MOD-1 WTG DYNAMIC ANALYSIS

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ABSTRACT

This paper discusses the dynamic analysis of the MOD-1 2000 KW horizontal-axis wind turbine. After briefly describing the MOD-1 design, the dynamic analysis used to evaluate the dynamic loads and structual interactions is discussed. The resonant frequency placement, the treatment of unsteady wind loading and the dynamic load sensitivity to frequency shifts are reviewed for the design.

MOD-1 DESCRIPTION

As shown in Figure 1, the MOD-1 WTG incorporates a two-bladed, downwind rotor driving an AC generator through a speed increaser atop a steel, truss-type tower. The major characteristics of the MOD-1 WTG are summarized in Figure 2; its major elements are briefly described as follows:

a. Rotor: Two variable-pitch, steel blades are attached to the hub barrel via three-row roller bearings which permit about 105 deg. pitch excursion from full feather to max. power. Blade pitch is controlled by hydraulic actuators which provide a maximum pitch rate of 14 deg/sec.

   The hub tailshaft provides the connection to the low-speed shaft and to the dual-tapered-roller main bearing, which supports the rotor and one end of the low-speed shaft.

b. Drive Train: Comprising the drive train are the low-speed shaft and couplings, the gearbox, and the high-speed shaft/slip-clutch which drives the generator. (Refer to Figure 3.) The slip-clutch precludes excessive torques from developing in the entire drive train due to extreme wind gusts and/or faulty synchronization.

c. Power Generation/Control: A General Electric synchronous AC generator is driven at 1800 RPM through the high-speed shaft. A shaft-mounted, brushless exciter controlled by a solid-state regulator and power stabilizer inputs provides voltage control.

d. Nacelle Structure: The core of the nacelle structure is the welded steel bedplate. All nacelle equipments and the rotor are supported by the bedplate, which provides the load path from the rotor to the yaw structure. Other equipments supported by the bedplate are the pitch control and yaw drive hydraulic packages, walkways, oil coolers, heaters, hydraulic plumbing, electronics boxes, cabling and the fairing. Redundant instrumentation booms, with wind speed, temperature and direction sensors are mounted on the upwind end of the fairing.

e. Yaw Drive: Unlimited yaw rotation is provided by the yaw drive system, comprising the upper and lower yaw structures, the two-motor hydraulic drive, the hydraulic yaw brake, and the large cross-roller yaw bearing. The yaw drive is capable of yawing the rotor/nacelle at 0.25 deg/sec. To provide adequate yaw drive stiffness, the yaw brake is fully activiated when not in a yaw maneuver and partially activated during the maneuver to avoid backlash in the yaw drive gear train.

f. Tower: The truss tower, as shown in Figure 4, is made up of seven vertical bays, including the base and top (pintle) sections. Tubular steel columns are used at the four corners to carry the main loads. Back-to-back channels serve as cross-members where loads permit. However, in most bays, tube-section cross-members are still required because of high loads and to reduce "tower shadow." Access to the yaw drive and nacelle area is provided via a cable-guided, gondola-type elevator.

g. WTG Weight: The final system weight (rotor, nacelle and tower) is expected to be about 650,000 lbs. This weight breaks down as shown in Figure 5.

GETSS COMPUTER CODE

The GETSS (GE Turbine System Synthesis) code was used to evaluate the dynamic loads of the complex MOD-1 WTG. Key objectives considered in the development of the code were: (1) to evaluate resonant frequency placements so that dynamic loads would be minimized, (2) to accurately determine the loads throughout the system so that adequate but not excessive design margins are provided, and (3) to determine the sensitivity to stiffness variations so that critical parameters can be carefllly controlled. By minimizing dynamic interactions, dynamic loading can be alleviated assuring a long life design.

The analytical approach used in GETSS is to modal synthesize the system at various rotor positions and to analyze the system in a piecewise linear manner. The WTG system is analyzed as six major substructures, Figure 6, using NASTRAN finite element models except for the blades which use a turbine blade code. These models serve the dual purpose of stress and dynamic analysis. The system is then synthesized from the substructure modes using a stiffness coupling synthesis code that includes the dynamic transformation. The stiffness matrices coupling the substructures together represent the bearing stiffnesses. The system is then analyzed at 45 degree rotor increments and the modal coordinates switched from model to model as the rotor turns. Using this approach the modal characteristics can readily be traced to the substructures and the modes contributing to the loads identified. The loads at various blade stations, the hub, the main rotor bearing, the yaw bearing and the tower base are determined for subsequent structural analysis. Accelerations and deflections at selected critical locations are also evaluated.

The code uses quasi-steady aerodynamics to evaluate the loads due to wind shear, tower shadow and gusts. The flow field includes a wind shear following the power law, Figure 7, and a three dimensional tower shadow that follows the tower geometry. The three dimensional tower shadow representation permits the sequential entering and exiting of the various blade stations into the retarded flow region.

GETSS CODE VERIFICATION

Prior to analyzing the MOD-1, the GETSS computer code was verified by analyzing the MOD-0 WTG for two operating conditions and comparing the analysis results with actual measurements, Reference 1. The comparisons included flatwise and chordwise bending moments at two blade stations, main shaft bending moments and torque, and tower accelerations and deflections. The comparison included waveform, harmonic content, peak and cyclic amplitudes. NASTRAN models of the tower and bedplate, similar to those used for MOD-1, were developed including the stairway and elevator rails, Figures 8 and 9. Excellent agreement of the predicted modes and resonant frequencies with modal test results verified the adequacy of the modeling of most of the structure. A typical comparison of the flatwise bending moment at the blade root is shown in Figure 10. More than 90 percent of the loads, accelerations and deflections were within 20 percent of the range of measured values. In general, the tower deflections and accelerations tended to be conservative. The waveform and harmonic content were reproduced and, in one condition, captured a 10P tower response caused by a tower stairway coupled mode. The tower shadow for each condition was based on NASA-LeRC wind tunnel duplicating the wind and rotor orientation relative to the tower. The good correlation of the analytical predictions with MOD-0 measurements permitted the use of minimum design margins for MOD-1.

MOD-1 RESONANT FREQUENCY PLACEMENT

The analysis of the MOD-1 used the same procedures described above. The NASTRAN tower and bedplate models are shown in Figures 11 and 12. The resulting resonant frequencies are shown in Figure 13. The lowest resonance results primarily from the effective stiffness of the electrical generation system. Blade flatwise bending is placed at approximately 2.6P and is below the tower lateral bending modes at approximately 3.0P. The first edgewise bending mode of the blade occurs at approximately 4.5P while tower torsion is above 8P. Blade torsion is at approximately 14.7P.

The sensitivity of the dynamic loads to variations in selected system parameters was investigated, Figure 14. Large variations in the soil stiffness, yaw bearing stiffness and shaft bearing stiffness were not found to significantly affect the loads. The position of the bedplate C.G. did not have a significant influence on the dynamic loads but, because of the weight, does have a significant effect on the tower loading. Variations on the order of 50 percent in the main rotor bearing and the blade retention bearing stiffness were, however, found to significantly influence the dyanmic loads. Critical to the dynamic loading are the yaw drive stiffness and twoer to blade frequency placements which can result in large load amplifications.

DESIGN LOAD DEFINITION

Design loads are determined considering the peak and cyclic loading that occurs for all operational environments throughout the life of the machine, Figure 15. Of major significance is the variation in the loads at a "constant" wind speed. Using a wind turbulance model and the WTG dynamic model, the dispersed loads are evaluated considering gust effects. Peak and cyclic load distributions are determined for nominal operating conditions by calculating the loads due to turbulence using a discrete gust analysis. MOD-0 load measurements were used to evaluate the adequacy of the load dispersions and correlate closely for flap bending moments on the blades. Significant load reductions are achieved by limiting the maximum operating wind velocity to a nominal 35 mph and an instantaneous value of 50 mph. Both positive and negative gusts are considered.

Additional design conditions include hurricane force winds and overspeed conditions due to desynchronism of the generator. Torsional loading of the system is dependent on pitch control system characteristics and is discussed in Reference 2.

CONCLUSIONS

Dynamic analysis of large horizontal-axis WTG systems involves complex structural interactions which can significantly affect dynamic loading and the resulting design requirements. For the MOD-1 WTG design a comprehensive analytical treatment of the loads throughout the operating regime was used to assure adequate but realistic design margins and to enable interactions and sensitivities to be examined. Because of the early state of development of WTG technology, MOD-0 load measurements were essential to the evaluation of the analytical code and provide needed guidance to the determination of MOD-1 design loads and conditions.

REFERENCES

1. Spera, D. A., "Comparison of Computer Code for Calculating Dynamic Loads in Wind Turbines", Proceeding of the 1977 Wind Turbine Structural Dynamics Workshop, November 15-17, 1977, Cleveland, Ohio.

2. Barton, R., "MOD-1 WTG Analysis", Proceedings of the 1977 Wind Turbine Structural Dynamics Workshop, November 15-17, 1977, Cleveland, Ohio.

DISCUSSION

Q. Did you connect the analyses for the four rotor positions?

A. The analysis is performed using piecewise linear models for four rotor positions using a total of eight models during each revolution. Continuity of deflection and velocity is maintained at the model switching points.

Q. Did you compare results with more than four rotor positions?

A. Because of the good comparison with experimental results using the four rotor position models, a finer analysis using additional rotor positions was not performed.

Q. Why did you calculate frequencies at different blade azimuths and what significance does it have?

A. The analysis was performed using a piecewise linear modal analysis combining the solutions for a series of blade positions, i.e., the modes at eight rotor positions per revolution were combined to obtain the loads during a revolution. Because the system dynamic characteristics change with rotor azimuth, a system modal analysis is required for each range of rotor azimuth angles. By using this type of analysis, the dynamic loads are readily traced to the modal characteristics of the system and corrective action to reduce dynamic load magnification can be determined.

Q. How are the maximum loads determined?

A. The maximum loads are determined by evaluating the steady state loads at prescribed wind conditions during a complete revolution of the rotor. A continuous load time history for all rotor positions is determined. This is then used with solutions for other wind conditions to determine the dispersed cyclic loads and the peak loads.

Q. Please elaborate on your turbulence model and its effect on loads.

A. The turbulence model was used to determine an equivalent gust velocity to use in calculating loads. The bandwidth considered spherical eddies having a size equivalent to one third the rotor radius or larger with pitch change to maintain constant torque. Analysis of partial rotor immersion with gusts of one third the rotor radius showed a negligible change in loading. The final loads analysis considered complete rotor immersion by a gust obtained by integrating the turbulence spectrum.

Q. What factors were used to set the cut out velocity of the MOD-1 at 35 mph (30 foot reference height)?

A. The cut out velocity was established from analysis of the peak and cyclic load variations with wind velocity and its effect on cost. Fatigue was a governing consideration. As the cut-out wind velocity increases the loads increase dramatically. The cut-out velocity of 35 mph was selected on the basis of cost and energy capture trade-offs which indicated a high cost for designing to higher wind velocities with little gain in yearly energy capture.

Q. Since you assessed frequencies and loads parametrically, did you take the opportunity to look at the effect of teetering or articulating the rotor?

A. Early in the design cycle, a preliminary evaluation of the effects of teetering the rotor was performed using the F762 code. The results did not indicate major reductions in blade loads in the most critical regions of the rotor, i.e., at the 50 percent span. The reduction in the loads on the bedplate and towers were not evaluated in depth in that the F762 code is primarily for rotor analysis and dis not determine loads throughout the system.

Q. How much time does a typical load analysis of the MOD-1 WTG require?

A. As the design evolves, changes are mde to the dynamic characteristics of all portions of the system. This includes the blades, bedplate, drive train, yaw drive and tower. Dynamic models of all these components are continually revised during the design process. Having established the dynamic characteristics of the various structural components, a set of ten design loads at critical interfaces throughout the system can be determined in approximately two weeks.

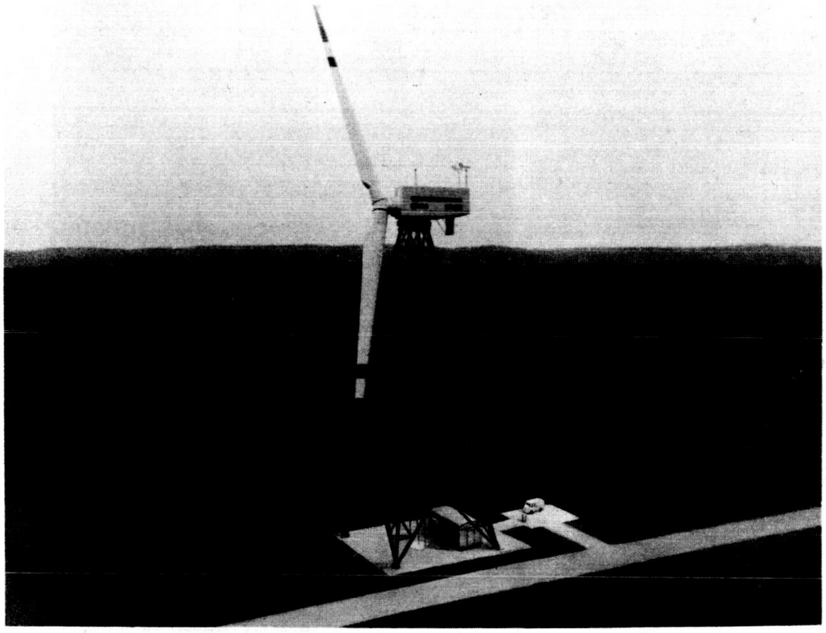

Figure 1. - MOD-1 wind turbine generator.

RATED POWER (SYSTEM)	2000 kW(e)
RATED WIND SPEED	25 MPH } AT 30 FT. ABOVE GROUND
CUT-IN WIND SPEED	11 MPH } AT 30 FT. ABOVE GROUND
CUT-OUT WIND SPEED	35 MPH } AT 30 FT. ABOVE GROUND
SURVIVAL WIND SPEED	150 MPH (AT ROTOR ℄)
CONE ANGLE	12°
INCLINATION OF AXIS	0°
ROTOR SPEED	35 RPM
BLADE DIA.	~ 200 FT
BLADE TWIST	11°
AIRFOIL	NACA 230XX
BLADE-GROUND CLEARANCE	~ 40 FT.
LIFE	30 YEARS WITH MAINTENANCE
ENVIRONMENT	-31°F TO $+120^{\circ}$F

Figure 2. - Major characteristics of MOD-1 wind turbine generator.

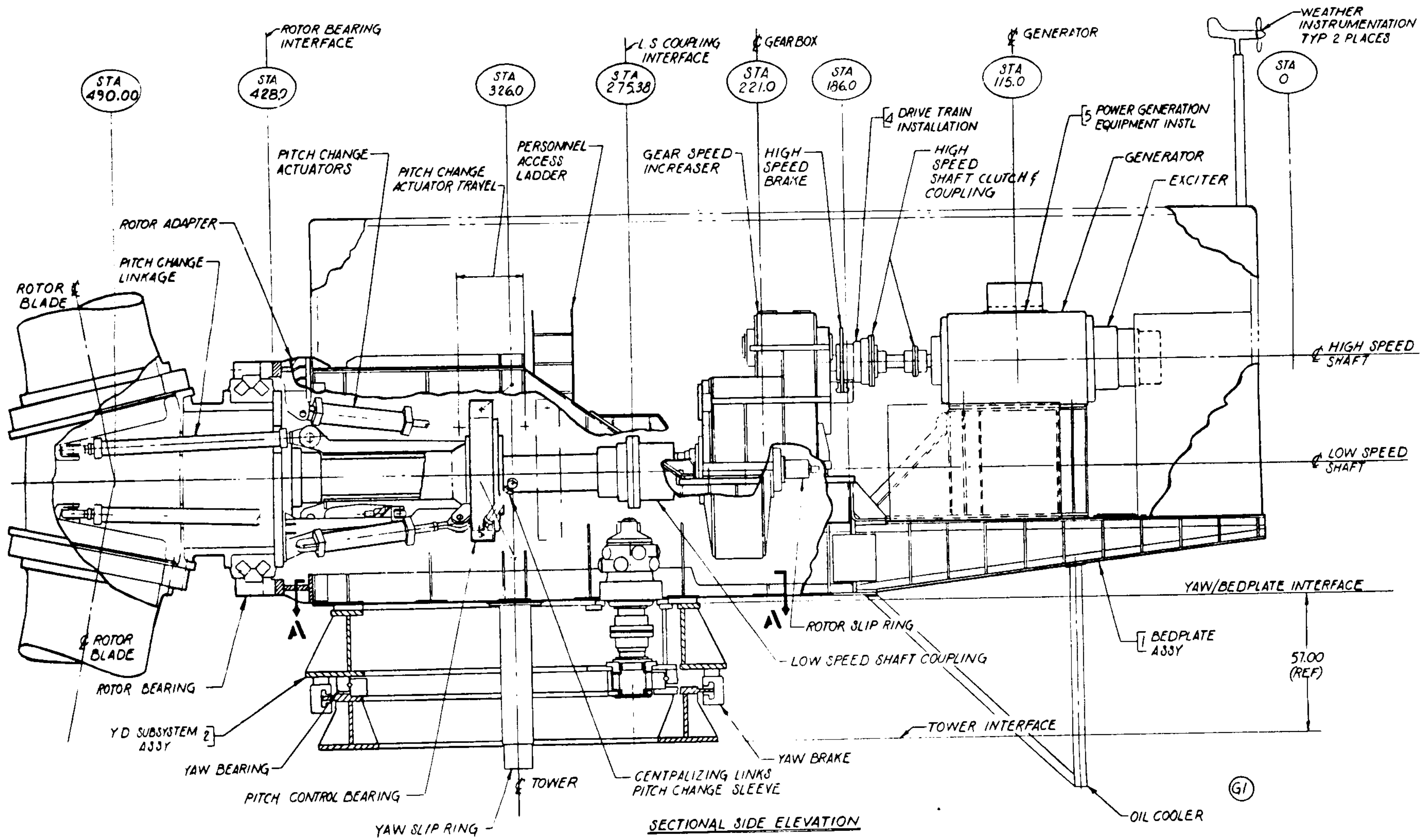

Figure 3. - Nacelle installations.

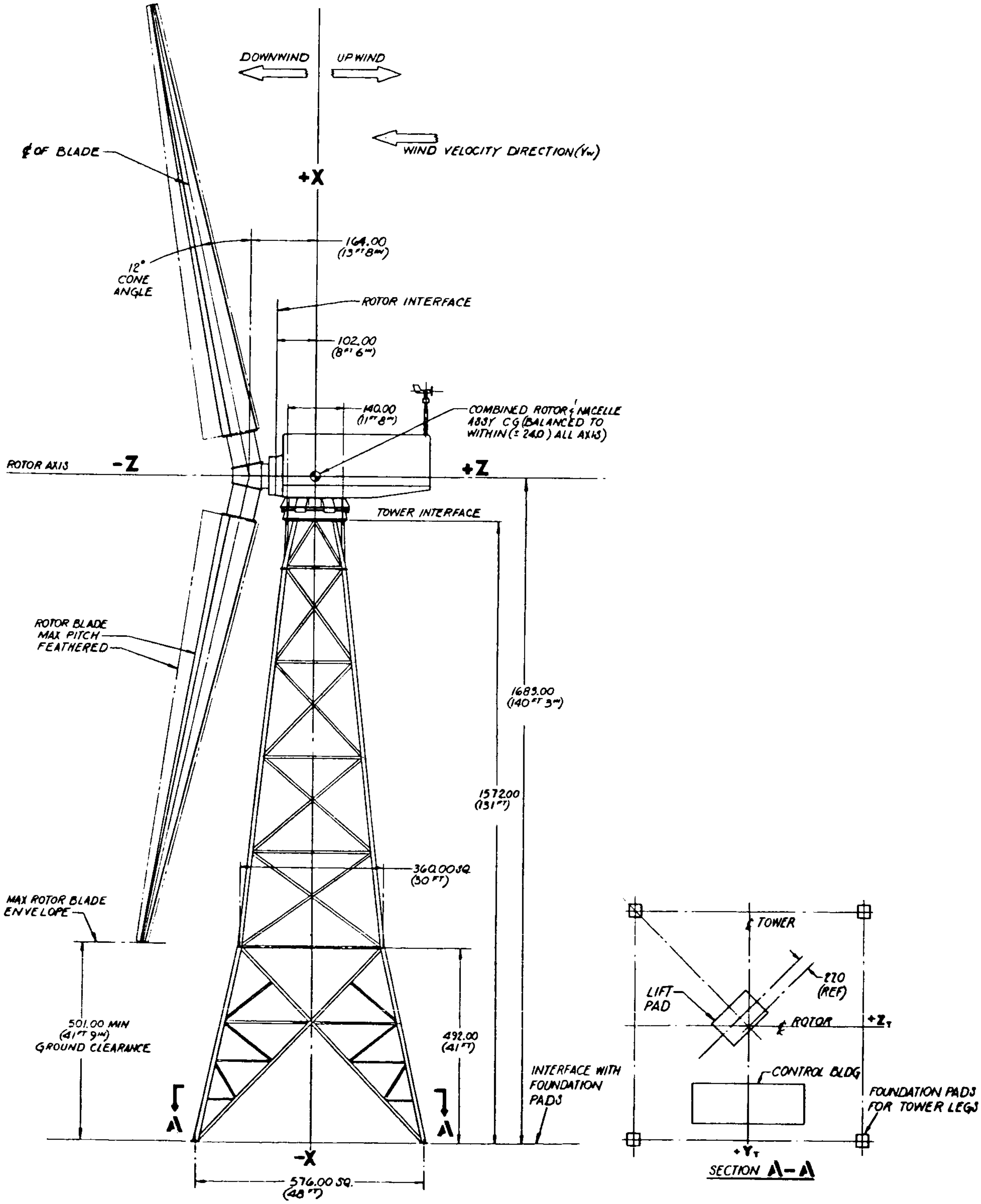

Figure 4. - Schematic drawing of MOD-1 wind turbine generator.

ROTOR ASS'Y.		103,000 LBS.
HUB	15,000 LBS.	
BLADES	36,000 LBS.	
BEARINGS/STRUCTURE	29,000 LBS.	
PITCH CONTROL MECH.	11,000 LBS.	
PITCH CONTROL HYDR.	12,000 LBS.	
NACELLE ASS'Y.		171,000 LBS.
BEDPLATE	68,000 LBS.	
FAIRING	5,000 LBS.	
GENERATOR/EXCITER	14,000 LBS.	
POWER GEN. EQUIP.	1,000 LBS.	
SHAFTS/COUPLINGS/CLUTCH	18,000 LBS.	
GEARBOX	58,000 LBS.	
LUBE/HYD. SYSTEMS	4,000 LBS.	
DATA ACQUISITION	1,000 LBS.	
CABLES/LIGHTS/ETC.	2,000 LBS.	
YAW ASS'Y		56,000 LBS.
BEARING SUPPORTS	47,000 LBS.	
YAW BRAKE	1,000 LBS.	
YAW DRIVE	8,000 LBS.	
TOWER ASS'Y		320,000 LBS.
STRUCTURE	313,000 LBS.	
ELEVATOR/MISC.	1,000 LBS.	
CABLING/CONDUIT	6,000 LBS.	
	TOTAL (EXCL. GROUND EQUIP.)	650,000 LBS.

Figure 5. - Weight breakdown for MOD-1 wind turbine generator.

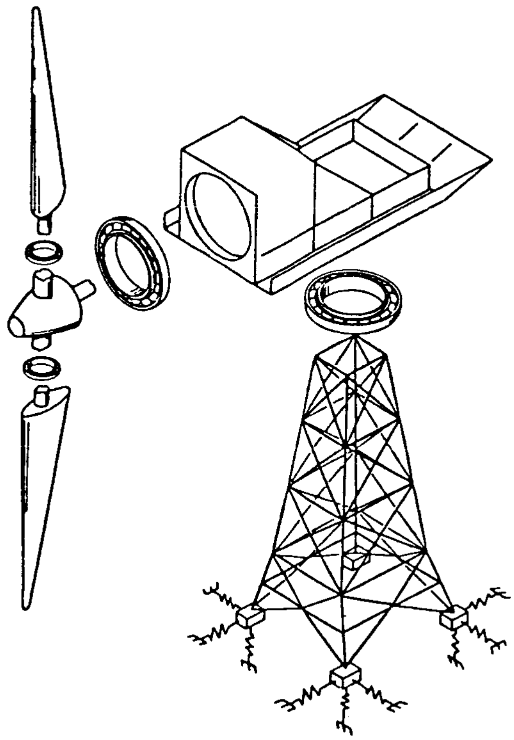

Figure 6. - Wind turbine generator synthesized from substructures.

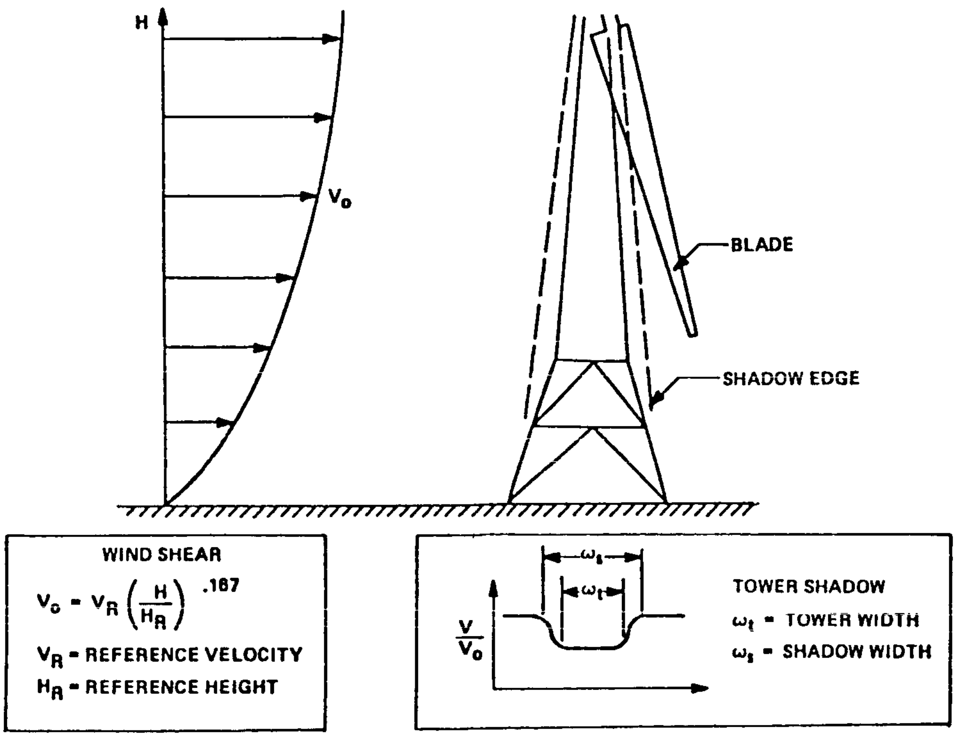

Figure 7. - Wind shear and tower shadow representation.

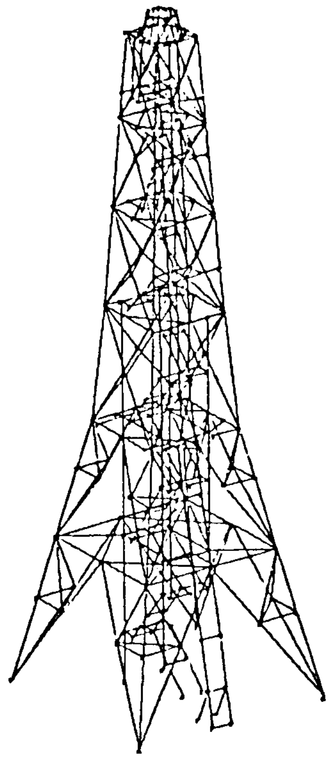

Figure 8. - NASTRAN model of MOD-0 tower.

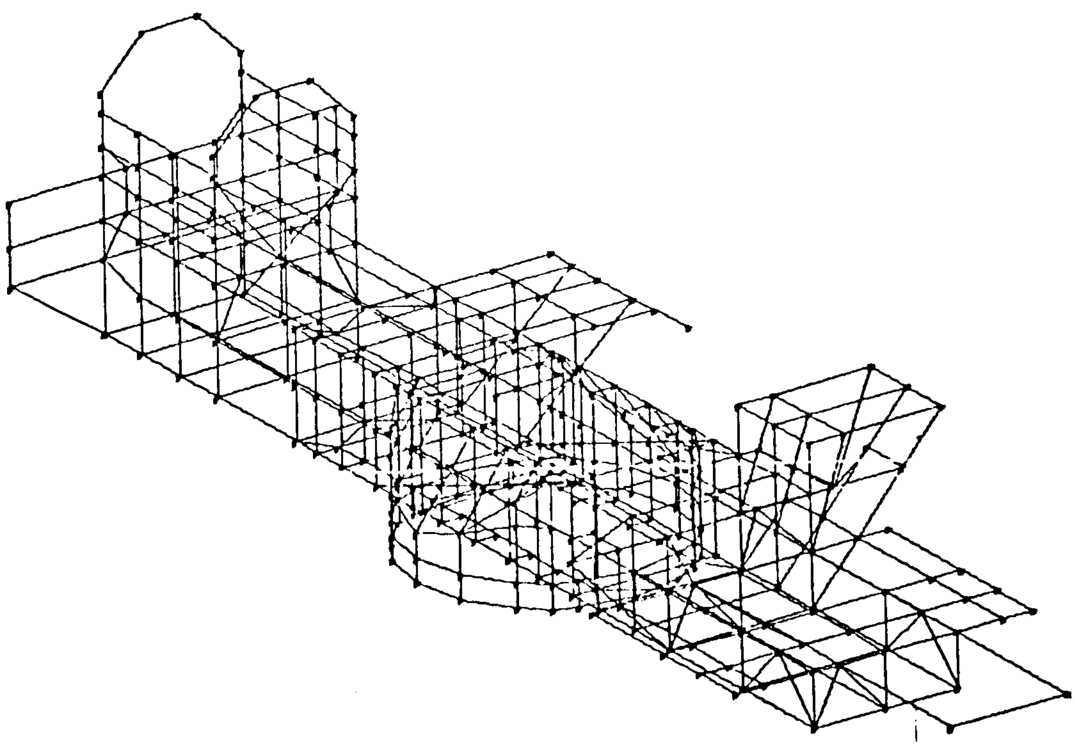

Figure 9. - NASTRAN model of MOD-0 bedplate.

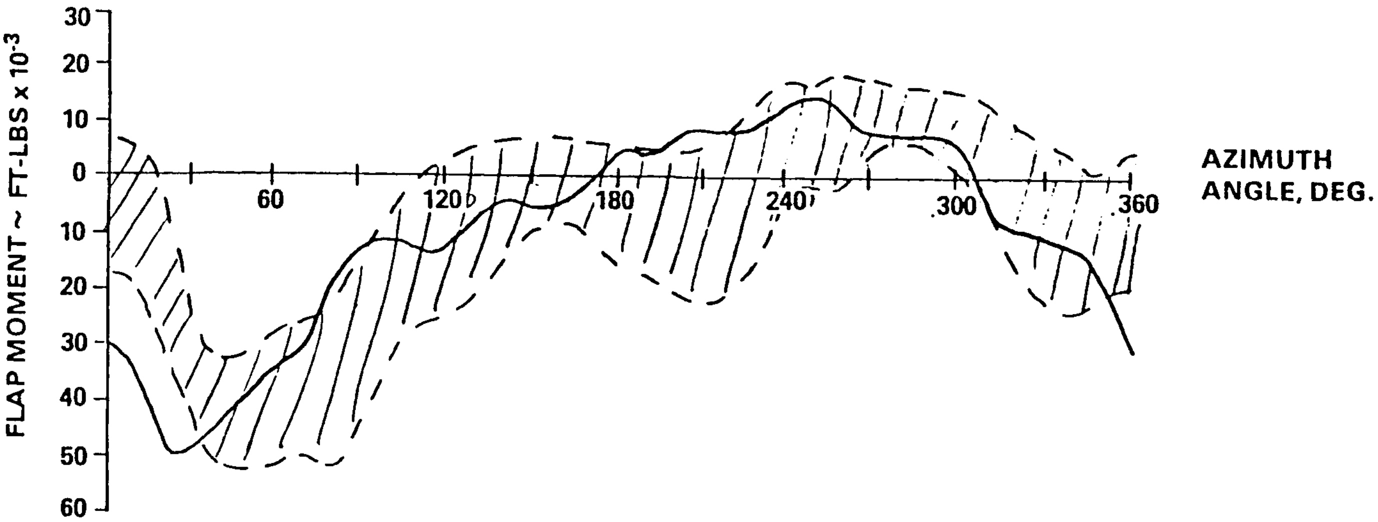

Figure 10. - Case 4 blade load comparison, flap moment at station 40.

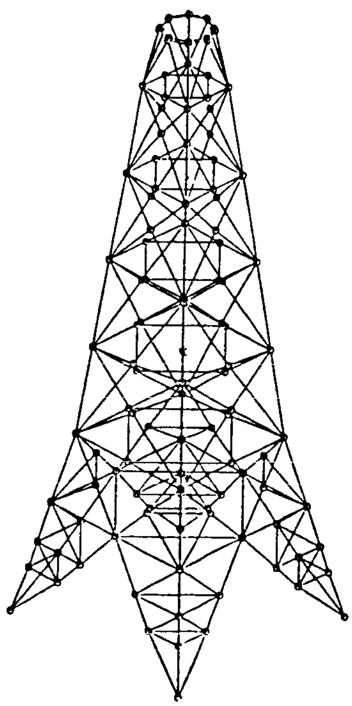

Figure 11. - Finite element model of MOD-0 tower.

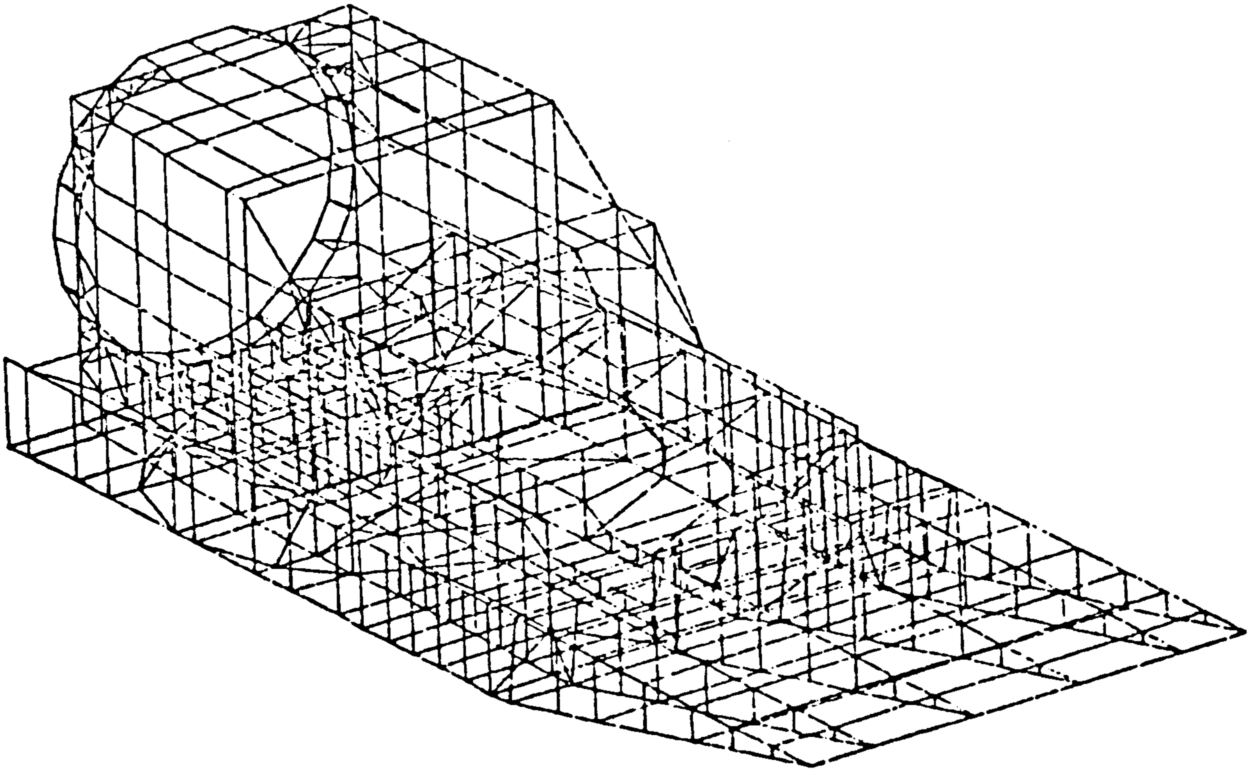

Figure 12. - Finite element model of MOD-0 bedplate.

MODE NO.	DESCRIPTION	FREQ (Hz)	FREQ (1/REV)
1	ROTOR ROTATION	0.39	0.67
2	1ST ROTOR FLATWISE-CYCLIC	1.42	2.44
3	1ST ROTOR FLATWISE-COLLECTIVE	1.52	2.60
4	TOWER BENDING-Y AXIS	1.81	3.10
5	TOWER BENDING-Z AXIS	1.91	3.28
6	1ST ROTOR EDGEWISE-CYCLIC	2.43	4.16
7	2ND ROTOR FLATWISE-CYCLIC	3.28	5.63
8	2ND ROTOR FLATWISE-COLLECTIVE	3.64	6.25
9	SHAFT TORSION	4.00	6.86
10	TOWER TORSION	4.18	7.16
11	3RD ROTOR FLATWISE-CYCLIC	6.41	10.99
12	3RD ROTOR FLATWISE-COLLECTIVE	6.62	11.35
13	BLADE TORSION-ANTISYMMETRIC	6.76	11.60
14	BLADE TORSION-SYMMETRIC	6.78	11.63
15	2ND ROTOR EDGEWISE COLLECTIVE	7.54	12.92
16	TOWER 2ND BENDING ~ Z AXIS	8.37	14.35
17	TOWER 2ND BENDING ~ Y AXIS	8.70	14.91
18	2ND ROTOR EDGEWISE - CYCLIC	9.10	15.59

Figure 13. - Summary of resonant frequency (blade vertical position).

- SOIL STIFFNESS - NOT SIGNIFICANT
- YAW DRIVE TORSIONAL STIFFNESS - CRITICAL
- TOWER/BLADE TUNING - CRITICAL
- BLADE RETENTION BEARING STIFFNESS - SIGNIFICANT
- MAIN ROTOR BEARING - SIGNIFICANT
- YAW BEARING STIFFNESS - NOT SIGNIFICANT
- SHAFT BEARINGS - NOT SIGNIFICANT
- BEDPLATE C.G. - STATICALLY SIGNIFICANT

MOD-1 CRITICAL TUNING
PARAMETERS IDENTIFIED

Figure 14. - Summary of parametric variations.

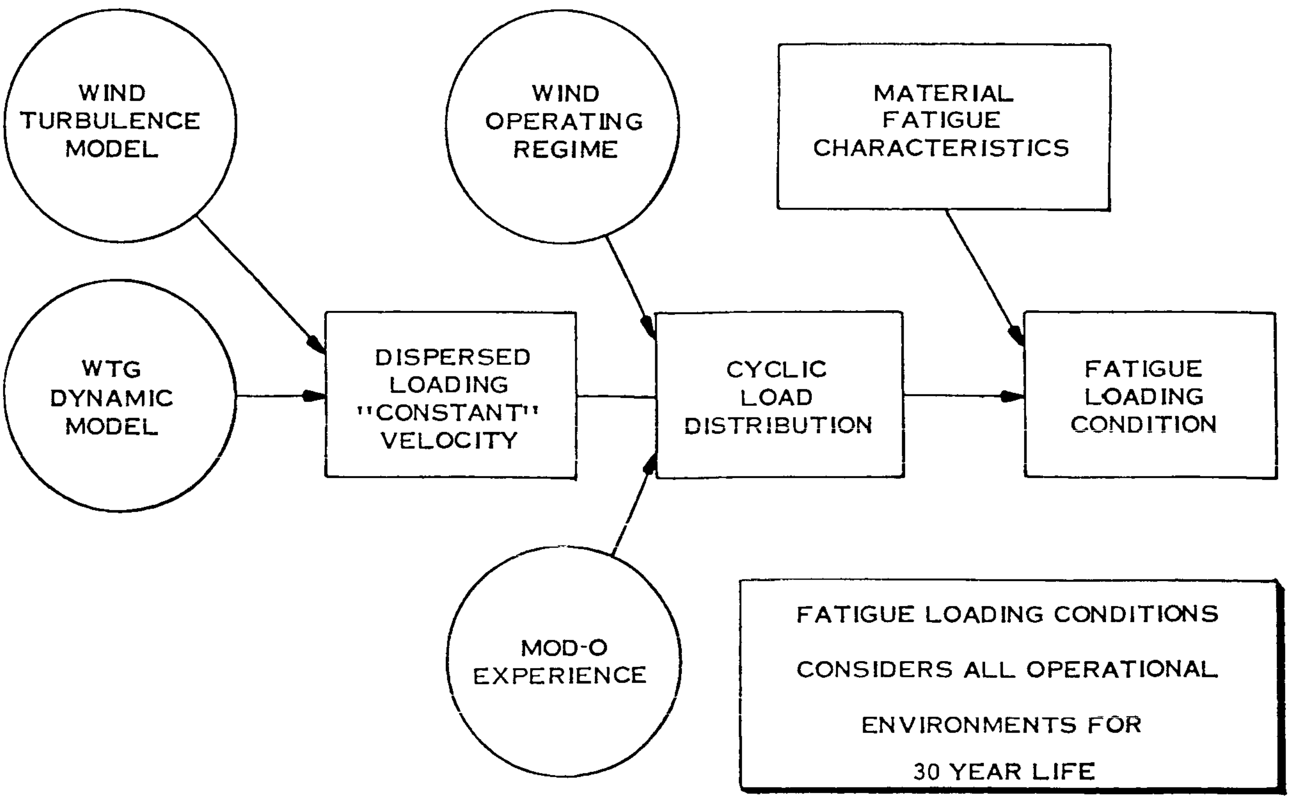

Figure 15. - Determination of design loads.