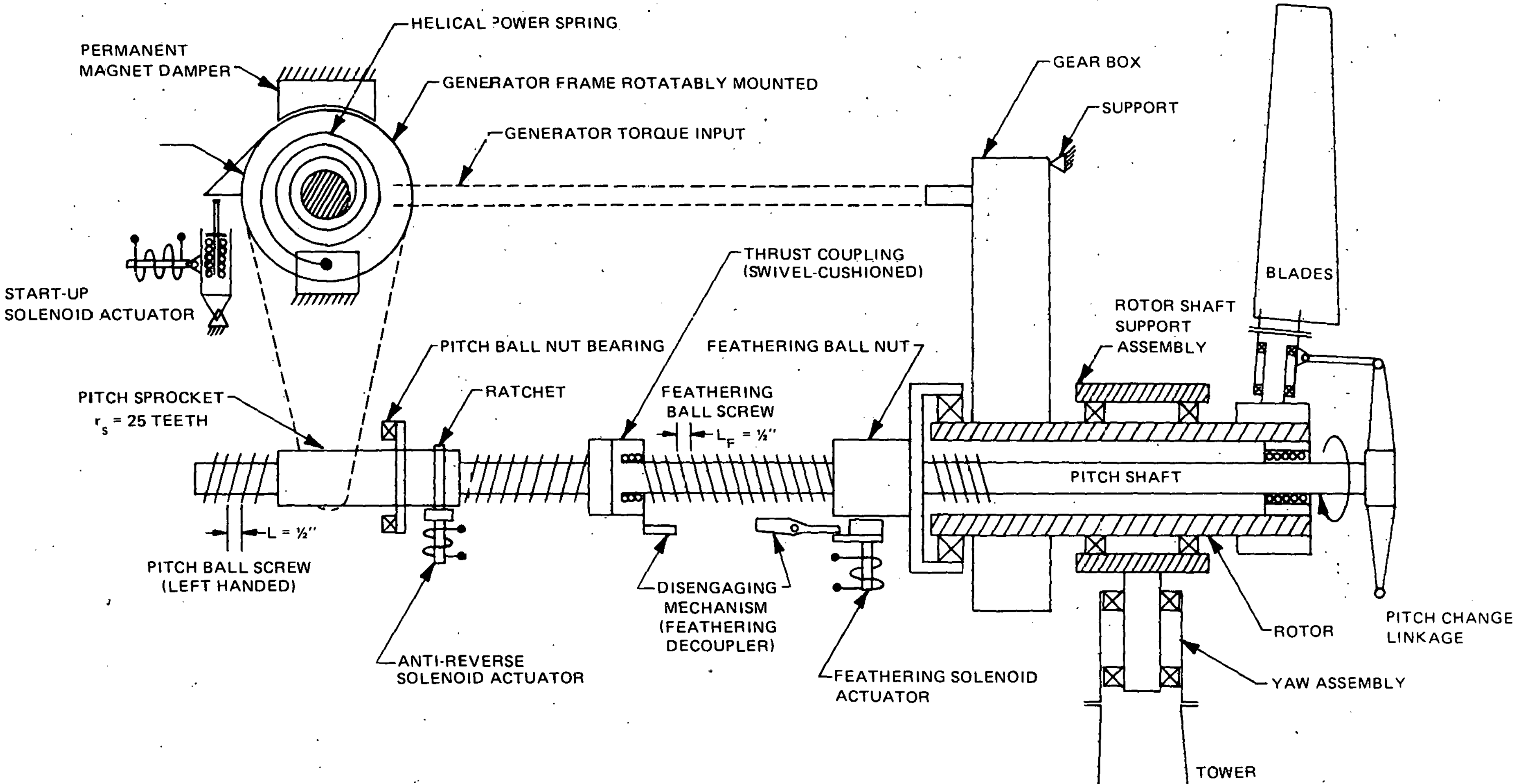


Figure 4-5 ELECTROMECHANICAL CONTROL SYSTEM SCHEMATIC DIAGRAM - FDR

4.2.2.1.5 When the rotor attains operating speed, the generator is energized and the system shifts to control phase.

4.2.2.1.6 If operating speed is not attained within a preset time, start up is aborted (see 4.2.2.4).

4.2.2.2 <u>Control Phase</u>

4.2.2.2.1 As the generator is energized, automatic pitch control is maintained by the system in the following manner.

4.2.2.2.2 <u>Generator Angular Position</u>

(a) The generator is mounted so that its frame is free to rotate. The frame axis is coupled to a helical spring to a damper and to sliprings.

(b) The induction generator typical characteristic is a considerable torque (and power output) variation for a small variation in speed (30% for 15 rpm difference around 1850 rpm).

(c) Angular position of the generator frame is defined by the equilibrium of the generator reaction torque on its frame and the counter balancing spring.

4.2.2.2.3 <u>Damping</u>

Damping of the rotation is provided by a permanent magnet rotary damper coupled to the generator frame.

4.2.2.2.4 <u>Electric Power Transfer</u>

Since the generator rotates about six turns, the electric power is transferred through sliprings.

4.2.2.2.5 <u>Pitch Position</u>

The generator frame position is transferred through a chain, ball nut and ball screw to the pitch horn. The pitch horn longitudinal position defines the pitch angle of the blades.

4.2.2.3 Feathering

4.2.2.3.1 Feathering Signal "Normal" - Automatic Restart

(a) High wind speed cut-out – When wind speed exceeds a preset value the system feathers to protect itself from overstress.

(b) Low wind speed - When wind speed falls below a present value the system feathers because it is useless to keep it operating. That wind speed is defined as cut-in speed.

(c) High temperature or overload - The system feathers to protect itself.

(d) If feathering occurs because of the reasons mentioned above, automatic restart is possible when appropriate conditions are met.

Remark: Feathering does not require electrical power.

4.2.2.3.2 Feathering Signal - "Emergency" - Manual Restart

(a) Manual - Whenever the operator wants to shut down the system.

(b) Loss of power or voltage.

(c) Overspeed - When the rotor speed exceeds a preset value.

(d) Vibration - When excessive vibration caused by unbalanced rotor (icing or mechanical failure) or other reason, the system feathers.

(e) Because of the nature of the originating signals mentioned above, manual restart is required.

4.2.2.3.3 Feathering Sequence

(a) Signal for shutdown (normal or emergency).

(b) Solenoids are de-energized.

(c) Failsafe solenoid actuator lever engages feathering ball nut.

(d) Anti-reverse solenoid actuator lever "leans" on the ratchet.

(e) Momentum of the rotor causes feathering ball screw to thread itself toward the full feathered position.

(f) The feathering ball screw pushes the pitch horn outward and causes the blade to rotate toward feathered position.

(g) The feathering ball screw (rotating) pulls the pitch ball screw (non-rotating) through a swivel joint.

(h) The pitch ball screw backdrives the pitch ball nut and the sprocket.

(i) The pitch ball nut rotation is transferred by a chain to a sprocket coupled to the generator frame.

(j) The generator frame rotates and charges the spring.

(k) As fully feathered condition is achieved, a cam actuated by the swivel position disengages the feathering actuator lever to terminate feathering.

(l) If feathering is not achieved within a preset time, the generator can be operated as a motor to ensure complete feathering.

(m) The anti-reverse actuator lever engages and prevents the spring loaded system from pulling back.

(n) With the feathering actuator mechanically lifted and the anti-reverse actuator engaged, the system is "free-wheeling" in the sense that the feathered rotor may be driven by the wind in a forward or reverse direction with no effect on the feathering system.

In that position, all stresses are released from the ball nuts and only the chain is loaded through the ratchet and prevents the spring from returning blade pitch to operating position as long as actuators remain de-energized.

(o) The levers configuration assures that even if feathering was not completely achieved, clockwise rotation of the rotor will cause the feathering sequence to proceed, and in non-rotation or anti-clockwise rotation of the rotor, the "last" position of the blades will be maintained.

4.2.2.4 <u>Aborted Start</u>

If turbine does not achieve operating speed within a preset time, the feathering solenoids are again de-energized to initiate feathering.

At the same time, the generator is operated as a motor to insure that blades will be completely feathered. Without such motor operation, it is possible that the rotor will not reach adequate rotational speed to achieve full feathering on its own.

4.2.2.5 <u>Main Control Parameters (Figure 4-5)</u>

Pitch rod travel - 8 in.

Feathering ball screw lead - L_F = 1/2 in.

Pitch ball screw lead - L = 1/2 in.

Pitch sprocket r_s - 25 teeth

Generator sprocket R_g - 70 teeth

Overall ratio $i = \frac{r_s}{L.R_g} = 0.714\ (\text{in.}^{-1})$

$\frac{\text{General Frame Angle}}{\text{Pitch Angle}} = 23.4$

Generator characteristic - See Figure 4-6

Spring characteristic requirement - See Figure 4-6

Damper value - 1 in. lb/rpm

Remark: From pure control considerations, as shown later, the larger i, the better performance is achieved in control accuracy and reduction of hysteresis effects and pitch moment effects. However, a large i affects the backdriving in the feathering phase due to high acceleration applied to the generator frame and to low lead angles on the pitch ball screw. For i - 0.75 and lead of 0.5 in. in feathering ball screw, an acceptable compromise is achieved.

4.3 CONTROL SYSTEM ANALYSIS SUMMARY (6, 7)

4.3.1 Scope

The pitch control system uses the generator frame as torque sensor and as a torquer.

The pitch moment forces are fed back to the generator frame through the pitch rod and considerable friction forces are involved. (See also flow sheet - Figure 4-7)

An analysis was performed in order to evaluate the sensitivity of the system to various parameters such as friction, inertia, pitch moment loads and torque derivatives relative to wind and pitch position.

Hysteresis was also evaluated and was found to be large and unpredictable for the parameters described in 4.1.3.3.

Evaluation was made for several ratios (i) in order to find a compromise between accuracy, mechanical encumbrance and acceptable feathering rate.

4.3.2 Method

4.3.2.1 Loads and Derivatives

Loads and derivatives were collected from various references into tables to facilitate the analysis.

4.3.2.2 Friction

Friction was evaluated, taking into consideration various bearing configurations.

4.3.2.3 Pitch Control Kinematic

Pitch control kinematic functions were developed and a table of kinematic relations between pitch angle, travel and generator rotation was set for several ratios.

4.3.2.4 Load Path

Load path was evaluated from the blades (following flow sheet Figure 4-7) through all bearings, linkages and friction surfaces, ball nuts and chain to the generator frame.

The loads in terms of torque on the generator frame were computed for two directions (from blades point of view).

(1) When generator frame is "driving" against all friction forces and pitch moments (by reducing electrical torque, the spring drives the frame) and causes pitch angle to decrease (DC). (See also Figure 4-6)

(2) When generator frame is "driven" (by gaining electrical torque overcoming the spring) and "let" the pitch angle to be increased by the wind (IN). (See also Figure 4-6)

4.3.2.5 Pitch Moment Derivatives

Pitch moment derivatives with respect to wind and pitch angle position were computed in terms of generator frame torque derivatives for both directions DC and IN in order to have a common computation base.

4.3.2.6 Rotor Torque Derivatives

Rotor torque derivatives with respect to wind and pitch angle position were translated through the gear box and appropriate efficiencies into terms of generator frame torque derivatives.

4.3.2.7 Inertia Values

Inertia values of all major moving elements were evaluated and translated (through appropriate ratios) to generator axis in two groups.

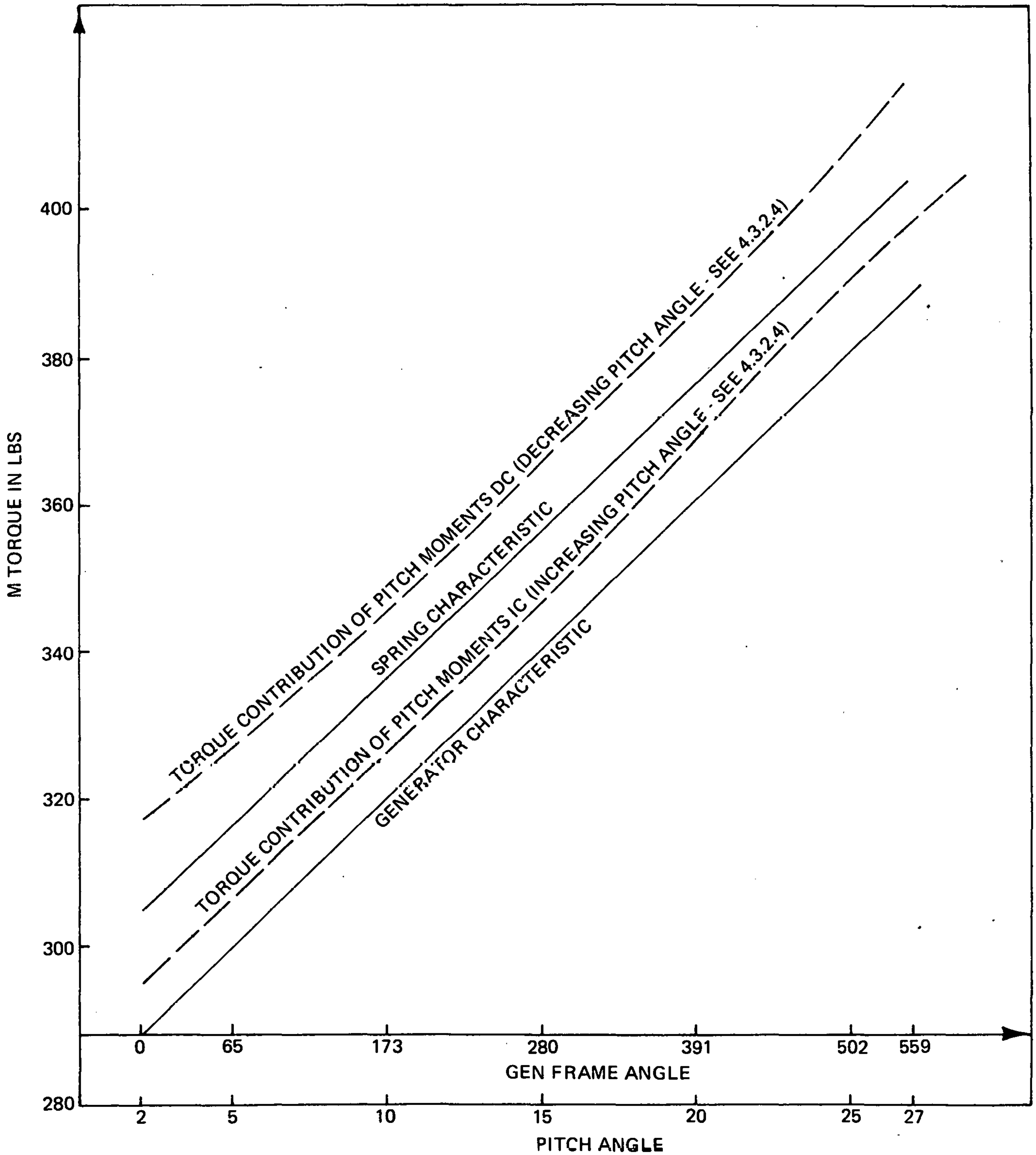
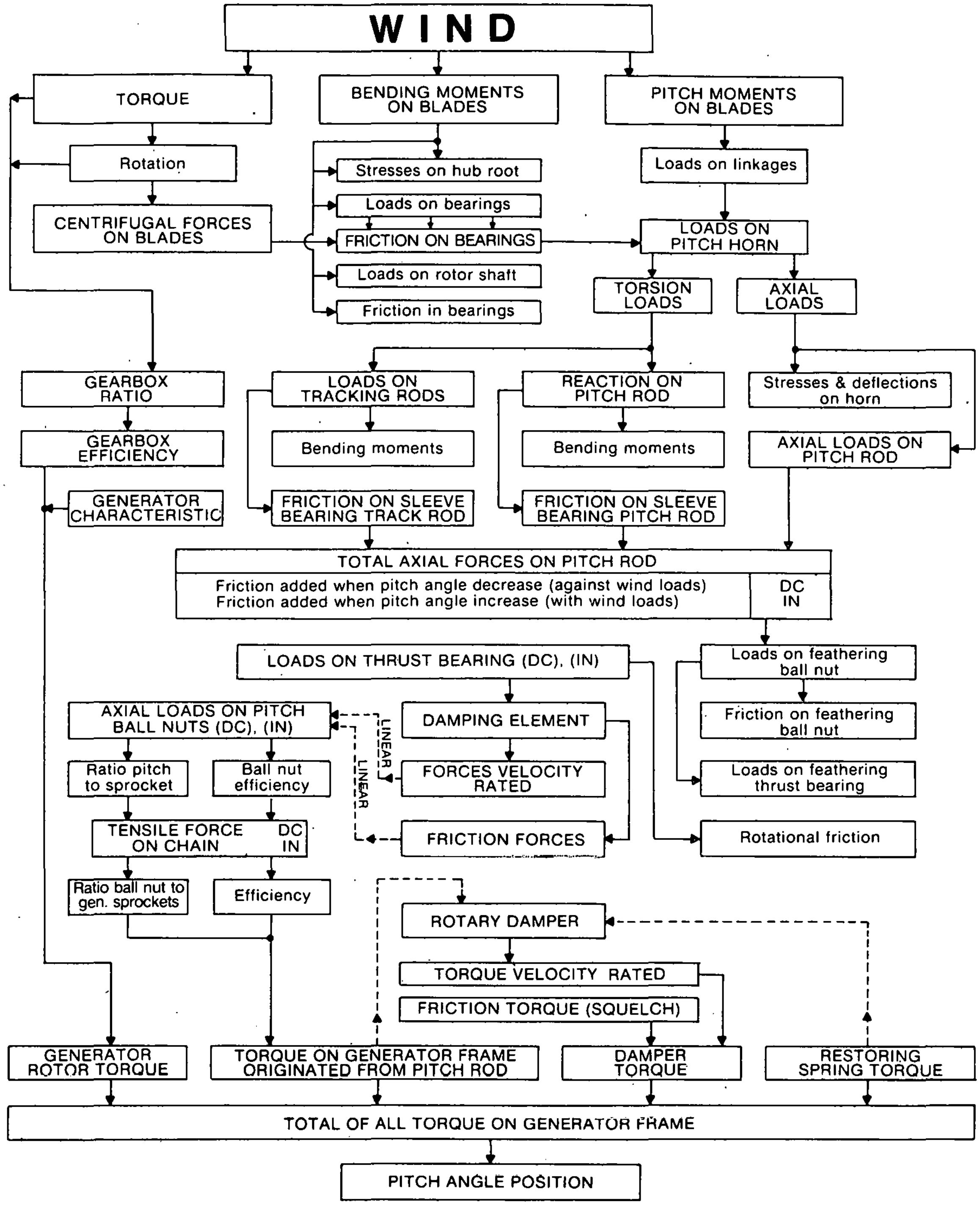


Figure 4-6 GENERATOR CHARACTERISTICS & SPRING CHARACTERISTIC REQUIREMENT
(Torque Contributions of Pitch Moment and Friction Effects.)

Figure 4-7 CONTROL SYSTEM FLOWSHEET



(1) High Speed Group

Rotor, hub, gears and generator rotor inertias translated to generator rotor axis were added to represent total inertias accelerated directly by the wind.

From those values, rotational acceleration derivative was computed.

(2) Frame Group

Blades, hub, linear moving masses, ball nuts and chain, generator frame with adaptors and sprockets inertias, translated to generator frame, were added to represent total inertias accelerated by generator frame torques (originated by electrical torques, pitch moments and restoring spring).

4.3.2.8 Generator Rotor Acceleration

Generator rotor acceleration function was evaluated. Wind torque derivatives were found to be considerable. Generator rotor acceleration causes increase in electrical torque which is opposed to the acceleration. The solution of the differential equation shows an exponential decrease in acceleration with time constant of 0.14 second.

4.3.3 Analysis and Discussion of Findings

4.3.3.1 Bearings

Blade bearing friction was found to be considerable, mainly because of the centrifugal forces. Best solution was to use angular contact ball bearings.

In order to reduce the friction on the pitch control rod, linear ball bearings were selected.

4.3.3.2 Pitch Kinematics

Duc to appropriate selection of angle range and geometrical design, a good linearity between blade pitch angle and the pitch rod, linear translation was achieved. Loads were also reduced to a minimum.

4.3.3.3 Loads On Pitch Rod

Since blade pitch axis is not located close to the aerodynamic center of pressure, large torsional moments are applied on the control axis and significant forces are needed to perform the control due to direct loads applied and parasitic friction forces originating from that effect.

Due to friction, there are considerable differences in forces needed to increase the pitch angle and to decrease it.

Those forces are transmitted through the ball screws to the generator frame and affect the accuracy of the frame as a sensor divice by introducing a hysteresis effect. That load variation is intensified by the pitch ball screw efficiency (assumed 85% for "good conditioned" ball screws).

4.3.3.4 Torque Derivatives

Rotor torque derivatives (on generator frame) were found to be considerable and significant, with increasing values in higher winds.

Pitch moment derivatives were found to be insignificant for first approach.

The control system was expected to be extremely sensitive in high winds and this led to reduction of cut-out speed and to a design of a damper on the generator frame axis.

4.3.4 Major Changes in the Control System

Following the system analysis, the following changes have been made:

4.3.4.1 Mechanical Advantage

In order to improve the mechanical advantage, the ratio between the generator frame and pitch rod were changed from i = 0.125 (as in CDR) to i = 0.714 ($in.^{-1}$).

The generator power range was enlarged from 10% over rated to 30% (so a larger torque reaction was available).

By those changes the controlling torque available was increased by a factor of 17 and the hysteresis, reduced appropriately.

4.3.4.2 Sliprings

Since the changes cause the generator frame to turn 6 revolutions (from start up to feathering), the electric power from the generator was transferred through sliprings instead of a reeled cable.

4.3.4.3 Damper

A magnetic rotary damper (based on eddy currents), coupled directly to the generator frame, was designed (see 4.4.4).

4.3.4.4 Spring

A power spring, coupled to the generator frame shaft was designed. By that method, the restoring moment was measured directly and the hysteresis was reduced practically to that of the spring only (see 4.4.5).

4.3.4.5 Cut-Out Speed

Since the control system appeared to be "nervous" in high wind speeds, cut-out wind speed was reduced from 50 mph to 35 mph (without significant loss in energy).

4.3.5 Control System Dynamic Performance

4.3.5.1 Model

The dynamic analysis was based on a 3 degrees of freedom model (RPM, pitch angle and generator frame rotation), Coulomb friction was also considered.

4.3.5.2 Conditions Analyzed

A 3.4 mph gust step function was analyzed for initial wind speed values of 15.5 mph (rated), 23 mph and 35 mph (cut-out).

4.3.5.3 Conclusions

(a) The system was found to be stable and slightly oscillatory. The natural frequency is from 0.3 Hz near rated to 0.6 Hz near cut-out.

(b) The gust response is excellent. Maximum allowable gust (step function) is from 15 mph at rated speed to 21 mph near cut-out (providing that total wind speed should not exceed cut-out value (35 mph) for more than a few seconds - otherwise shut down will be initiated).

(c) Speed control is performed with a standard deviation of less than 2% through operating range.

4.4 MECHANICAL DETAILS

4.4.1 General Configuration

An aluminum structure, based on longitudinal channel sections fastened by intermediate supports, is described in Figure 4-8. The steel rotor shaft support, serving as a main structural element, is fastened to the channels and to the yaw bearing assembly (Figure 4-10), providing a rugged structure.

The packaging configuration consists of two levels.

(a) The upper level contains the generator, the magnetic damper, the restoring spring, the sliprings and the mechanical supports.

(b) The lower level contains the pitch control system which include the pitch and feathering ball nuts and screws and all the mechanisms needed for feathering, and feathering decoupling. The entire mechanism is dust protected by the channel sections and covers.

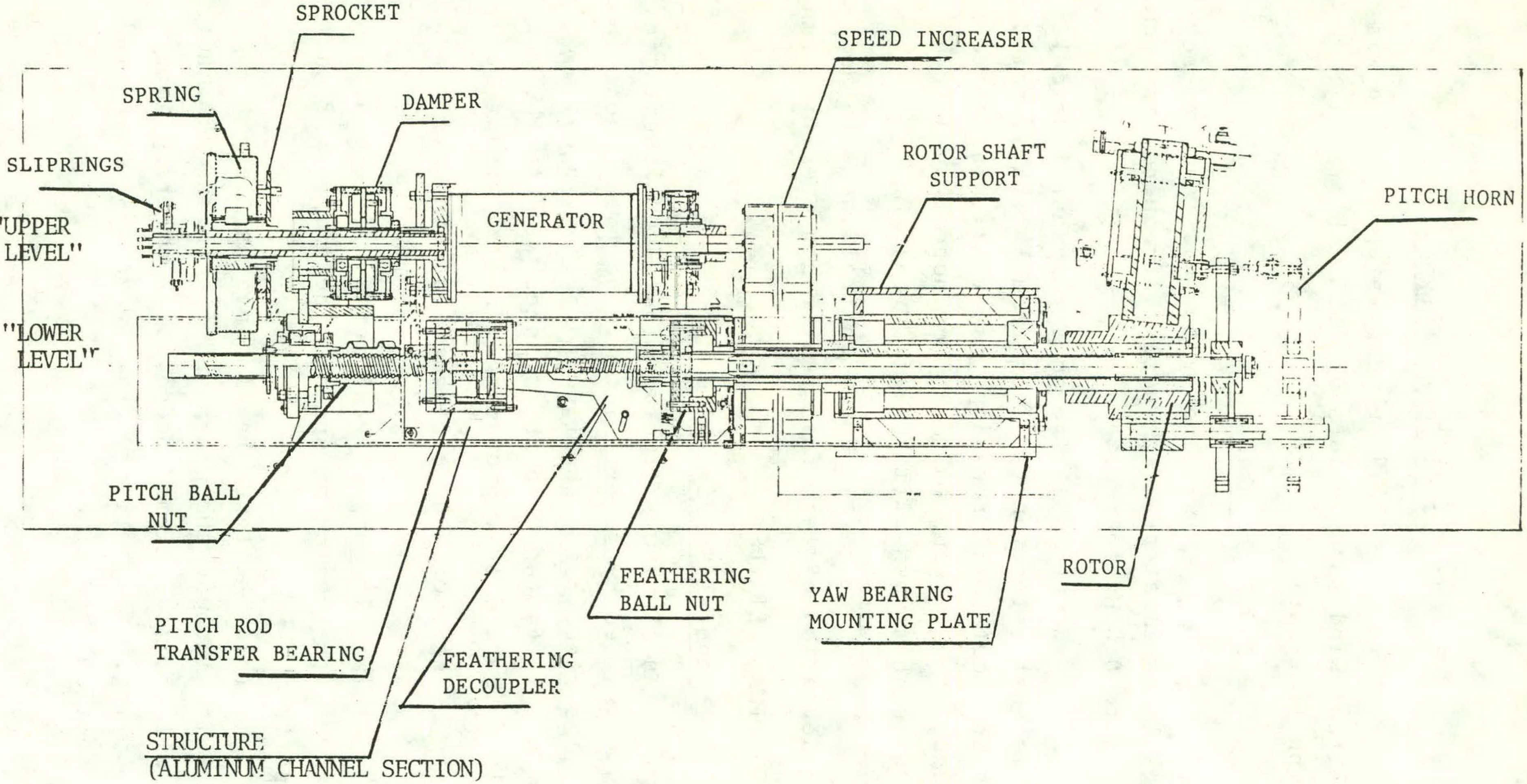


Figure 4-8 MECHANICAL SUBSYSTEM ASSEMBLY - FDR

The motion between the generator frame and the pitch control mechanism is transmitted by a chain and sprockets while the power from the rotor is transmitted to the generator through a speed increaser (gear box).

4.4.2 Pitch Rod Transfer Bearing (Figure 4-9)

That element serves as a swivel and transmits the linear motion of the feathering ball screw (rotating) to which the pitch horn is connected, to the pitch ball screw (non-rotating). The housing rotation is prevented by stabilizer arms. In feathering sequences, it travels toward the gear box, the stabilizer arm disengages the feathering actuator as feathering position is reached. In unfeathering sequences, it travels toward the pitch ball nut. The stabilizer de-activates the shock stop microswitch (See Chapter 5). The camplate, mounted on the pitch rod bearing assembly, secures the shock stop to engaging position (see also 4.2.2.3.3). Abrupt shocks due to sudden acceleration are prevented by the belleville springs.

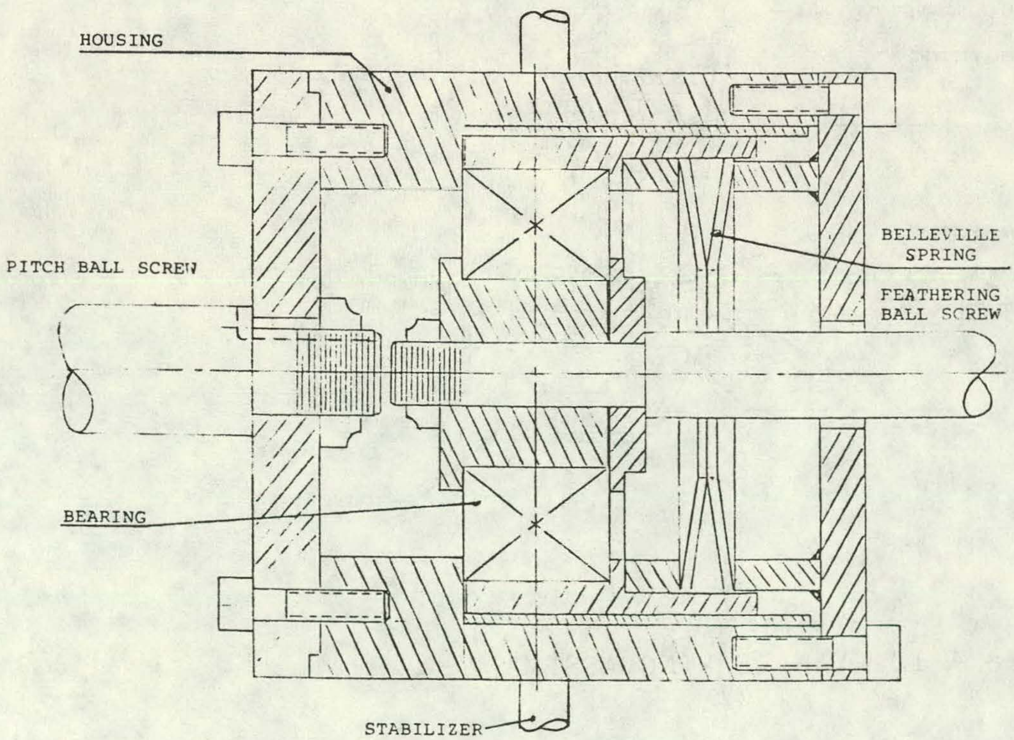


Figure 4-9 PITCH ROD TRANSFER BEARING

4.4.3 Yaw Bearing Assembly (Figure 4-10)

The yaw bearing assembly provides a low friction rotational axis to the system, to align itself with the prevailing wind. The design is quite simple and provides a very strong and rugged structure.

A flanged vertical shaft is bolted to the horizontal interface on top of the tower. The shaft supports two large angular contact bearings, selected because of their good radial and axial carrying capacity and low friction.

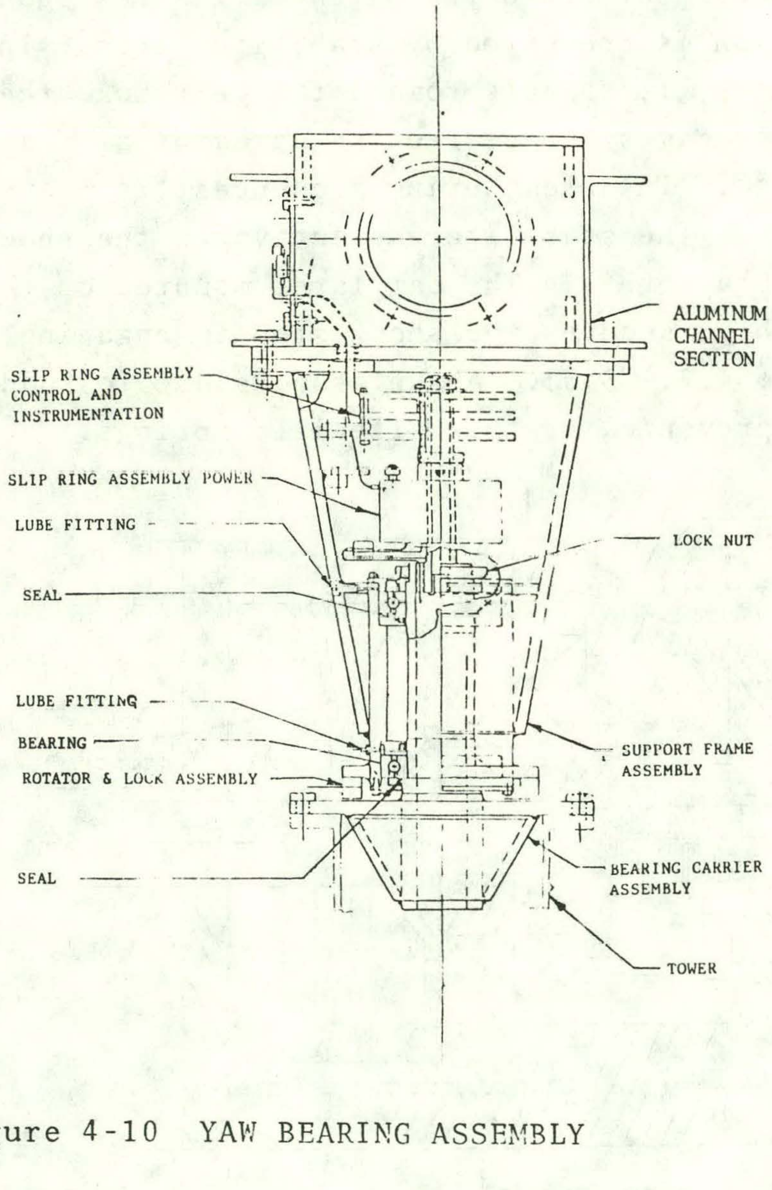


Figure 4-10 YAW BEARING ASSEMBLY

The bearings support the support frame assembly. The upper surface of the support frame assembly bolts to the mechanical sub-system, providing the final connection with the tower.

The support frame assembly is an aluminum tubular bearing support, welded into a conical, rolled aluminum plate section topped by an alunimum plate at the mechanical sub-system interface.

The yaw assembly provides room for sliprings. There are two slipring assemblies.

(a) The slipring assembly for power.

(b) Slipring assembly for control and instrumentation.

The instrumentation section is for the test phase and can be deleted for production.

4.4.4 Magnetic Damper Assembly

In order to stabilize the control system, a magnetic damper was designed. The damper consists of two large ring magnets arranged parallel to each other, coaxial with the generator frame and fastened to the shaft. A cooper disc is fastened to the outer case ring and there is an air gap between the two ring magnets. The magnets are magnetized with 16 north/south fields around the facial perimeter. As the generator frame rotates due to a change in torque, the shaft will also rotate the magnets relatively to the cooper disc. This action produces eddy current losses in the cooper disc and the counter torque results in a braking action to the initial angular acceleration, which is proportioned to the shaft angular velocity. (Figure 4-11)

Figure 4-11 MAGNETIC DAMPER DESIGN PARAMETERS

T_{ec} (IN-LB) = $2.27 \times (10)^{-3} \times \mathcal{B}^2 A t R^2 S$

WHERE:

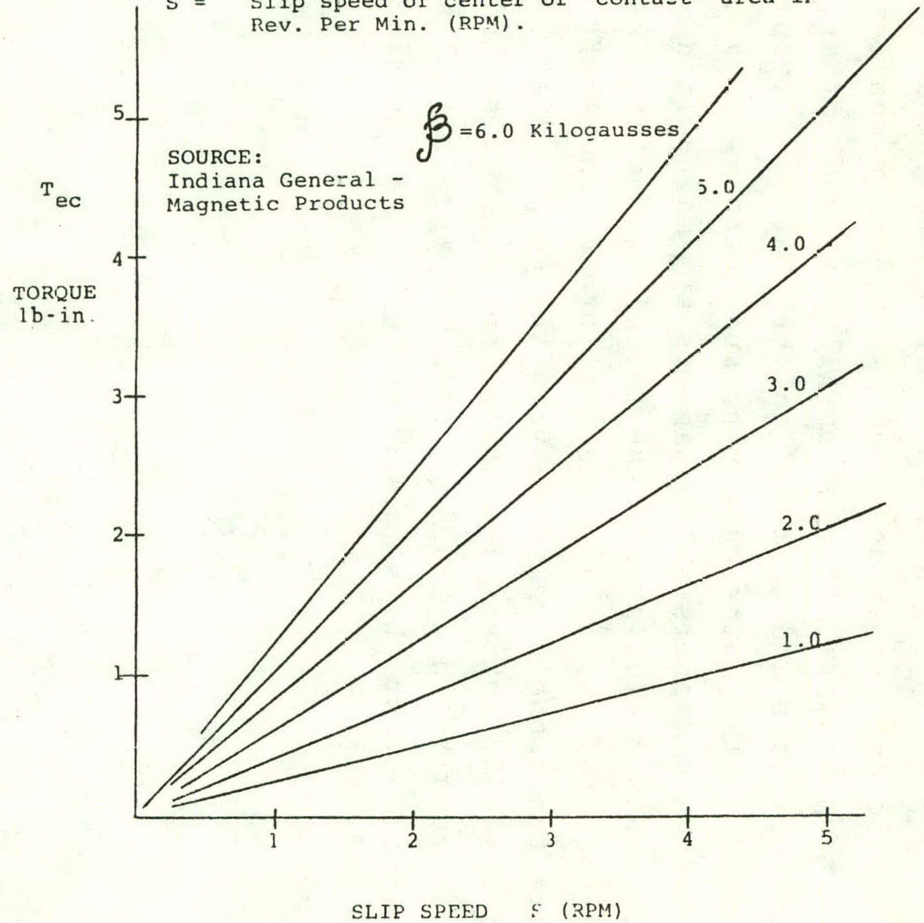
$\mathcal{B}$ Is mean flux density in kilogausses.

A = Area covered by magnet (one side looking through) in square inches.

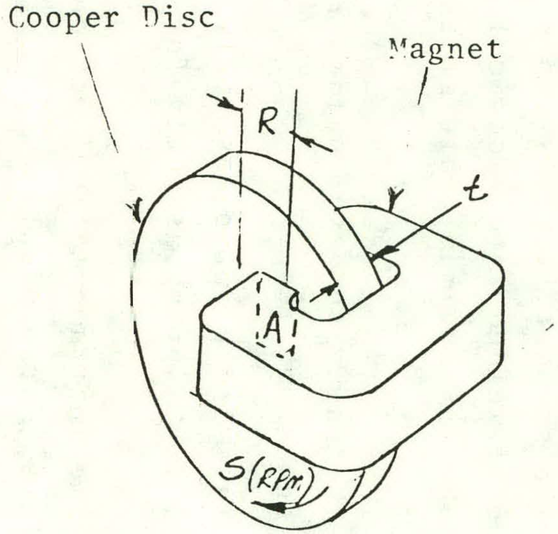
t = Thickness of copper disc.

R = Radius from center of rotation.

S = Slip speed of center of "contact" area in Rev. Per Min. (RPM).



EDDY-CURRENT TORQUE



Dynamic analysis of the control system reveals that the selected damping parameters (1 lb in/rpm), will provide sufficient damping in the operational range.

The damper bearings (which are sealed) serve also as generator frame support (Figure 4-12).

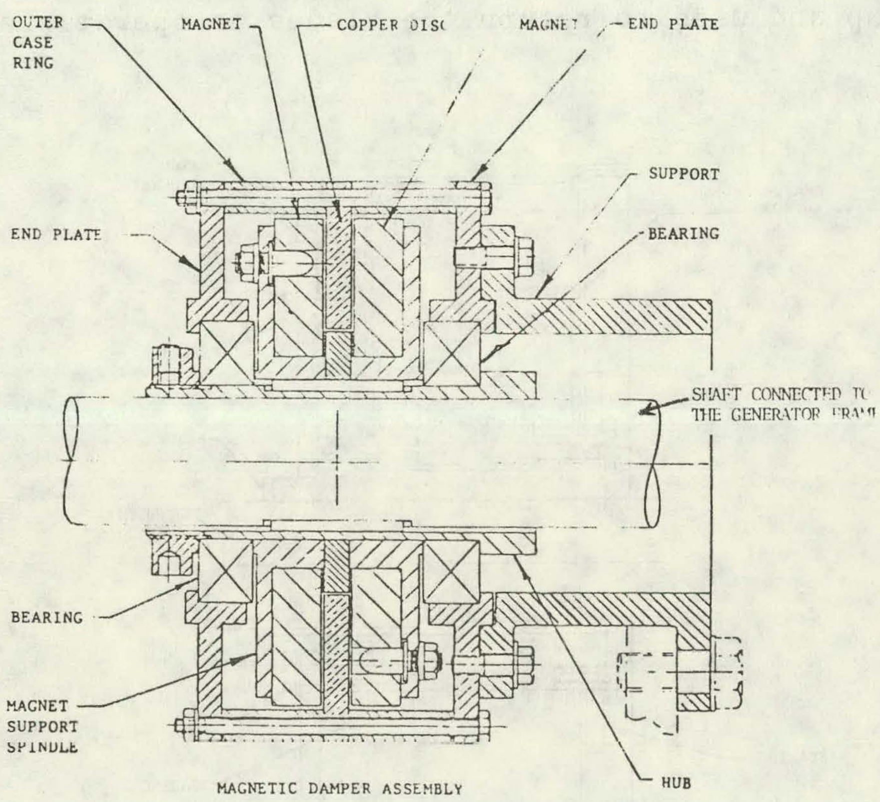


Figure 4-12 MAGNETIC DAMPER ASSEMBLY

4.4.5 Generator Balance Spring Assembly (Figure 4-13)

The balancing of the torque is performed by a coiled power spring. The spring is wound into a case and linked to the generator frame shaft. The spring can be preset to the desired torque in a laboratory and retained in position by the service lock pin (which is also a safety device to prevent unwinding when the system is serviced). The spring develops an increasing resistive torque during the control phase (about 1.5 turns) and an asymptotic characteristic during the feathering cycle when it is wound 6 turns. The stored energy is released in start up and used to return the blades to operational angles.

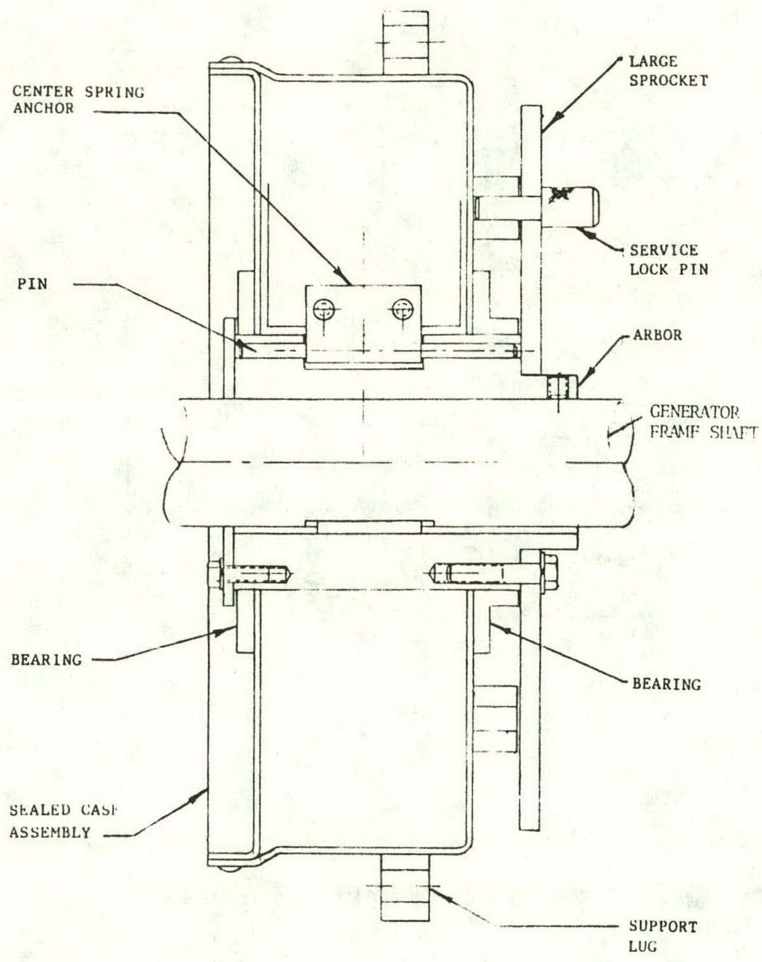


Figure 4-13 GENERATOR BALANCE SPRING ASSEMBLY

4.4.6 Nacelle and Spinner Assembly (Figure 4-14)

The nacelle provides an aerodynamically clean surface over the mechanism and protects the interior from atmospheric effects.

The nacelle was designed in a slim configuration to reduce the drag from side winds. It is removable for servicing. Bulkheads attached to the primary structure support the nacelle. Air openings are provided for cooling and screened to prevent dust and animal entry.

Main features of the nacelle are:

Filament wound structure, light weight and impervious to outdoors environment.

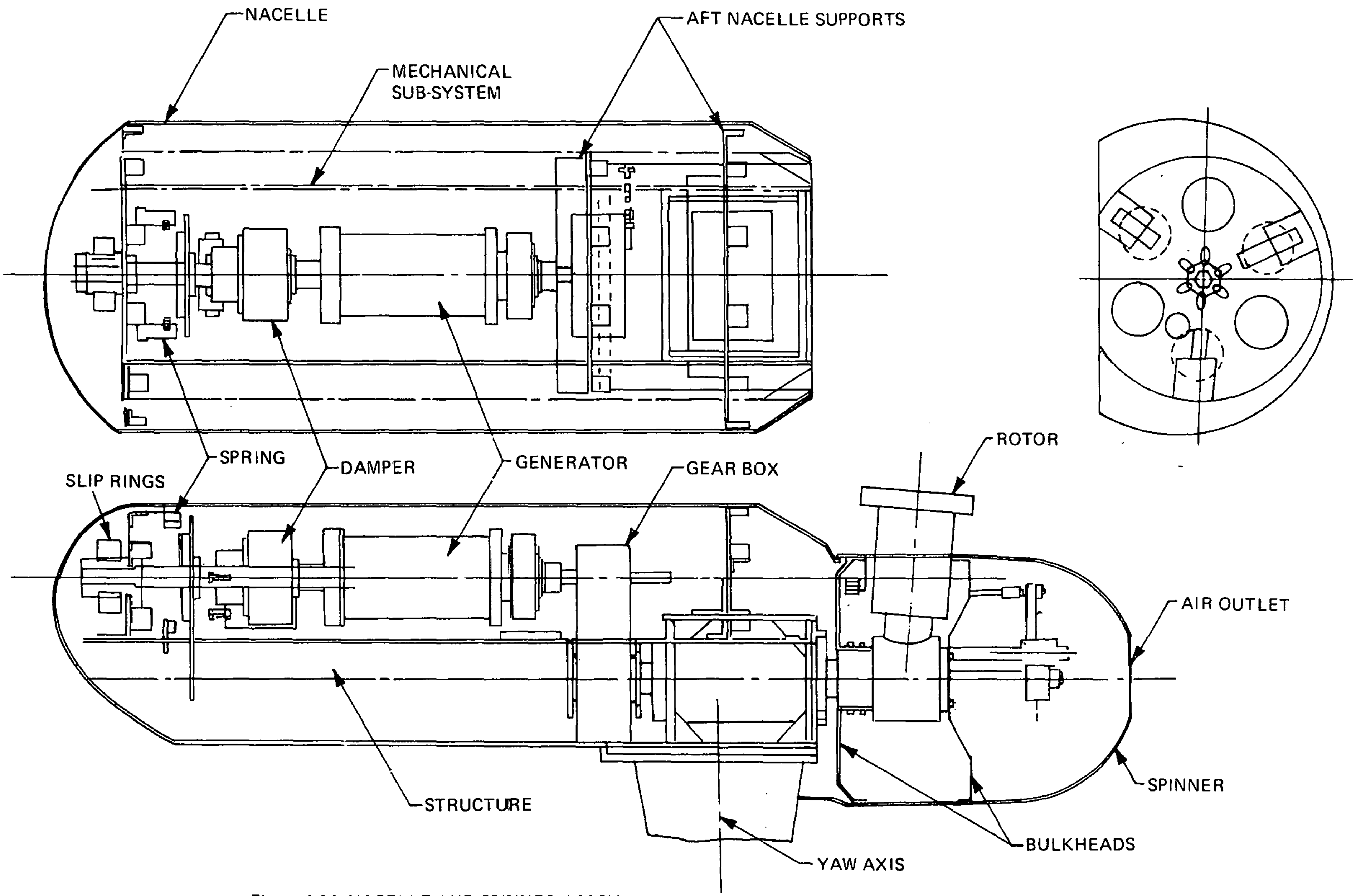


Figure 4-14 NACELLE AND SPINNER ASSEMBLY

5.0 ELECTRICAL SUBSYSTEM

5.1 HISTORICAL SUMMARY

The electrical subsystem consists of the electric generator and its power circuits, electrical controls and the utility interface. The electrical subsystem remained essentially unchanged from the proposal design, with the following major exceptions:

5.1.1 Two-Speed Generator

A two-speed induction generator was originally proposed to provide a very low cut-in wind speed, thus maximizing energy recovery in a 10 mph mean annual wind speed. This design was compared to a single-speed induction generator, at PDR-2, as summarized in the trade study of Table 5-1.

Table 5-1
GENERATOR ALTERNATIVES TRADE STUDY

ITEM	TWO SPEED	SINGLE SPEED
HARDWARE COST (1) (2)	$3,702	$3,441
ANNUAL kWh (2)	16,366	15,385
COE (2)	$ .0611/kWh	$ .0618/kWh
RISK FACTOR	0.95	1.0
EVALUATED COE (2)	$ .0643/kWh	$ .0618/kWh
SELECTION:	SINGLE SPEED GENERATOR	
REASON:	• LOWEST EVALUATED COST OF ENERGY • SIMPLICITY • RELIABILITY • WEAR AND FATIGUE	

(1) COST FOR 10,000th UNIT, 1978 $
(2) FOR TRADE STUDY COMPARISON ONLY

As a result of this study, a single speed induction generator was selected for its lower technical risk, greater simplicity and reliability. Other impacts of this change include the following:

- Reduced complexity for the electrical control system.
- Increased cut-in wind speed from 6 mph in the proposal to 8.8 mph in the final design.

5.1.2 Startup Alternatives

The SCI proposal design utilized the induction generator as a motor to start the SWECS rotor. This was compared to aerodynamic start at PDR-2, as shown in the trade study, Table 5-2. As a result, aerodynamic startup was selected as being the most cost effective while avoiding voltage flicker on the electric power system and resulting in lower wear and fatigue for the generator, gear box and rotor.

Table 5-2
START-UP ALTERNATIVES TRADE STUDY

ITEM	TWO SPEED	SINGLE SPEED
HARDWARE COST (1) (2)	$3,860	$3,860
ANNUAL kWh (2)	15,385	15,156
COE (2)	$ .065/kWh	$ .066/kWh
RISK FACTOR	1.0	1.0
EVALUATED COE (2)	$ .065/kWh	$ .066/kWh
SELECTION:	AERODYNAMIC START	
REASON:	• LOWEST EVALUATED COST OF ENERGY • AVOIDS VOLTAGE FLICKER PROBLEMS • ALLOWS SMALLER DIESEL GENERATOR • WEAR AND FATIGUE FOR GENERATOR, GEAR BOX AND ROTOR	

(1) COST FOR 10,000th UNIT, 1978 $
(2) FOR TRADE STUDY COMPARISON ONLY

5.1.3 Electrical Versus Mechanical Sensors

In the proposed design, various mechanical limit switches were utilized as torque sensors. These were replaced by electrical sensors, such as a watt transducer for sensing motoring operation. These changes were made to provide better access to control elements (at ground level instead of nacelle) for adjustment and for repair. Reliability of the system was improved accordingly.

5.1.4 Utility Interface

To provide protection for SWECS electrical equipment and that of the interconnected utility grid, the following protective relays were added:

(a) Over/under frequency relay
(b) Over voltage relay

Both these devices serve to disconnect the SWECS from the interconnected utility grid and from its own local power system in the event of self excitation of the induction generator. Self excitation has been known to occur if the utility power source is interrupted while significant capacitance remains connected in parallel with the induction generator.

5.2 DESIGN FEATURES

Electrical power and control systems have been designed to provide fully-automatic and fail-safe operation of the wind turbine generator. Primary features of the design include:

- 6.5 kW induction generator
- Direct utility interface - no synchronizing or power-conditioning equipment required
- Inherent voltage, frequency and rotor speed control established by the utility grid
- Electro-mechanical control relays which are immune to most environmental disturbances
- Most electrical components located at ground level

5.3 TYPE CONSIDERATIONS

5.3.1 Comparative Tests

The three-phase induction machine has long been used as an efficient generator. The 4 kW SWECS, however, was required to generate at 240 volts, single phase, and the efficiency of a single-phase machine of this type was questionable. SCI therefore commissioned a series of tests to be performed by the California State University at Pomona. These tests were to find the efficiency, under various load conditions, of several types of induction motors, driven above their synchronous speeds and generating directly into the utility 240 volt single-phase power grid.

5.3.2 Test Results

These tests are covered in Chapter 11, Tests and Instrumentation, and in Reference 20. The tests showed that surprisingly high efficiencies were attainable using readily available capacitor-run single-phase induction motors driven above their synchronous speeds. Peak efficiencies as high as 89% were demonstrated, using a standard 5 HP motor.

5.3.3 Generator Selection

The SCI 4 kW SWECS was designed to produce 5.7 kW at rated wind speed and 30% higher, or 7.41 kW, at cut-out. It was desired to select the smallest generator which could cover these requirements, over a wide range of ambient temperatures (See Table 1-1), and still provide a 25-year life. An oversize generator could have been chosen but this would add to the cost and weight and would reduce energy production because of the larger fixed losses associated with the larger unit. It was therefore decided to select the generator size and design to give adequate insulation life while allowing moderate overheating for the short time periods in which maximum ambient temperature and maximum wind speeds occur simultaneously.

5.3.4 Generator Life Calculations

A generator rated 6.5 kW was selected. Class H (high temperature) insulation was specified, though the 6.5 kW rating was based on Class F insulation. Calculations showed that the insulation would last more than 80 years in a 10 mph mean annual wind, and more than 20 years in an 18 mph mean annual wind. These calculations were based on the following conservative assumptions:

- Ambient is 60°C for 25% of the time
- Ambient is 40°C for 25% of the time
- Ambient is 30°C for 50% of the time
- Class F temperature rating is 115°C rise above 40°C ambient
- Class H insulation is rated 140°C rise above 40°C ambient
- Normal insulation life is 7 years based on continuous operation at rated total temperature
- Temperature rise above ambient is proportional to (generated kW)^2
- Insulation life is cut in half for continuous operation 10°C above rated total temperature
- Insulation life is doubled for continuous operation 10°C below rated total temperature

5.4 ELECTRICAL CONTROLS

5.4.1 Scope

Electrical controls consist primarily of a number of 110 volt electro-mechanical relays, mounted in a relay panel at ground level. A schematic diagram of the control system is shown in Figure 5-1. Components located in the nacelle are shown surrounded by boxes. Electrical connections between the nacelle and ground are made through sliprings located just below the nacelle in the yaw structure.

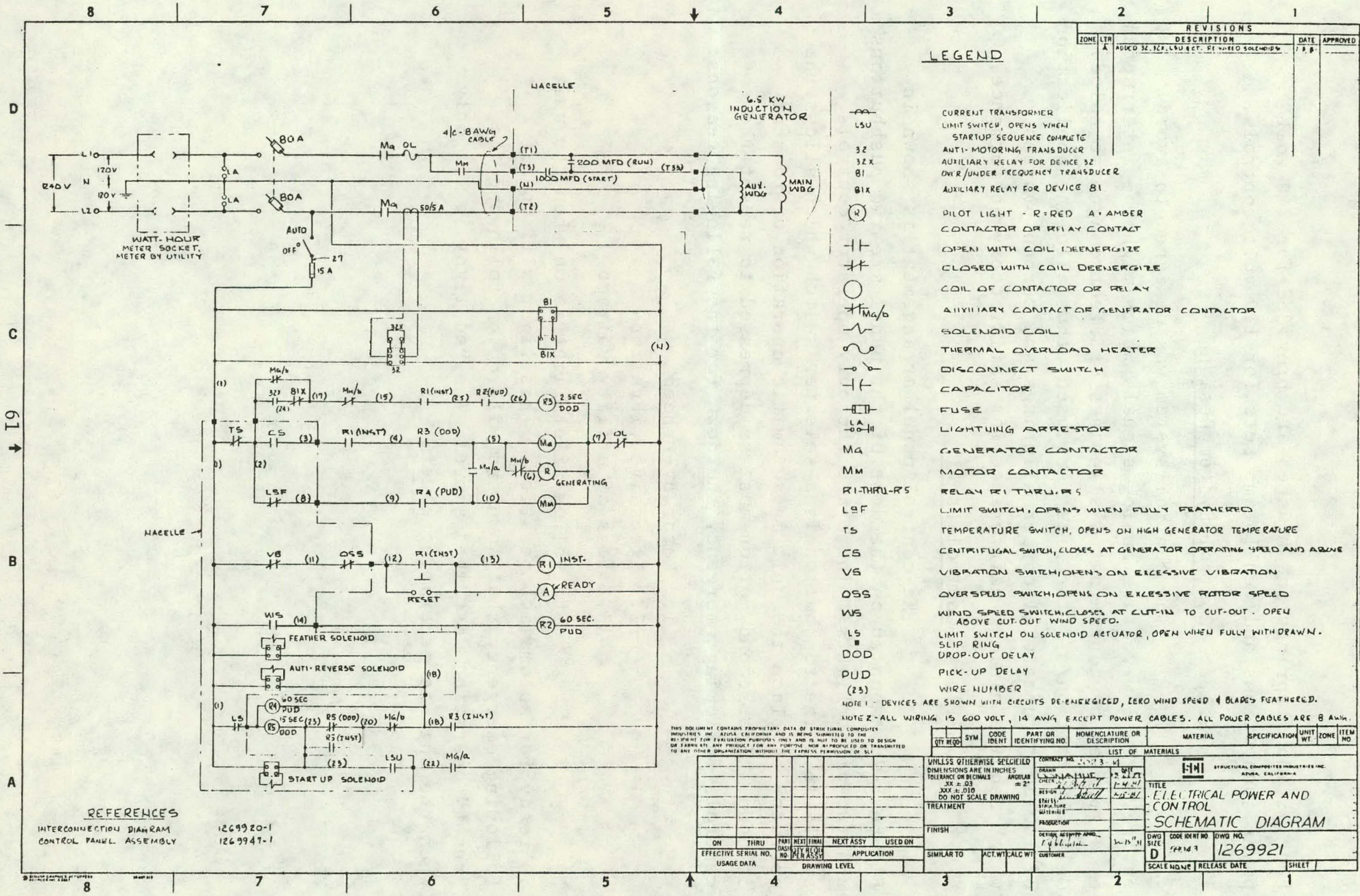


Figure 5-1 ELECTRICAL CONTROLS SCHEMATIC DIAGRAM

Four 35 amp sliprings and brushes are provided for 240 volt power connections and ten 10 amp sets for 110 volt controls.

5.4.2 Control System Design

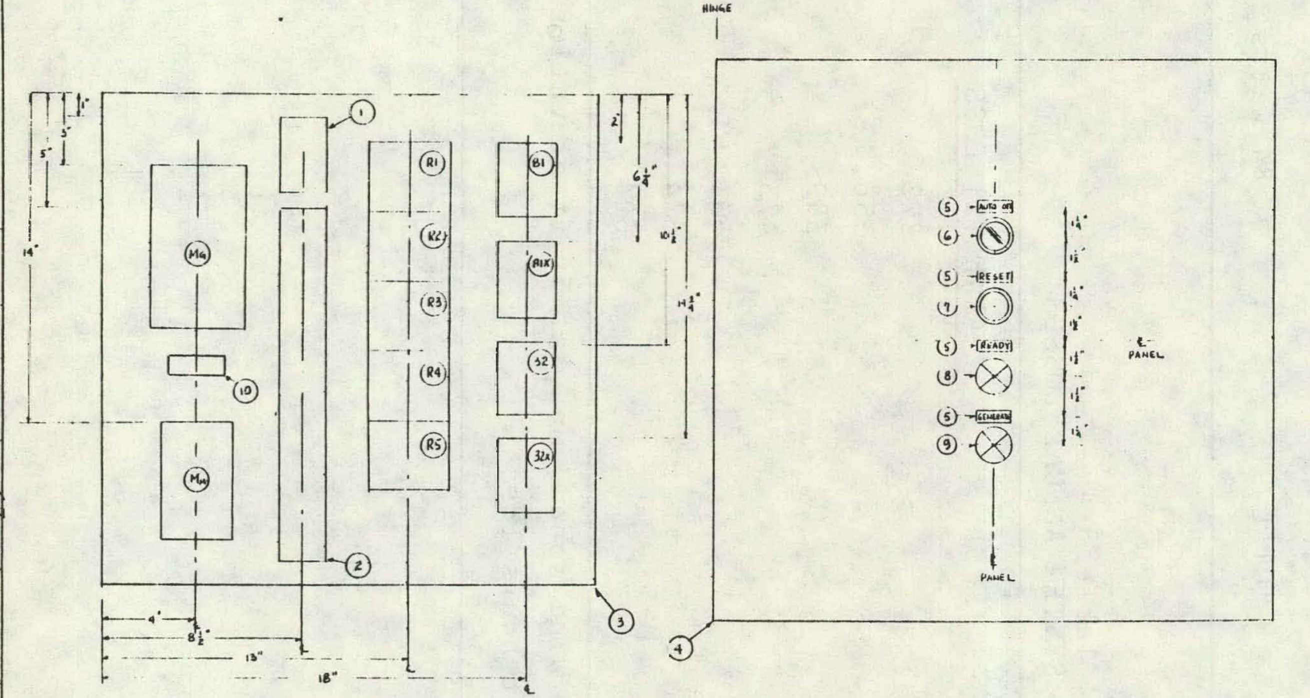
The control system was designed to avoid entirely any low-voltage analog signals which might be interrupted or distorted by contaminated sliprings or electromagnetic noise. Only 110 volt on/off circuits were used, thus allowing application of low-cost commercial copper sliprings and carbon brushes which require minimum maintenance for satisfactory performance.

5.4.3 Control Panel

The control panel arrangement is shown in Figure 5-2. Mounted on the door of the panel are two pushbuttons and two pilot lights. These are the only operator controls available. The top button is a selecter switch which allows the operator to place the unit in automatic operation or to shut it down. The second button must be depressed to re-set the controls following emergency shutdown for the following reasons:

1. Rotor overspeed
2. Excessive vibration
3. Loss of utility voltage

The yellow pilot light is on when the unit is ready for automatic operation. This light will go off following emergency shut down and will go on following the "reset" operation described above. The red light is lit only when the unit is generating power.



QTY REQD	SYM	CODE IDENT	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	MATERIAL	SPECIFICATION	UNIT WT.	ZONE	ITEM NO.
1	32			WATT TRANSDUCER, SQ. D. CAT. NO. TTW 25000					
1	32X			TRANSDUCER RELAY, SQ. D. CAT. NO. PR-101-TR					
1	MG			MOTOR STARTER, 2-POLE, SIZE 2, CUTLER-HAMMER CAT. NO. B10DNUF					
1	MM			CONTACTOR, 2 POLE, 30 AMP, CUTLER-HAMMER CAT. NO. C30CN2					
1	R1			300 VOLT CONTROL RELAY, SQ. D CAT. NO. GO-20					
1	R2			GO-00-GE					
1	R3			GO-40-GD					
1	R4			GO-00-GE					
1	R5			GO-00-GD					
1	B1			FREQUENCY TRANSDUCER, 58 TO 62 HZ, SQ. D. CAT. NO. TTF25240-62					
1	B1X			TRANSDUCER RELAY, SQ. D. CAT. NO. PR-101-TR					
1	1			FUSE BLOCK, 15 AMP FUSE					
50	2			TERMINAL BLOCK, SQ. D. TYPE GM-3					
1	3			REMOVABLE MOUNTING PANEL, PART OF ITEM 4					
1	4			HINGED ENCLOSURE, NEMA 3R, HOFFMAN CAT. NO. A24R24BHC					
4	5			NAMEPLATE, LAMINATED PLASTIC, ½ x 1½ INCH, MARKING AS SHOWN					
1	6			SELECTOR SWITCH, CUTLER-HAMMER CAT. NO. 10250T1311					
1	7			PUSHBUTTON, CUTLER-HAMMER CAT. NO. 10250T104R					
1	8			INDICATING LIGHT, AMBER, CUTLER-HAMMER CAT. NO. 10250T36					
1	9			INDICATING LIGHT, RED, CUTLER-HAMMER CAT. NO. 10250T34					
1	10			CURRENT TRANSFORMER, 50 TO 5 AMP					

LIST OF MATERIALS

REFERENCES

SCHEMATIC DIAGRAM 1269921-1
INTERCONNECTION DIAGRAM 1269920-1

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES
TOLERANCE ON DECIMALS .XX ± .03 .XXX ± .010 ANGULAR ± 2°
DO NOT SCALE DRAWING
TREATMENT
FINISH
SIMILAR TO | ACT. WT | CALC WT

CONTRACT NO. 2023-14
DRAWN GB 1-5-41
CHECK 1-4-41
DESIGN 1-15-41
STRESS/STRUCTURE MATERIALS
PRODUCTION
DESIGN ACTIVITY APPD 1-1-41
CUSTOMER

STRUCTURAL COMPOSITES INDUSTRIES INC. AZUSA, CALIFORNIA

TITLE ELECTRICAL POWER AND CONTROL CONTROL PANEL ASSEMBLY

DWG SIZE D | CODE IDENT NO. 58943 | DWG NO. 1269947
SCALE NONE | RELEASE DATE | SHEET 1

ON | THRU | EFFECTIVE SERIAL NO. | USAGE DATA | PART DASH NO. | NEXT ASSY | FINAL ASSY | DRAWING LEVEL | NEXT ASSY | USED ON | APPLICATION

COPY

Figure 5-2 ELECTRICAL CONTROL PANEL

5.4.4 Control System Losses

The electrical control system was carefully designed to minimize parasitic losses associated with such systems. Shown below are the calculated power requirements and losses which were deducted from the annual kilowatt hours to give those summarized in Table 8-5.

Table 5-3
CONTROL SYSTEM POWER REQUIREMENTS

Solenoids	2 x 20 watts	= 40 watts
Relays	5 x 6 watts	= 30 watts
Size 2 Starter	1 x 14 watts	= 14 watts
	TOTAL	84 watts

Table 5-4
CONTROL SYSTEM ANNUAL LOSSES

AAWS (1)	AkWh (2) LOSS
8 mph	272
9 mph	336
10 mph	389
12 mph	473
14 mph	531
16 mph	574
18 mph	604

(1) Annual average wind speed, mph, at 30 ft elevation

(2) Annual kilowatt hours loss

5.5 UTILITY INTERFACE

U.S. utility interface requirements for induction generators consist of a lockable disconnect switch and provisions for a separate meter. These are shown on the schematic diagram, Figure 5-1. Additional protective relays have been provided to guard against self excitation of the induction generator, as covered in Paragraph 5.1.4. For a more detailed discussion of utility and standby generator interface requirements, see Reference 21.

5.6 RECOMMENDED CHANGES

The changes listed below were discussed at the Final Design Review. SCI has agreed that these changes should be incorporated, but they are not reflected in the drawings and descriptions contained in this report.

1. Provide automatic startup following loss of voltage on the utility grid.
2. Provide an accurate rotor speed sensor in place of centrifugal switches.
3. Consider eliminating the operation of the generator as a motor to ensure that full feathering is attained.
4. Use square D frequency relay, catalog No. 810UF in place of the frequency transducer and separate relay shown on the drawings.

6.0 TOWER

6.1 HISTORICAL SUMMARY

6.1.1 Proposal

The tower proposed was a free standing monolithic tapered composite tube made by SCI's patented TFT process. Trade off studies between guyed and free-standing tower were performed. Tower erection was proposed by using a ginpole and a winch.

6.1.2 PDR I and II

Tower trades between free-standing composite or steel tapered poles and guyed straight composite or steel poles was presented. The recommendation was for a tapered composite free-standing pole.

6.1.3 CDR and FDR

A design was presented with supporting analysis in CDR and some minor changes were made at FDR. The basic erection method remained the same except that use of a towtruck was suggested instead of a winch.

6.2 FINAL DESIGN (FDR)

6.2.1 Description Of The Tower (Figure 6-1)

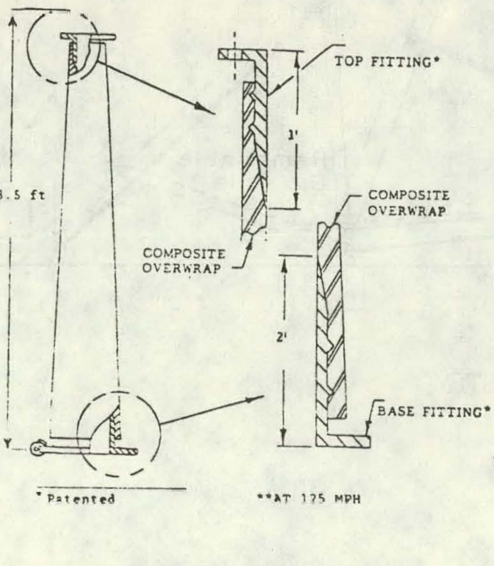
The tower consists of a free-standing monolithic tapered composite wound tube. The tower base and top fittings of steel are wound in place during tower fabrication and become a permanent part of the tower.

The winding is made by the SCI special TFT process (transverse filament tape - patented) which allows high speed filament winding of tapered wall thickness on large tapered mandrels. TFT can be wound with up to 90% of its E-glass reinforcing filaments aligned with the long axis of the tube, thus yielding higher axial stiffness than conventional winding process.

An isophtalic polyester resin is used for the matrix and a polyurethane paint for the environmentally resistant outer coating.

The composite tower can be light in weight compared to an un-guyed steel pole tower, since the composite construction allows tailoring of the wall thicknesses, diameters and directional properties to match the applied moments and loads, and the axial strength to weight ratio is much higher than that of the mild steel used in conventional towers.

One special attention area for the composite tower, however, is the stiffness and natural frequency. Since TFT composites have roughly the same axial modulus to density ratio as steel, they tend to produce structures of approximately the same weight as steel, when outer contour, stiffness, and natural frequency are held constant. With composite towers, however, the effective stiffness, EI, may be increased by using larger diameters (with due regard to tower wake and elastic instability) than used on a comparable steel tower. In this manner, a composite tower of equal stiffness and lighter weight than a comparable steel tower may be designed. The composite tower material is essentially non-corroding and is non-conductive. It must be coated with polyurethane or equivalent paint for ultra-violet protection and a suitable ground is needed for lightning protection.



DESCRIPTION	
HEIGHT (ft)	48.5
BASE DIAMETER (in)	24
TOP DIAMETER (in)	12
BASE WALL THICKNESS	0.4
TOP WALL THICKNESS	0.15
TOWER COMPOSITE WEIGHT (lbs)	804
BASE FITTING WEIGHT (lbs)	400
TIP FITTING WEIGHT (lbs)	66
TOTAL WEIGHT (lbs)	1270
TIP DEFLECTION (in)**	19

Figure 6-1 TOWEP DESCRIPTION

*Patented **AT 125 MPH

6.2.2 Tower Erection

The tower erection concept is shown in Figure 6-2. A conventional concrete foundation is provided, with seven embedded anchor bolts, for the tower.

Intermediate Plate - Two hinges are fastened to the plate and the tower is secured with eight bolts to the intermediate plate (Figure 6-3). A similar smaller foundation is provided for the erection pulley. The ginpole, cables, and tower baseplate hinge are provided with the SWECS so that only a towtruck is needed for the tower erection. The tower and nacelle are assembled and the pulley, ginpole and cables are rigged. The towtruck is then used to raise the tower to vertical. Once the tower is vertical and the bolts are in place, the ginpole and pulley may be removed and stored until needed to lower the tower for maintenance. The erection stay cable is stowed by tieing it to the cleat provided.

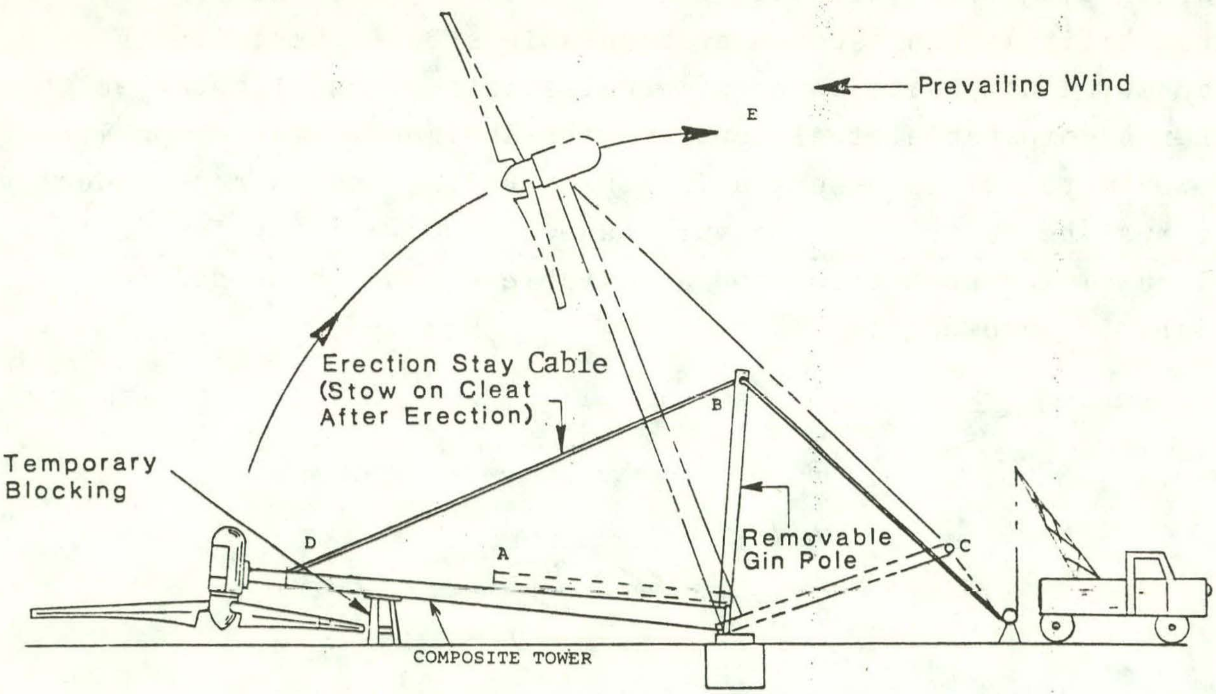


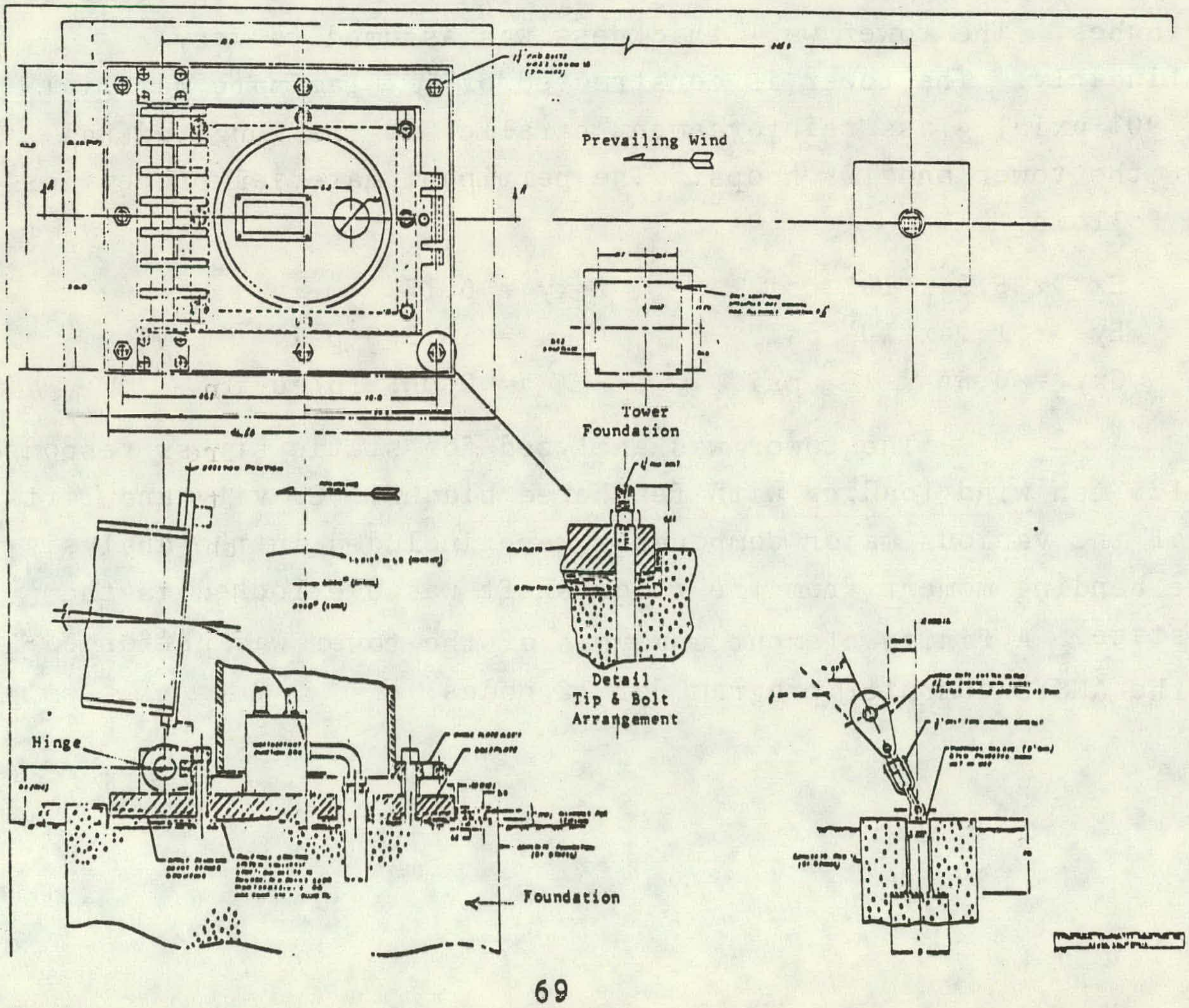
Figure 6-2 TOWER ERECTION SYSTEM

6.2.3 Tower Foundation (Figure 6-3)

The foundations will be site specific. The designs will be determined by the subsurface soil properties, location of the water table, and the depth of frost penetration. The characteristics of the site must be evaluated by a competent soil geologist aided by core sample of the structures beneath the proposed location.

In the unusual case where the wind machine must be constructed over mud, silt, peat or artificial fill, special foundation systems that include an adjustable guy system may be needed to properly compensate for settlement. If excavation into the water table is necessary, bottoming of the foundation on bedrock or refusal is particularly important. In regions subject to frost penetration, the pole foundation must project a minimum of three feet beneath the frost line. The hoist foundation must project a least a foot beneath the expected frost level.

Figure 6-3 TOWER FOUNDATION AND HINGE DETAIL



6.3 TRADE OFF AND SUPPORTING ANALYSES

6.3.1 Trade Off Summary

Height of a tower is a classical case of trade off between first cost and long term cost of energy. Average wind speed (energy) and tower initial cost both go up as tower height increases. Another aspect is the shipment. A one piece tower longer than 40 feet may require special permits for truck shipment in many localities so a multi-piece tower was also considered. Trade offs between guyed and un-guyed composite, steel and steel truss towers were performed (2). Aspects of cost, weight, transport and aesthetics were compared and led to the present design.

6.3.2 Tower Analysis (16)

The model used was a free-standing, 50 ft high TFT tower supported on a fixed concrete foundation. The tower tapers linearly from a base diameter of 24 inches to a tip diameter of 12 inches. The tower wall thickness was assumed to vary quasi-linearly. The tower is constructed of TFT laminate material having 90% axial glass reinforcement parallel to the longitudinal axis of the tower and 10% hoops. The pertinent material properties are as follows:

$E_x = 5.3 \times 10^6$ psi	$\nu_{xy} = 0.17$
$E_y = 1.9 \times 10^6$ psi	$G_u = 33{,}500$ psi
$G_{xy} = 0.56 \times 10^6$ psi	$\rho = 0.065$ lb/cu in.

The tower was analyzed for static stress response under 125 mph wind loading with feathered blades. Gravity and lift loads of the various major components were included in the analysis but the bending moment from the rotor shaft was overlooked in the first stage. A finite element analysis of the tower was performed using the ANSYS computer program for 12 nodes.

The margins of safety against local buckling of the tower were computed using the procedures of Ref. (17). A knock down factor of 0.4 was included in the margin of safety calculations. The results of the analysis of the free-standing tower are summarized in Table 6-1. The safety margins on strength are over 3 for the loads assumed and all the buckling safety margins are positive. However, it is recommended to increase the tower wall thickness in order to improve the tower endurance. It can be done on the same mandrel just by adding several TFT layers.

Table 6-1 TOWER CHARACTERISTICS AND SAFETY MARGINS FOR 125 MPH WIND

TOWER HEIGHT (ft)	EI (lbs-in.^{2})x 10^9	GJ (lbs-in.^{2})x 10^9	STATIC PRESSURE (psi)	SAFETY MARGIN ON STRENGTH	BUCKLING ALLOWABLE (psi)x 10^3	SAFETY MARGIN ON BUCKLING
0			7,890	3.3	10.9	0.4
5	9.8	2.1	7,570	3.4	11.5	0.5
10	8.4	1.8	7,230	3.6	12.1	0.7
15	7.1	1.5	6,880	3.9	12.8	0.9
20	6.0	1.3	6,480	4.2	13.6	1.1
25	4.3	0.9	6,930	3.8	12.6	0.8
30	3.0	0.6	7,350	3.6	11.5	0.6
35	2.0	0.4	7,680	3.4	10.2	0.3
40	1.3	0.3	7,620	3.4	8.8	0.2
45	0.7	0.2	6,350	4.3	6.9	0.10
50*			2,350	13.3	4.8	1.1

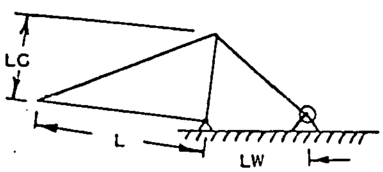
* Tower shortened to 48.5 ft - analysis is still valid.

6.3.3 Base And Top Fittings

The steel fittings for the base and top joints of the tower are shown in Figure 6-1. Each joint consists of a steel fitting, the TFT composite of the tower and a TFT hoop overwrap layer. The joint was considered unbonded along the interface of the steel fittings and the TFT. This was done to conservatively approximate a condition of possible adhesive failure due to long term environmental exposure. Detailed finite axisymetric element mathematical models of each joint were developed (144 elements and 194 nodes, for the base fitting and 115 elements and 163 nodes for the top fitting) and analyzed to demonstrate structural integrity of the joint design configurations.

6.3.4 Erection System Figure 6-4 (See Also Figure 6-2)

Figure 6-4 ERECTION SCHEMATIC



- GEOMETRY
  - TOWER LENGTH, L = 48.5 ft
  - GIN POLE LENGTH, LG = 18.4 ft
  - DISTANCE FROM TOWER TO WINCH, LW = 22 ft
  - INITIAL ERECTION ANGLE, $\theta = 5^{o}$
- WIND LOADS
  - MAXIMUM CROSS WINDS DURING ERECTION 15 mph
- GRAVITY LOADS
  - TOWER WEIGHT = 1270 lbs
  - NACELLE/ROTOR ASSEMBLY WEIGHT = 1500 lbs
- MAXIMUM INDUCED LOADS
  - GIN POLE, 6770 lbs
  - WINCH WIRE, 7165 lbs
  - ERECTION STAY, 5950 lbs
  - AXIAL LOAD TOWER BASE, 5730 lbs

Based on the information and assumptions listed above, all parts were designed for a safety factor of five.

7.0 STRUCTURAL AND DYNAMIC ANALYSIS

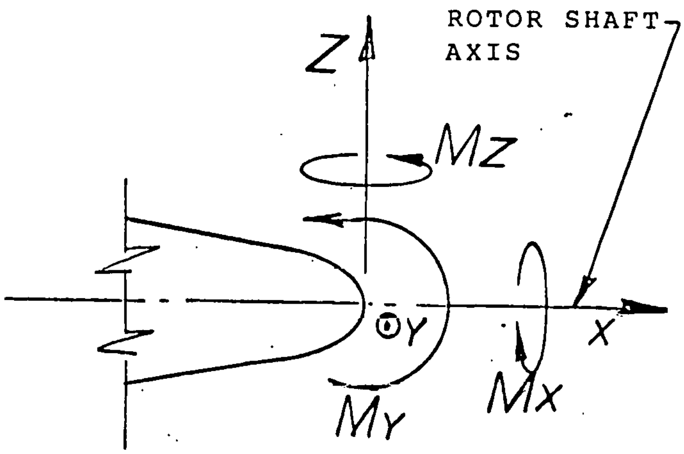
7.1 STRUCTURAL ASPECTS HIGHLIGHTS (4, 5)

7.1.1 Rotor Shaft (See Figure 7-1 For Axis Definition)

Table 7-1 LOADS ON ROTOR SHAFT FOR DIFFERENT CASES

LOAD \ CASE	RATED	CUT-OUT (10% overspeed + ice)	SURVIVAL
Mx (ft-lbs)	600	600	0
My (ft-lbs)	940	± 3500 (5100)	620
Mz (ft-lbs)	± 63	± 198	352
Fx (lbs)	1573	144	Negligible
Fy (lbs)	Negligible	562	400
Fz (lbs)	Negligible	-1137	Negligible

Figure 7-1 AXIS DIRECTIONS FOR ROTOR SHAFT



The design loads were computed from Table 7-1 and are presented in Table 7-2.

Table 7-2 DESIGN LOADS FOR ROTOR SHAFT

LOAD	MAXIMUM STATIC LOADS		FATIGUE LOADS	
	VALUE	SOURCE	VALUE	SOURCE
Mx	600 ft-lbs	RATED	0 TO 600 ft-lbs	RATED
My	5100 ft-lbs	CUT-OUT + OVERSPEED + ICE ON TIP	± 3500 ft-lbs	CUT-OUT
M_z	200 ft-lbs	CUT-OUT	± 200 ft-lbs	CUT-OUT
Fx	1600 lbs	RATED	0 TO 1600 lbs	RATED
Fy	800 lbs	CUT-OUT	0 TO 560 lbs	CUT-OUT
F_z	1150 lbs	CUT-OUT	0 to 1200 lbs	CUT-OUT

Rotor shaft material is a 4142H annealed (Y.P 70,000 lbs/in.^{2}) tube ϕ 3.75 x 1.75 in.

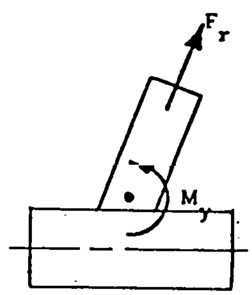
After machining, the stresses will not exceed 18,000 lbs/in.^{2}, giving a safety factor of more than 3.8. Fatigue values are below the endurance limit.

7.1.2 Blade Support At Hub

The loads on the blades during operation are presented in Table 3-2. The design loads are presented in Table 7-3.

LOAD	VALUE	SOURCE
Fr (lbs)	2400	Radial loads + 10% overspeed + 10 lbs ice on tip
My (ft-lbs)	-3600	Flatwise bending moment at survival
My (ft-lbs)	-400 to -1200	Fatigue condition (cut-out)

Table 7-3 DESIGN LOADS FOR BLADE SUPPORT AT HUB



The critical part to be checked is the welding to the hub. The maximum stress expected there is 4800 $lbs/in.^2$ which is a low value (safety factor more than 3).

7.1.3 Bearings

Structural bearings (blades, rotor, yaw) were designed to a $\frac{C}{P}$ value of more than 6 which means practically infinite life.

The pitch control bearings were designed to withstand more than 10^6 feathering cycles under full load.

7.1.4 Ball Screws And Nuts

The ball screws were designed to withstand more than 10^6 feathering cycles under full load.

7.1.5 Solenoid Selection

Solenoids have been selected so that the minimum safety factor for pulling force (when hot) will be more than 4, taking the friction into consideration.

7.1.6 Summary

All the main control devices and structural elements have been carefully selected and found adequate with acceptable performance and life expectancy.

7.2 AEROELASTIC ANALYSIS SUMMARY (11)

The wind turbine blade has been analyzed for stability, considering the following type of instability.

(a) Torsional divergence (rotating and feathered)

(b) Pitch-flap flutter

Stability was examined for the design position of the elastic axis (30% of chord at blade root), as well as excursion of the elastic axis of $\pm$ 20% of chord. In all cases examined, the blade was found to be stable in the expected operating range. The stiffness (k_b) between the blade pitch bearing and the control system was assumed (conservatively) and analyzed for values between 10^3 and 10^4 (ft-lbs/radian). The blade was found to be stable for all values of k_b that were examined above 10^3 (ft-lbs/radian).

7.3 MODAL ANALYSIS (11)

7.3.1 The Model

A finite element model of the structure, composed of 84 beam elements, with 462 degrees of freedom was formed (Figure 7-2).

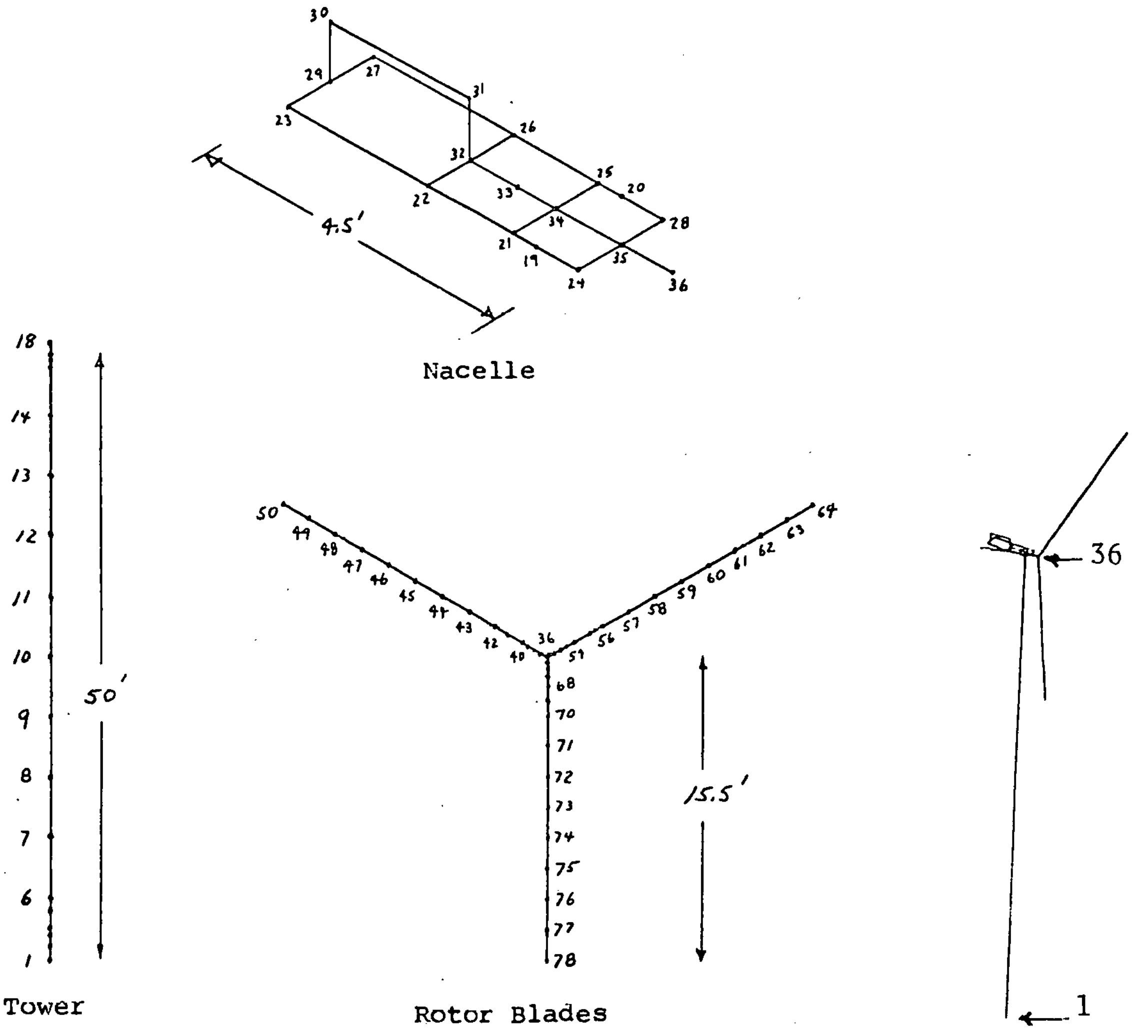


Figure 7-2 FINITE ELEMENT MODEL WITH NODES NUMBERED

Lumped masses were incorporated at several points to account for the masses of the mechanical elements.

In order to represent the yaw freedom of the nacelle and the spin freedom of the rotor, the yaw bearing (node 18) and the rotor shaft (node 36) were given extremely low torsional stiffnesses.

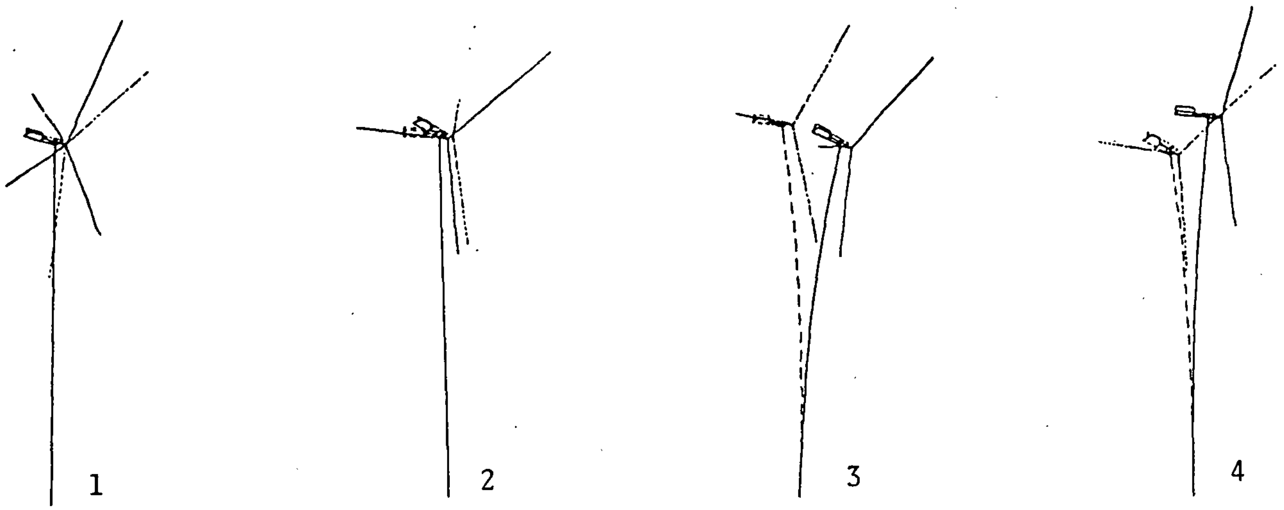
Centrifugal stiffening effects were not included in any of the finite element analyses. Tip masses on the blades were not included either. It was found that the increase in natural frequency due to centrifugal stiffening tended to offset the decrease in frequency due to the added mass at the tip.

7.3.2 Natural Frequency Analysis

The twelve natural frequencies and associated mode shapes were calculated for two pitch angles of the blades (0^o and 90^o). The first four frequencies and the description of the mode for 0^o and 90^o blade pitch (identical) are described in Figure 7-3.

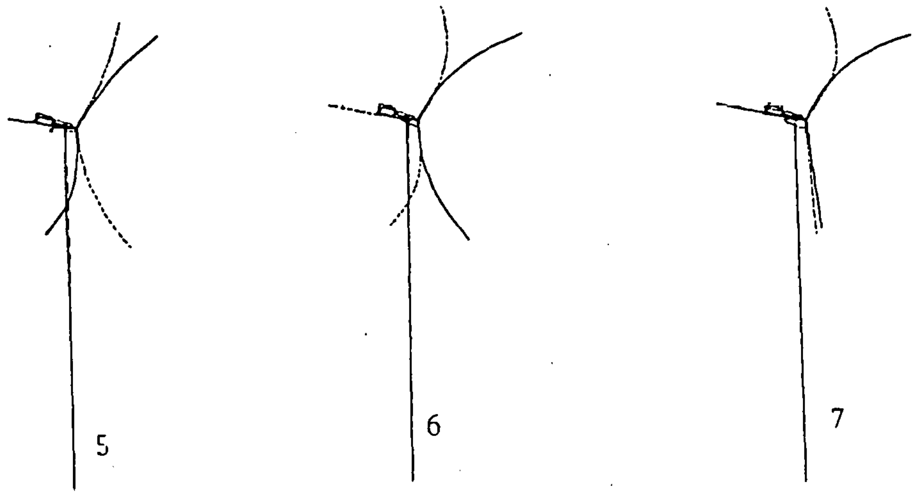
The description of the modes 5, 6 and 7 for 0^o pitch is given in Figure 7-4 and for 90^o pitch is given in Figure 7-5.

Figure 7-3 FOUR FIRST MODES AND FREQUENCIES (0^o and 90^o BLADE PITCH)



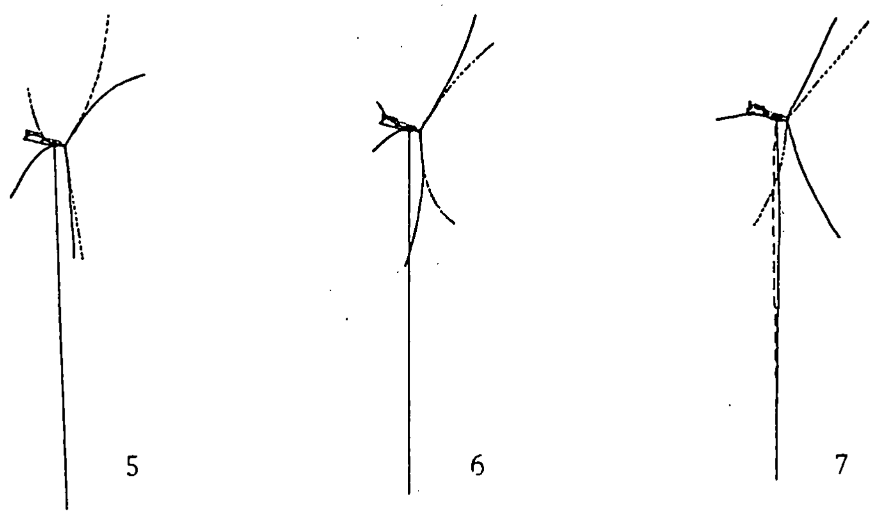
MODE NUMBER	MODE	FREQUENCY (Hz)
1	RIGID BODY MOTOR SPIN	0.14
2	RIGID BODY NACELLE YAW	0.18
3	TOWER FORE-AFT BENDING	0.766
4	TOWER LATERAL BENDING	0.782

Figure 7-4 5TH, 6TH AND 7TH MODES AND FREQUENCIES (0^o BLADE PITCH)



MODE NUMBER	MODE	FREQUENCY (Hz)
5	TOWER FORE-AFT WITH ROTOR PITCH	3.75
6	TOWER FORE-AFT WITH BLADE BENDING IN PHASE	4.19
7	NACELLE AND BLADE YAW	4.89

Figure 7-5 5TH, 6TH AND 7TH MODES AND FREQUENCIES (90° BLADE PITCH)



MODE NUMBER	MODE	FREQUENCY (Hz)
5	BLADE FLAP	4 09
6	HUB YAW AND BLADE FLAP	4.19
7	NACELLE PITCH AND BLADE FLAP	5.48

The Campbell diagrams are shown in Figures 7-6 (0° blade pitch) and 7-7 (90° blade pitch).

The critical modes are as follows:

(a) Mode 5 (0° pitch) 3.75 Hz excited by 3 per rev tower shadow during start up and shut down at 75 rpm.

(b) Mode 6 (0° pitch) 4.19 Hz excited by 3 per rev tower shadow during start up and shut down at 84 rpm. This is 8% below the lower operating speed (91 rpm).

(c) Mode 7 (0° pitch) 4.89 Hz excited by tower shadow at 98 rpm. This is 3% above the normal operating range of 91 to 95 rpm. In those speeds the blade pitch will be other than 0° and it reduces the effect.

The critical situations can be overcome by avoiding prolonged operation through critical speeds during start up and shut down and by the good speed control of the rotor.

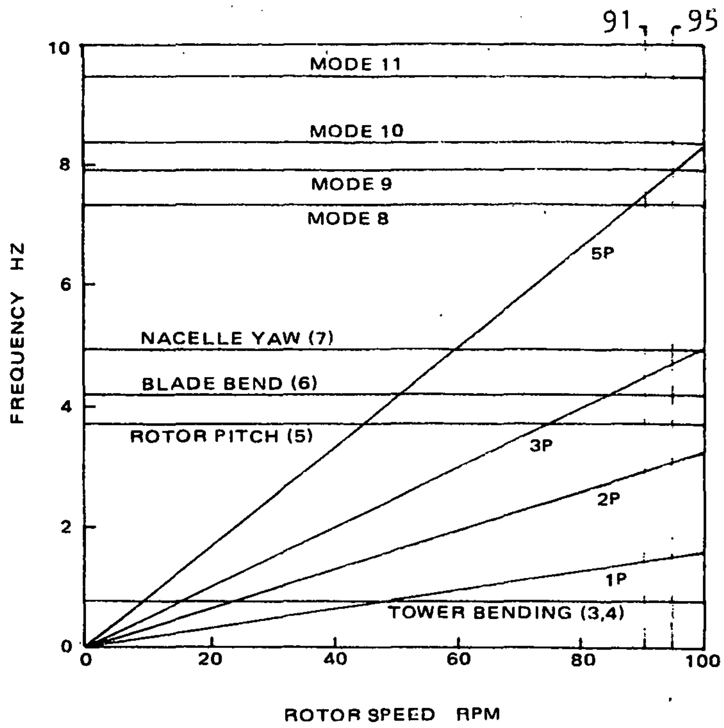


Figure 7-6 CAMPBELL DIAGRAM (0° BLADE PITCH)

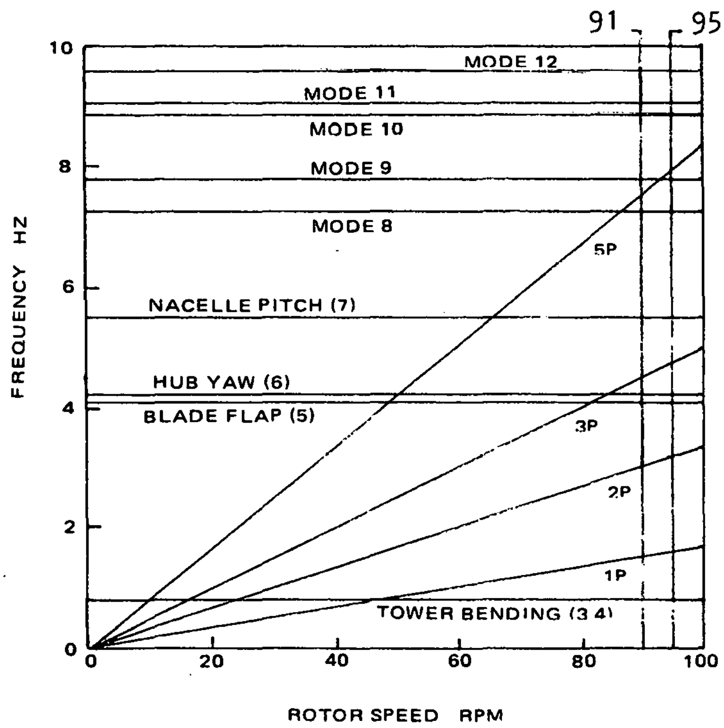


Figure 7-7 CAMPBELL DIAGRAM (90° BLADE PITCH)

8.0 COST ANALYSIS

8.1 HISTORICAL SUMMARY

Table 8-1 summarizes the development of estimates for annual energy production, installed cost and cost of energy from the original proposal to the final design. It can be seen that the predicted energy production remained quite constant while cost estimates increased as the design progressed.

Table 8-1
DEVELOPMENT OF COST OF ENERGY (1)

ITEM	PROPOSAL APRIL, 1979	PDR-2 JUNE, 1980	CDR SEPT., 1980	FDR FEB., 1980
AkWh (2)	15,000	16,059	15,801	15,676
INSTALLED COST (3) $	6,705	7,428	7,428	8,542
AOM COST (4) $	50	140	148	148
COE (5)	6¢	6.8¢	6.9¢	8¢

(1) All costs are in 1978 dollars for the "standard" SWECS design.

(2) Annual kwh at 10 mph mean annual wind speed measured at 30 ft elevation.

(3) Installed cost = hardware cost x 1.5 plus installation cost for 10,000th unit.

(4) AOM = Annual operating and maintenance cost.

(5) COE = Cost of energy, cents per kwh.

8.2 METHODOLOGY

8.2.1 General

The most effective parameter for measuring the worth of a SWECS is the cost of energy at a specified annual average wind speed (AAWS). The SCI 4 kW machine was to be optimized for operation in a 10 mph AAWS, and to produce electrical energy for 6 cents per kilowatt hour under these conditions. Of secondary interest was the cost of energy at 12, 14, 16 and 18 mph AAWS.

8.2.2 Cost of Energy Calculations

Cost of energy estimates were based on the following expressions:

$$COE = \frac{((1.5 \times HWC) + IC) \times (FCR) + (AOM)}{AkWh}$$

WHERE:
- HWC = Hardware cost, dollars. This is the FOB cost multiplied by 1.5 to account for transportation costs, dealer's markup, etc.
- IC = Installation costs, dollars, including those for foundations and site preparations.
- FCR = Annual fixed charge rate (0.129).
- AOM = Uniform annual operation and maintenance costs, dollars.
- AkWh = Annual kilowatt hours produced, when operating at 95% availability.

8.3 HARDWARE COSTS

The cost of hardware for the 4 kW SWECS was based on the 10,000th production unit manufactured at a rate of 10,000 units per year.

8.3.1 Purchased Parts

The cost of purchased parts was based on vendor quotes whenever possible. These were found to consistently follow a learning curve of 95%. This value was reduced to 92.5% in the SCI estimates to account for evolutionary design changes and the fact that the simple parts would eventually be manufactured by the SWECS manufacturer, thus eliminating vendor profit and overhead.

8.3.2 Raw Materials

The cost of raw materials was based on vendor quotes and, for composite materials, SCI in-house cost data.

8.3.3 Tooling Costs

Costs for mandrels and molds for composite components were based on vendor quotes.

8.3.4 Fabrication Costs

Fabrication costs for composite components were based on SCI experience and performance records. Costs for fabricating each metal part were estimated by finding:

Size, weight, shape and type material.
Machining operations required.
Cutting and welding operations required.

Each operation was then assigned a cost, based on unit values obtained from commercial shops and SCI experience. Final results were checked on a dollars per pound basis.

This approach was used to find the cost of the first production unit. A learning curve was then applied to find the cost of the 1000th and 10,000th unit. A learning curve of 87% was selected for fabricated parts as representing the average for composite and metallic components.

8.3.5 Labor Costs

Labor costs for hardware items include those for SCI manufactured composite components plus those for final assembly, quality control, shop testing and preparation for shipment of the assembled SWECS. Labor rates assigned to each operation were selected on the basis of the skill required and the SCI rates in effect for that particular skill.

8.3.6 Summary of Hardware Costs

Shown in Table 8-4 is a summary of estimated hardware costs for the 1st, 1000th and 10,000th unit.

Table 8-2 HARDWARE COST SUMMARY $ (1)

COMPONENT	1ST UNIT	1,000th UNIT (2)	10,000TH UNIT (2)
MECHANICAL ASSEMBLY & NACELLE	5,747	1,949	1,387
ROTOR, BLADES & SPINNER	4,304	1,163	760
ELECTRICAL POWER & CONTROL	1,387	638	492
TOWER, YAW ASSEMBLY & ERECTION SYSTEM	3,733	1,063	710
TOTAL PURCHASED MATERIALS	15,171	4,813	3,349
LABOR HOURS (HR)	378.5	94.4	59.4
LABOR COST	1,794	447	282
DLO @ 145%	2,601	648	409
G & A @ 15%	2,935	886	606
TOTAL COST LESS FEE	22,501	6,794	4,646
TOTAL COST (10% FEE)	24,751	7,474	5,111

(1) 1978 Dollars for the "standard" SWECS design.

(2) Costs are based on the following learning curves:

Purchased Parts - 92.5%

Fabricated components and shop labor - 87%

Overall - 89.3%

8.4 INSTALLATION COSTS

The cost of installation was based on known costs to install similar equipment and information available from other SWECS installations. In 1978 dollars, these are estimated to be $1,120 for the first installation, $916 for the 1,000th and $876 for the 10,000th installation.

8.5 INSTALLED COST

Table 8-3 shows development of installed cost for the 1st, 1000th and 10,000th production units.

Table 8-3 INSTALLED COST $ (1)

	1ST UNIT	1,000TH UNIT	10,000TH UNIT
HARDWARE FOB	24,751	7,474	5,111
x 1.5	37,125	11,211	7,666
INSTALLATION	1,120	916	876
TOTAL INSTALLED COST	38,245	12,127	8,542

(1) 1978 Dollars for the "standard" SWECS design.

8.6 ANNUAL OPERATING AND MAINTENANCE COSTS

The cost for routine annual operating and maintenence (AOM) was based on estimates that one service call would be required every two years. Labor and material costs for machine failures were based on calculations which showed 97.4% availability for the SWECS. Annual O & M costs are summarized in Table 8-4.

Table 8-4

ANNUAL OPERATING AND MAINTENANCE COSTS $ (1)

EVENT	ANNUAL COST
REGULAR PREVENTIVE MAINTENANCE	55
FAILURE REPAIR LABOR COST	43
FAILURE MATERIAL COST	50
TOTAL AOM COSTS	148

(1) 1978 Dollars

8.7 ANNUAL KILOWATT - HOURS

8.7.1 Wind Speed Distribution

Annual kilowatt hours (AkWh) were calculated using the Rayleigh wind speed distribution curves, represented by the following equation

$$H = 8766\epsilon^{-\left(\frac{\pi}{4}\left(\frac{v}{\bar{v}}\right)^2\right)}$$

WHERE:

v = wind speed

$\bar{v}$ = mean annual wind speed

H = number of hours per year in which wind speed is equal to or greater than v

8.7.2 Wind Speed Variation With Height

The following equation was used to determine wind speed at different heights:

$$\frac{v}{V_Z} = \left(\frac{h}{h_Z}\right)^{1/7}$$

WHERE:

v = wind speed at height h

V_Z = wind speed at height h_Z

The above relationship was used for SWECS energy and output performance only. More severe wind shear conditions were assumed for SWECS structural load calculations.

8.7.3 Standard Conditions

In determining annual kilowatt hour production, the following conditions were assumed:

- Steady-state winds only
- No losses for directional wind changes
- Standard sea-level conditions
- 95% SWECS availability

8.7.4 Efficiency Of Generator And Gear Box

The gear box efficiency was assumed to be constant at 94%. The generator efficiency varies with load and was assumed to follow the curve of Figure 8-1.

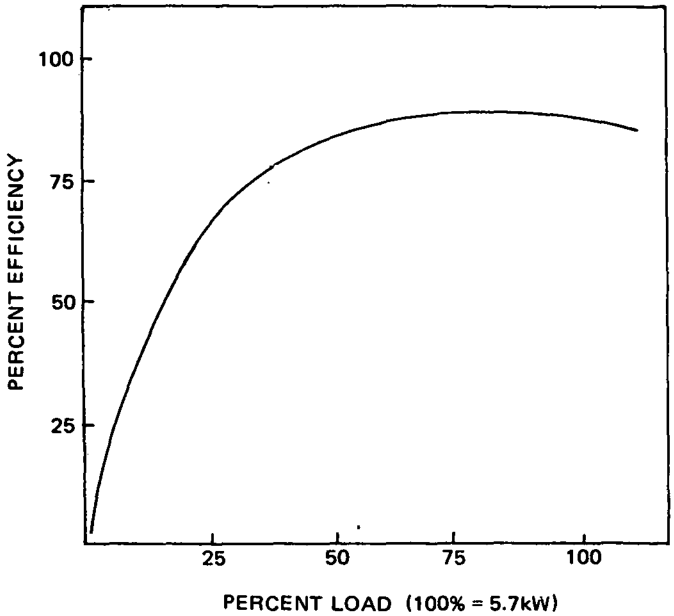


Figure 8-1 GENERATOR EFFICIENCY

8.7.5 Annual Production

Annual production, at a range of mean annual wind speeds, is summarized in Table 8-5.

Table 8-5
ANNUAL KILOWATT HOURS PRODUCTION

$\bar{v}$ (1) mph	AkWh (2)
8	8,732
9	12,207
10	15,676
12	22,041
14	27,156
16	30,740
18	32,801

(1) $\bar{v}$ = Mean annual wind speed measured at 30 ft elevation.

(2) Annual kilowatt hours corrected for 95% availability and control system losses.

8.8 COST OF ENERGY

Results of cost of energy calculations for the "standard" production unit are given in Table 8-6. Figure 8-2 summarizes these data in graphical form for the 10,000th production unit. It is seen that the 6 cents per kWh target requires a mean annual speed of 12 mph.

Table 8-6 COST OF ENERGY $ (1)

	1ST UNIT	1,000TH UNIT	10,000TH UNIT
INSTALLED COST	38,245	12,127	8,542
ANNUAL OPERATING & MAINTENANCE	148	148	148
COST OF ENERGY, $/kWh			
(2) $\bar{v}$ = 10 mph	0.324	0.109	0.080
$\bar{v}$ = 12 mph	0.230	0.078	0.057
$\bar{v}$ = 14 mph	0.187	0.063	0.046
$\bar{v}$ = 16 mph	0.165	0.056	0.041
$\bar{v}$ = 18 mph	0.15	0.052	0.038

(1) 1978 Dollars for "standard" production unit.

(2) Mean annual wind speed measured at 30 ft elevation.

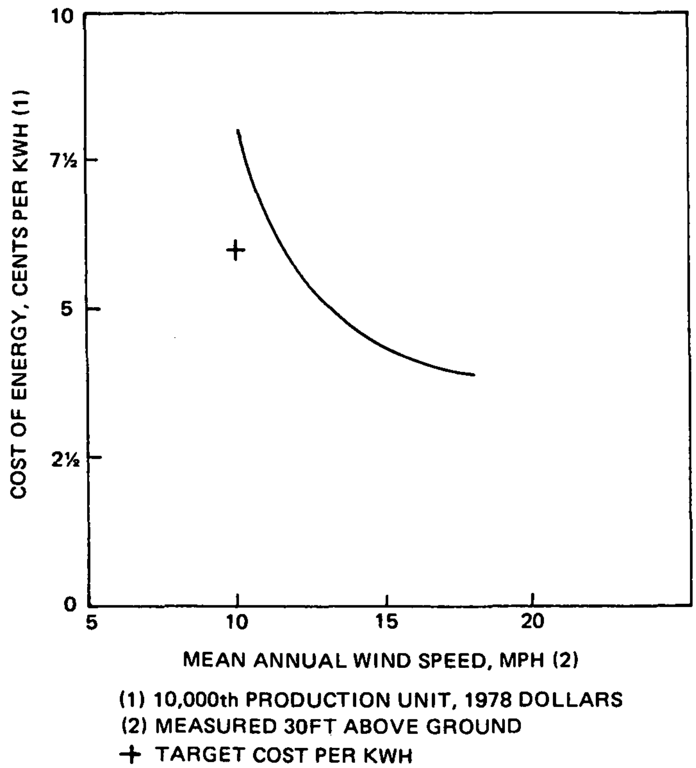


Figure 8-2 COST OF ENERGY VS MEAN ANNUAL WIND SPEED

8.9 WEIGHT AND COST SUMMARY

A summary of weights and costs, for both the "standard" and "special" SWECS designs is given in Table 8-7. Costs are for the first production unit. The "special" design requires a heavier tower and extensive protection for salt water spray environment, as covered in Reference 19.

Table 8-7 WEIGHT AND COST SUMMARY

	WEIGHT (lb)		LABOR (mph)		MATERIAL COST $	
	STD	SPECIAL	STD	SPECIAL	STD	SPECIAL
MECHANICAL ASSEMBLY & NACELLE	904	904	117.6	119.6	5,747	6,395
ROTOR, BLADES & SPINNER	660	688	150	151	4,304	4,611
SUB-TOTAL ATOP TOWER	1564	1592	267.6	270.6	10,051	11,006
ELECTRICAL POWER & CONTROL	75	90	45	48	1,387	1,751
TOWER, YAW & ERECTION SYSTEM	1720	1932	65.9	69.9	3,733	3,945
SWECS TOTAL	3359	3614	378.5	388.5	15,171	16,702
G & A @ 15%					2,276	2,505
FEE @ 10%					1,745	1,921
TOTAL COST					19,192	21,128

(1) 1978 Dollars, first production unit, FOB Azusa, California

9.0 AVAILABILITY, FMEA, MAINTAINABILITY AND SAFETY ANALYSES

9.1 AVAILABILITY METHODOLOGY

Availability of the SWECS was determined by estimating failure rates and repair times for each major component and subsystem. The availability was then found from the following expressions:

$$\text{AVAILABILITY} = \frac{\text{MTBF}}{\text{MTBF} + \text{MTTR}}$$

$$\text{MTBF} = \left(\sum_{i=1}^{n} \lambda_{ai} \right)^{-1} \quad \text{(mean time between failures, hours)}$$

λ_{ai} = PREDICTED FAILURE OF THE i^{th} PART IN FAILURE/HOUR

n = TOTAL NUMBER OF PARTS

$$\text{MTTR} = \frac{\sum \left(\lambda_b R_p \right)}{\sum \lambda_b} \quad \text{(mean time to repair, hours)}$$

λ_b = SUBSYSTEM PREDICTED FAILURE RATE IN HOURS

R_p = REPAIR TIME REQUIRED TO PERFORM REPAIR IN HOURS

9.2 AVAILABILITY RESULTS

Results of the availability analyses for the SWECS as a whole, are as follows:

MTBF	17,300 HOURS
MTTR	453 HOURS
AVAILABILITY	97.4 PERCENT

9.3 FAILURE MODE AND EFFECTS ANALYSIS

Results of a failure mode and effects analysis (FMEA), for the complete SWECS, are summarized in Table 9-1.

Table 9-1 FMEA

FAILURE MODE	POSSIBLE CAUSE	EFFECT	PREVENTIVE OR CORRECTIVE ACTION
STRUCTURAL FAILURE OF BLADES	FATIGUE OR BUCKLING	VIBRATION PROBLEM. POSSIBLE FURTHER DAMAGE	SAFE LIFE DESIGN OF BLADES. SWECS SHUT DOWN AND FEATHERED BY VIBRATION SWITCH OR HIGH WIND SPEED. (REQUIRES MANUAL RESET)
UNBALANCED BLADES	ICE BUILD-UP ON BLADES	DAMAGE TO SWECS REMOTE POSSIBILITY OF INJURY TO PUBLIC FROM FLYING ICE	SWECS SHUT DOWN AND FEATHERED BY VIBRATION SWITCH. (REQUIRES MANUAL RESET)
GEAR OR COUPLING FAILURE	MISALIGNMENT	OVERSPEED OF ROTOR. LOSS OF POWER OUTPUT.	SAFE LIFE DESIGN. ANTI-MOTORING RELAY AND OVERSPEED SWITCH SHUTS DOWN AND FEATHERS SWECS. (REQUIRES MANUAL RESET)
GEAR BOX SEIZURE	BEARING FAILURE. OIL LEAK.	GRADUAL SLOWING DOWN UNTIL GENERATOR TRIES TO MOTOR.	SAFE LIFE DESIGN. REGULAR PREVENTIVE MAINTENANCE. SWECS SHUT DOWN AND FEATHERED BY ANTI-MOTORING RELAY.
GENERATOR OVERHEATS	PITCH CONTROL MALFUNCTION	GENERATOR DAMAGE	SWECS SHUT DOWN AND FEATHERED BY THERMAL OVERLOAD RELAY.
GENERATOR WINDING SHORT CIRCUIT	OVERHEATED GENERATOR	LOSS OF POWER. HIGH CURRENT SURGE FROM LINE	SWECS SHUT DOWN AND FEATHERED BY CURRENT LIMITING FUSE.
GENERATOR WINDINGS OPEN CIRCUIT	FATIGUE DUE TO TEMPERATURE CYCLING	OVERSPEED AND LOSS OF POWER.	OVERSPEED SWITCH SHUTS DOWN AND FEATHERS SWECS.
UTILITY INTERCONNECTION LOST	SHORT CIRCUIT ON UTILITY LINES	POSSIBLE SELF EXCITATION OF GENERATOR AND OVERVOLTAGE ON INTERCONNECTED SYSTEM.	SWECS SHUT DOWN AND FEATHERED BY OVER/UNDER FREQUENCY RELAY.

Table 9-1 FMEA (continued)			
FAILURE MODE	POSSIBLE CAUSE	EFFECT	PREVENTIVE OR CORRECTIVE ACTION
MAIN PITCH ROD BREAKS	MISALIGNMENT. FATIGUE.	LOSS OF AUTOMATIC PITCH AND FEATHER.	SAFE LIFE DESIGN. BLADES FEATHER AERODYNAMICALLY, UNLOADING SWECS.
LOSS OF CONTROL POWER	SHORT CIRCUIT.	LOSS OF SWECS CONTROLS.	FEATHER AND SHUT DOWN OF SWECS. BY FAIL SAFE SOLENOID.
PITCH ARM ON ONE BLADE BREAKS	FATIGUE	ONE BLADE MAY FLUTTER.	SWECS FEATHERED AND SHUT DOWN BY VIBRATION SWITCH.
FEATHERING ACTUATOR FAILS	MECHANICAL DAMAGE.	LOSS OF FEATHER. POSSIBLE OVERSPEED.	FAIL SAFE MECHANICAL DESIGN OF ACTUATOR.
VIBRATION SWITCH FAILURE	MOISTURE ACCUMULATING AND FREEZING, DAMAGING SWITCH.	POSSIBLE LOSS OF BLADE IF EXCESS VIBRATION OCCURS.	SAFE LIFE DESIGN OF VIBRATION SWITCH.
OVERSPEED SWITCH FAILURE	MOISTURE ACCUMULATING AND FREEZING, DAMAGING SWITCH.	POSSIBLE OVERSPEED OF ROTOR AND LOSS OF BLADE.	SAFE LIFE DESIGN OF OVERSPEED SWITCH.
CURRENT LIMITING FUSE FAILS	FAULTY EQUIPMENT.	GENERATOR DAMAGE IF GENERATOR WINDINGS SHORT CIRCUIT.	POWER COMPANY'S BACKUP CIRCUIT BREAKER TRIPS, SHUTTING DOWN AND FEATHERING SWECS.
THERMAL OVERLOAD RELAY FAILS	FAULTY EQUIPMENT.	GENERATOR DAMAGE IF GENERATOR OVERHEATS.	BACKUP THERMAL SWITCH IN GENERATOR SHUTS DOWN AND FEATHERS SWECS.
YAW BEARING SEIZURE	INSUFFICIENT LUBRICATION. CONTAMINATION.	NO RESPONSE TO CHANGES IN WIND DIRECTION. POSSIBLE REVERSED ROTATION OF ROTOR.	SWECS FEATHERED AND SHUT DOWN BY ANTI-MOTORING RELAY.

Table 9-1 FMEA (continued)

FAILURE MODE	POSSIBLE CAUSE	EFFECT	PREVENTIVE OR CORRECTIVE ACTION
POWER SPRING FAILS	FATIGUE	SWECS FEATHERED AND SHUT DOWN.	FAIL SAFE DESIGN.
UNFEATHERING SOLENOID FAILS.	SPRING FAILURE.	QUICK RETURN FROM FEATHER. ROTOR MAY NOT PICK UP ENOUGH SPEED TO START UP.	SAFE LIFE DESIGN, REGULAR PREVENTIVE MAINTENANCE. RE-FEATHERING ACCOMPLISHED BY MOTORING GENERATOR.
CHAIN BREAKS.	FATIGUE	PITCH CONTROL LOST. SWECS FEATHERS AERODYNAMICALLY.	FAIL SAFE DESIGN.
TOWER FAILS	LOADS IN EXCESS OF DESIGN LOADS.	DESTRUCTION OF SWECS. DANGER TO PERSONNEL AND PROPERTY.	SAFE LIFE DESIGN OF TOWER.
FOUNDATION FAILS.	EXCESS LOAD.	DESTRUCTION OF SWECS. DANGER TO PERSONNEL AND PROPERTY.	CONSERVATIVE DESIGN FOR LOCAL SOIL CONDITIONS.

9.4 MAINTAINABILITY

Results of a maintainability analysis are summarized in Table 9-2.

Table 9-2 MAINTAINABILITY

REQUIREMENTS	RESOLUTION
EASE OF ACCESS FOR SWECS REPAIR AND MAINTENANCE.	PROVIDE ADEQUATE ACCESS HATCHES. TOWER ERECTION SYSTEM FACILITATES LOWERING OF THE NACELLE AND TOWER FOR MAINTENANCE.
READY AVAILABILITY OF TOOLS AND PARTS.	USE COMMERCIALLY AVAILABLE FASTENERS AND "OFF THE SHELF" REPAIR PARTS WHENEVER POSSIBLE. DEALER/DISTRIBUTOR SYSTEM TO STOCK SPARES.
MAXIMUM PREVENTIVE MAINTENANCE INTERVAL.	USE MOISTURE PREVENTION, BEARING SEALS AND HIGH QUALITY LUBRICANTS. USE OF INERT COMPOSITE MATERIALS. PAINT OR PLATE ALL EXPOSED PARTS. PROVIDE COVERED, DUST-PROTECTED COMPARTMENTS FOR BALL SCREWS.

9.5 SAFETY ANALYSIS

Table 9-3 covers SWECS safety requirements and the method of satisfying these requirements. The applicable OSHA section is also listed.

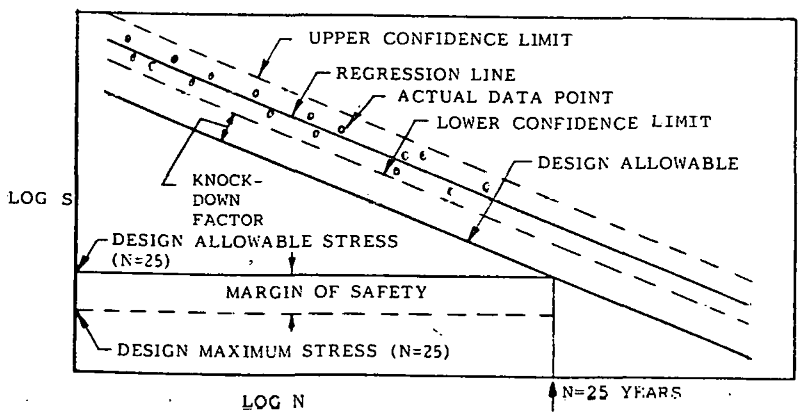
Table 9-3 SAFETY

O.S.H.A. SECTION NUMBER	REQUIREMENT	PREVENTIVE ACTION
7707.2	HEAD PROTECTION	WEAR HARD HATS DURING ERECTION OF TOWER.
7722.3	ELECTRICAL. REFERS TO NEC FOR MINIMUM REQUIREMENTS. ALL EQUIPMENT MUST BE "APPROVED". NEC DEFINITION OF APPROVED IS TESTED AND APPROVED BY U.L. OR OTHER NATIONALLY RECOGNIZED TESTING INSTITUTIONS.	USE U.L. OR OTHER APPROVED COMPONENTS IN ELECTRICAL SYSTEM WHERE AVAILABLE.
7733.7	HOIST AND WIRE ROPE RATINGS SUITABLE FOR LOADS CARRIED.	USE FACTOR OF SAFETY OF 5 FOR ALL ERECTION EQUIPMENT.
7751.5	DE-ENERGIZE LINES AND EQUIPMENT WHILE WORKING ON SYSTEM.	LOCKOUT SWITCH WITH UTILITY INTERFACE.
7753.6	METAL TOWER CONSTRUCTION (A) GUY LINES (B) CLEARANCE OF POWER LINES	COMPOSITE TOWER IS NON-CONDUCTING. SAFE LIFE DESIGN OF HINGE AND ERECTION SYSTEM. SAFE LOCATION OF SWECS.

9.6 SAFE LIFE DESIGN

The term "safe life design" for composite materials is defined in Figure 9-1.

Figure 9-1 SAFE LIFE DESIGN DETERMINATION FOR COMPOSITE MATERIALS



10.0 COMPONENT DEVELOPMENT

10.1 DEVELOPMENTAL ACTIVITIES

The following developmental activities were accomplished during the Phase I effort:

1. Tests of single phase induction generators.
2. "Breadboard" testing of electro-mechanical control system.
3. Blade fabrication process development (not completed).

10.2 INDUCTION GENERATOR TESTS

10.2.1 Scope

In order to verify performance estimates of single phase induction generators, it was decided to perform tests of representative standard machines while generating directly into the power lines. With the approval of Rockwell International, a series of such tests were performed by the California State Polytechnic University, Pomona. Complete test results are included in Reference 20. The objectives were to determine efficiency and power factor from no load to 125 percent full load for the conditions covered below. These conditions were selected so as to provide the following practical design information:

- When a standard single-phase motor with published performance is selected, its performance as a generator can then be estimated from the test results.
- When a three-phase motor with published three phase performance is selected, its performance as a single-phase generator can then be estimated from the test results.

The three-phase induction motor was included because it is readily available in multi-speed designs and in much larger ratings. It is also often much less expensive in the larger sizes because of the huge industrial market for three-phase machines.

10.2.2 <u>5 HP Single-Phase Capacitor Run Induction Motor</u>

A. Operated as a motor.

B. Operated as an induction generator.

10.2.3 <u>7 1/2 HP Three Phase Induction Motor</u>

A. Operated as a three-phase motor.

B. Operated as a three-phase generator.

C. Operated as a single-phase motor with the unused winding connected to the line through various values of capacitance.

D. Operated as a single-phase generator with the unused winding connected to the line through various values of capacitance.

10.2.4 <u>Significant Results</u>

The results below show dramatic improvement in efficiency when the single-phase induction generators were driven backwards. Backwards, in this case, refers to the direction opposite to that which the machine wants to turn as a motor.

- SINGLE-PHASE CAPACITOR RUN MOTOR

A.	Peak efficiency as a motor	78.4%
B.	Peak efficiency as a generator	
	Normal Rotation	80.8%
	Reverse Rotation	89.0%

- THREE-PHASE INDUCTION MOTOR

A.	Peak efficiency as a three-phase motor	84.0%
B.	Peak efficiency as a one-phase generator	
	Normal Rotation	73.4%
	Reverse Rotation	83.7%

10.2.5 Backward Rotation

An explanation of observed results, when driving a single-phase induction motor in the "backward" direction, is included in Reference 20. Briefly, the phenomenon is believed to result from the action of positive and negative rotating fields present in all single-phase induction machines. When driven above synchronous speed in the "backward" direction, the torques developed by these two fields augment each other. Conversely, in the "forward" direction, the two fields produce opposing torques.

10.2.6 Other Results

Below are listed other results which are considered important to the 4 kW SWECS design:

- Power factor is improved significantly by "backward" operation of a single-phase capacitor-run generator.
- Tests confirm the following method to determine generator output kW to match the single-phase motor HP rating:

$$\text{Output kW} = \frac{\text{Rated HP} \times .746 \times \text{Gen. P.F.}}{\text{Motor eff.} \times \text{Motor P.F.}}$$

Calculated output - 4.4 kW

Tested output = 4.2 kW* at rated current

- Similarly, the following expression has been used for a three-phase motor operating as a single-phase generator:

$$\text{Output kW} = \frac{2}{3} \; \frac{\text{Rated HP} \times .746 \times \text{Gen. P.F.}}{\text{Motor eff.} \times \text{Motor P.F.}}$$

Calculated output - 4.86 kW
Tested output = 4.74 kW at rated current*

* Interpolated between test points and corrected for 230 volts

10.3 BREADBOARD TESTS

10.3.1 Objectives

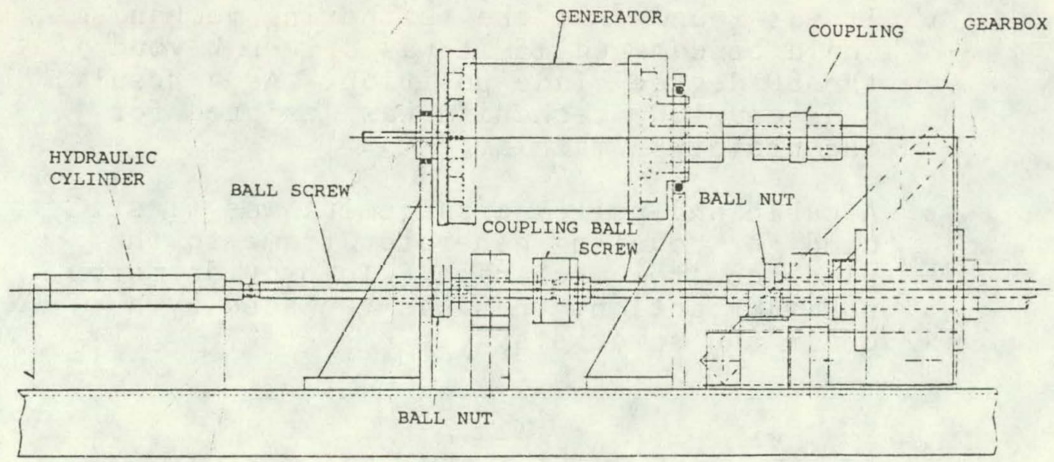
A decision was made early in the Phase I effort to produce a mockup (breadboard) of the proposed electro-mechanical control system. Objectives of the breadboard tests were:

- To demonstrate operation of the torque-actuated blade pitch control system when generating directly into the utility grid.
- To demonstrate emergency blade feathering using stored energy in the spinning rotor (rotor simulated by a variable speed drive).
- To charge a spring during emergency feathering and to demonstrate that blade pitch could be ramped to operating position during startup, using the energy stored in the spring.

10.3.2 Breadboard Layout

Basic design of the breadboard is shown in Figure 10-1. This unit was constructed, tested, modified and re-tested over a 6-month period.

Figure 10-1 BREADBOARD LAYOUT



10.3.3 Breadboard Operation

The SWECS rotor was simulated by means of a 15 HP variable speed drive, coupled to the SWECS main shaft by means of a chain and sprocket. The induction generator was driven through a 20 $\div$ 1 speed-increasing gear box and electrically connected to the utility grid.

By adjusting the variable speed driver, it was possible to cause the induction generator to run as a motor or generator, feeding up to 4 kW into the utility grid.

10.3.4 Results

Performance of the breadboard was found to be satisfactory though the following problems were uncovered and corrected in the final design:

- Withdrawal of solenoid actuator, following feathering, was not possible. This was corrected by selecting a more powerful solenoid (which required a large current to withdraw and a small current to hold in the withdrawn position) and a mechanical advantage.

- The hydraulic damper produced a large frictional drag on the system. It also developed high forces in the pitch control mechanism during feathering. The cylinder was replaced with an eddy-current damper coupled directly to the generator frame.

- It was found that the feathering mechanism could be damaged if it was driven beyond the 90-degree blade position. As a result, a de-coupling mechanism was designed for the prototype machine.

- A cable and pulley arrangement was first used for coupling generator frame to the pitch control mechanism. To provide more positive action, this was replaced by a chain and sprockets.

10.4 BLADE FABRICATION PROCESS DEVELOPMENT

Construction of a 5-foot long blade test section was planned, to help perfect the fabrication process prior to prototype fabrication. A test mandrel and set of blade molds were designed and purchased for this purpose but fabrication of the test blade was not completed for the final design review.

11.0 TESTS AND INSTRUMENTATION

11.1 SCOPE

As a part of the Phase I effort, it was necessary to develop a detailed plan for preliminary testing and instrumentation of the prototype prior to its shipment to Rocky Flats. A suitable site for these tests also had to be selected, as the winds at SCI's facilities were found to be inadequate.

11.2 TEST PLAN AND PROCEDURE

A detailed test plan and procedure was prepared (Reference 9). A list of equipment and instrumentation required and a list of spare parts deemed to be necessary to complete the SCI tests was included.

11.3 TEST SITE SELECTION

The San Gorgonio Pass, near Palm Springs, California was selected as the general site for prototype testing. To obtain a specific site, several site surveys were performed (Reference 10).

Southern California Edison's Devers substation was found to be the most suitable site in the San Gorgonio Pass. It offers high mean annual wind speed of 15 mph and the best security against vandalism. After lengthy negotiations, Southern California Edison agreed to make this site available to SCI for testing of the 4 kW prototype machine.

12.0 CONCLUSIONS AND RECOMMENDATIONS

The Phase I design and analysis resulted in a 4 kW SWECS design which met most program objectives. The resulting design had some unique features:

- Composite Tower and Blades
- Torque-Actuated Blade Pitch Control

It is recommended that the Phase II fabrication, testing and evaluation of the SCI final design be continued in order to confirm and debug the Phase I design. This is an essential step toward the final goal of commercialization of this promising SWECS design.

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