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Cleveland State University

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Wind/Ocean Technology Division**

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SUMMARY

This report covers the dynamic analysis of a horizontal axis fixed pitch wind turbine generator (WTG) that is rated at 56 kW. A mechanical Continuously Variable Transmission (CVT) was incorporated in the drive train to provide variable speed operation capability. One goal of the dynamic analysis was to determine if variable speed operation, by means of a mechanical CVT, is capable of capturing the transient power in the WTG/wind environment. Another goal was to determine the extent of power regulation possible with CVT operation.

The dynamic simulation of this drive train was performed with the Advanced Continuous Simulation Language (ACSL) computer program which is an improved version of the original IBM-CSMP program. The lumped parameter method was used to account for the inertia and stiffness effects of the drive train components in the WTG mathematical model. Also included were the effects of aerodynamic damping, generator slip and the variable speed ratio of the CVT. The method of bond graph definition of the mathematical model was used to generate the first order differential equations (state equations). Rotor torque, as determined from a WTG aerodynamic performance computer code (PROPCODE), and rotor speed were the inputs in the model. The corresponding power at the generator was the output parameter.

The WTG was subjected to three different wind gust profiles and their effect on the output power of the system was monitored. The first profile was a step increase of the wind velocity from zero to 30 mph to validate the drive train model. The second wind gust was a triangular pulse of six (6) second duration to determine the effectiveness of the CVT to regulate power output. The third wind profile consisted of a series of irregular triangular pulses to account for the random characteristics of an actual wind.

The ability of the CVT to influence power generation and regulation was investigated by using several time spans between successive CVT adjustments. These ranged from an instantaneous change to zero adjustment corresponding to a fixed speed ratio of the CVT. Currently, WTG's operate in the constant speed mode. Thus, the fixed speed ratio simulation served as a baseline for the results involved in variable speed operation. The speed of the generator was used as the signal for adjustment of the CVT ratio. For example, if generator speed was too high, then the ratio was decreased to slow the generator down. Similarly, if the speed was too low, then the ratio was increased to maximize power output from the system.

Aside from the validation of the model with a step change in wind velocity, the results of the triangular pulse indicated that the CVT in the speed change mode was capable of transient power regulation and capture. This conclusion prevailed even with the use of the random wind speed input profile. For these wind gust models, the generator power fluctuations were limited to within 2 percent of the nominal value for variable speed operation, whereas operation at fixed speed ratio resulted in a maximum power fluctuation of about 8 percent. Also, the WTG response time was considerably quicker for variable speed operation as opposed to the fixed speed mode.

Recommendations for future work would include investigations of the type of control needed for the CVT adjustment, the effect of speed changes on WTG life and actual testing of the WTG concept.

INTRODUCTION

With hydrocarbons as energy sources being depleted, alternate forms of power production are being developed and improved to meet the increasing needs of the world today. One such source of alternate energy is the wind, whose energy is captured with machines called Wind Turbine Generators (WTG's). There is a need for research to make WTG's cost effective by means of design simplification, component improvements and a better understanding of the dynamics resulting in more efficient power production. One approach considered to be an improvement to wind power machines is to include a mechanical Continuously Variable Transmission (CVT) to control the output power of the system and to increase the total amount of power produced. The use of CVT's in the automotive field is currently under investigation where size and power requirements (40 - 75 kW) permit using CVT's consisting of a system of belts and variable sheave pulleys.

This paper discusses the application of such a CVT as part of a 56 kW (75 hp) wind turbine drive train. One objective is to optimize the conflicting requirements of energy capture and system dynamics. Another objective is the potential use of low cost and simple controls resulting from the utilization of a CVT as compared to aerodynamic controls in the rotor. In the absence of any external control, the output from the generator fluctuates with wind speed. Hopefully, this investigation will show that the CVT will reduce the degree of power fluctuation from the system.

It is worth noting that the larger WTG's require a CVT which is much more complex than the simple hydraulically controlled pulley type. A split power CVT arrangement was used for a 400 kW machine (1). The split power type includes a mechanical CVT within a planetary gear train to control the speed of the planetary outer gear. This present study will determine if a small WTG system (less than 100 kW) can be controlled by a belt type CVT as compared to the more elaborate designs.

For this investigation, a wind turbine drive train including a continuously variable transmission was designed to operate at 56 kW. A dynamic simulation of this drive train was performed to determine the output power response using the CVT as the only method of control. The methods used for the drive train component design and selection are included in Appendix A. Bond graph techniques were used to model the system requiring inertia, stiffness and damping characteristics of the components. From this model, first order differential equations were generated as input for the Advanced Continuous Simulation Language (ACSL) computer program for the simulation on the Cleveland State University IBM-370 mainframe. A listing of the ACSL input is included in Appendix B.

DESIGN CONSIDERATIONS

The main components in the wind turbine drive train are the rotor, speed increaser, CVT and generator providing system input, adjustment and output. Table 1 presents the design parameters of the system.

TABLE 1

Design Parameters for 56 kW WTG Drive Train

Rotor	- Operating Range From 36 to 60 rpm
Speed Increaser	- Gear Ratio of 15.15:1
CVT	- Ratio Range From 2:1 to 3.33:1
Induction Generator	- 56 kW, 3% Slip, 1800 - 1854 rpm Range

The rotor was limited to 56 kW power production for the operating range of the WTG. A blade size was selected based on performance analyses using the NASA PROPCODE computer program. The gear ratio of the speed increaser was chosen to produce the required speed at the generator based on the rotor operating range and CVT ratio range. An induction generator was chosen to produce 56 kW at 1854 rpm based on 3 percent slip. This relation dictates that no power is produced at zero slip corresponding to a generator speed of 1800 rpm. The narrow ratio range of the CVT results in a smaller and simple design where only one pulley needs to be of a variable ratio, thus requiring only one actuator system instead of two.

The PROPCODE program is an aerodynamic performance code determining rotor power for a given airfoil design, wind speed, rotor diameter and rotor speed. Using these results the rotor speed was determined for a given wind speed to produce the maximum possible power. The performance analysis also indicates that variable speed operation compared to constant speed results in more power generation at any given wind speed. Figure 1 presents the calculated power levels generated by this WTG at various wind speeds for constant rotor speeds of 36, 40 and 48 rpm and a variable speed operation from 36 to 60 rpm.

The control system of the wind turbine limits the rotor speed to 60 rpm. However, if the control malfunctions, then the disk brake is applied and the WTG is rotated out of the wind. The purpose of the brake is to assure that the rotor speed limit is not exceeded. Using the brake to control power fluctuation is not practical because of the resulting high wear and overheating of the brake.

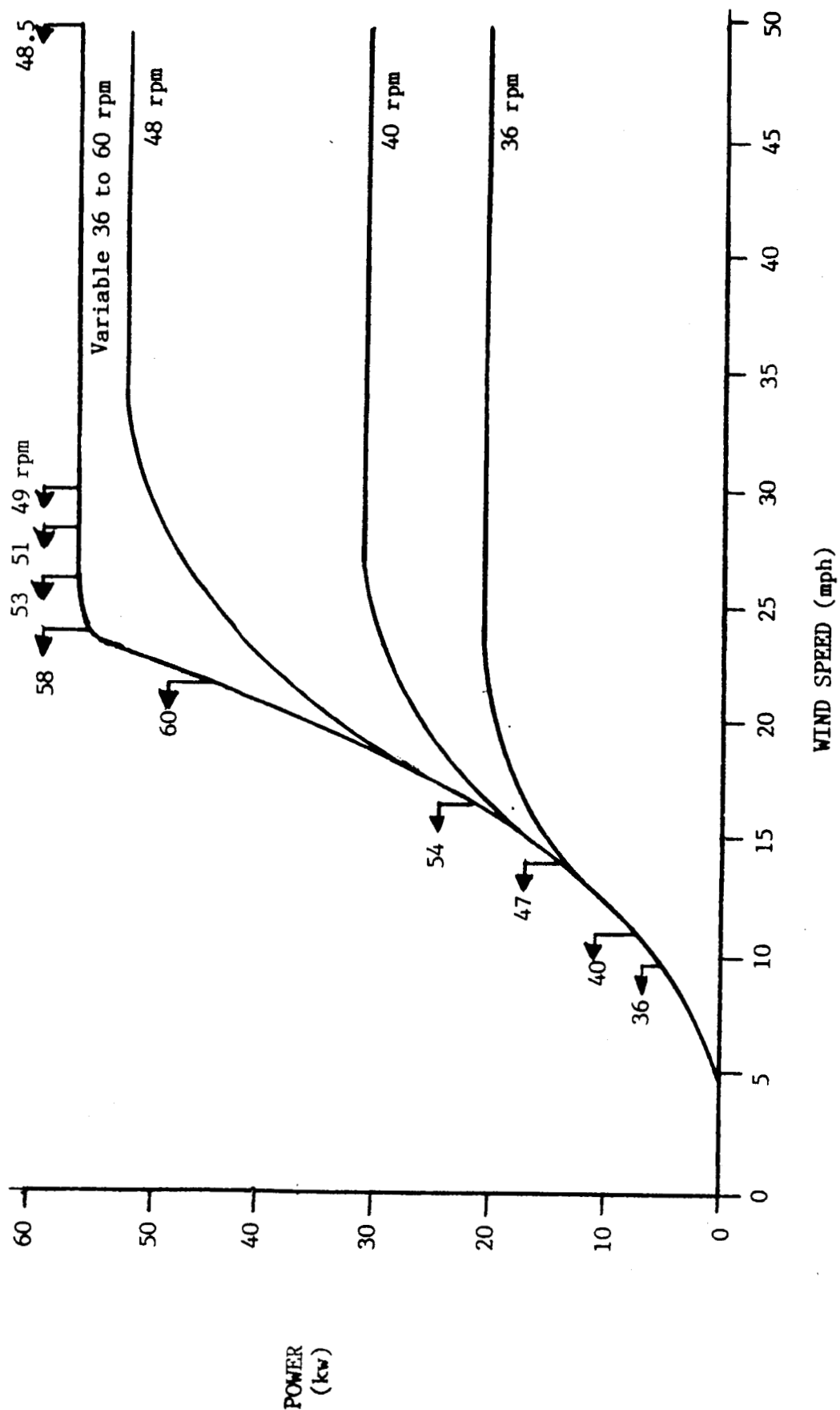
Table 2 summarizes the relation between wind, rotor and generator speed, CVT ratio and power production. Rotor speeds are chosen for a given wind speed to extract the most power from the wind. The rotor speed-power relation is from Figure 1.

TABLE 2

Drive Train Parameters During WTG Operation

Wind Speed (mph)	Rotor Speed (rpm)	CVT Ratio	Generator Speed (rpm)	Generator Slip (percent)	Power (kW)
8	36.0	3.30	1802	0.11	2
10	36.0	3.31	1805	0.27	5
12	40.0	2.98	1808	0.43	8
14	47.0	2.55	1813	0.70	13
16	54.0	2.23	1820	1.13	21
18	60.0	2.01	1828	1.55	29
20	60.0	2.02	1838	2.09	39
22	60.0	2.03	1847	2.63	49
24	58.0	2.11	1854	3.00	56
26	53.0	2.31	1854	3.00	56
28	50.6	2.42	1854	3.00	56
30	49.4	2.48	1854	3.00	56
32	48.5	2.52	1854	3.00	56
.
.
50	48.5	2.52	1854	3.00	56

Figure 1 Power vs. Wind Speed and Rotor Speed
as Predicted by the Performance Analysis



DYNAMIC MODEL OF SYSTEM

A layout of the drive train components is illustrated in Figure 2 as determined by the methods in Appendix A. The WTG system was modeled as a series of torsional springs and rotational inertias as shown in Figure 3. The effect of the speed ratios are included in this figure by multiplying each stiffness and inertia after a gear change by the corresponding speed ratio squared. The inertia (J) and stiffness (K) values of the individual components are lumped together judiciously to simplify calculations for this model as shown in Table 3. The methods used to determine the parameters of the individual components are explained in Appendix A. The model is solved in the next section using bond graph techniques to relate the inertia, stiffness and damping characteristics of the system along with the gear ratios and generator slip.

TABLE 3

Equations for Lumped Inertia and Stiffness Values

Corresponding to Figure 3

$$J1 = J1(\text{Blades} + \text{HUB} + \frac{\text{Low Speed Shaft} + \text{Brake}}{2})$$

$$K1 = K1(\text{Low Speed Shaft} + \text{Coupling 1})$$

$$J2 = J2(\frac{\text{Low Speed Shaft} + \text{Brake}}{2} + \text{Coupling 1} + \frac{\text{Speed Increaser}}{2})$$

$$K2 = K2(\text{Average Gear Mesh Stiffness})$$

$$J3 = J3(\frac{\text{Speed Increaser}}{2})$$

$$K3 = K3(\text{Coupling 2} + \text{CVT Input Shaft})$$

$$J4 = J4(\text{Coupling 2} + \text{CVT Input Pulley, Shaft} + \frac{\text{Idlers} + \text{Belt}}{2})$$

$$K4 = K4(\text{CVT Belt} + \text{CVT Output Shaft})$$

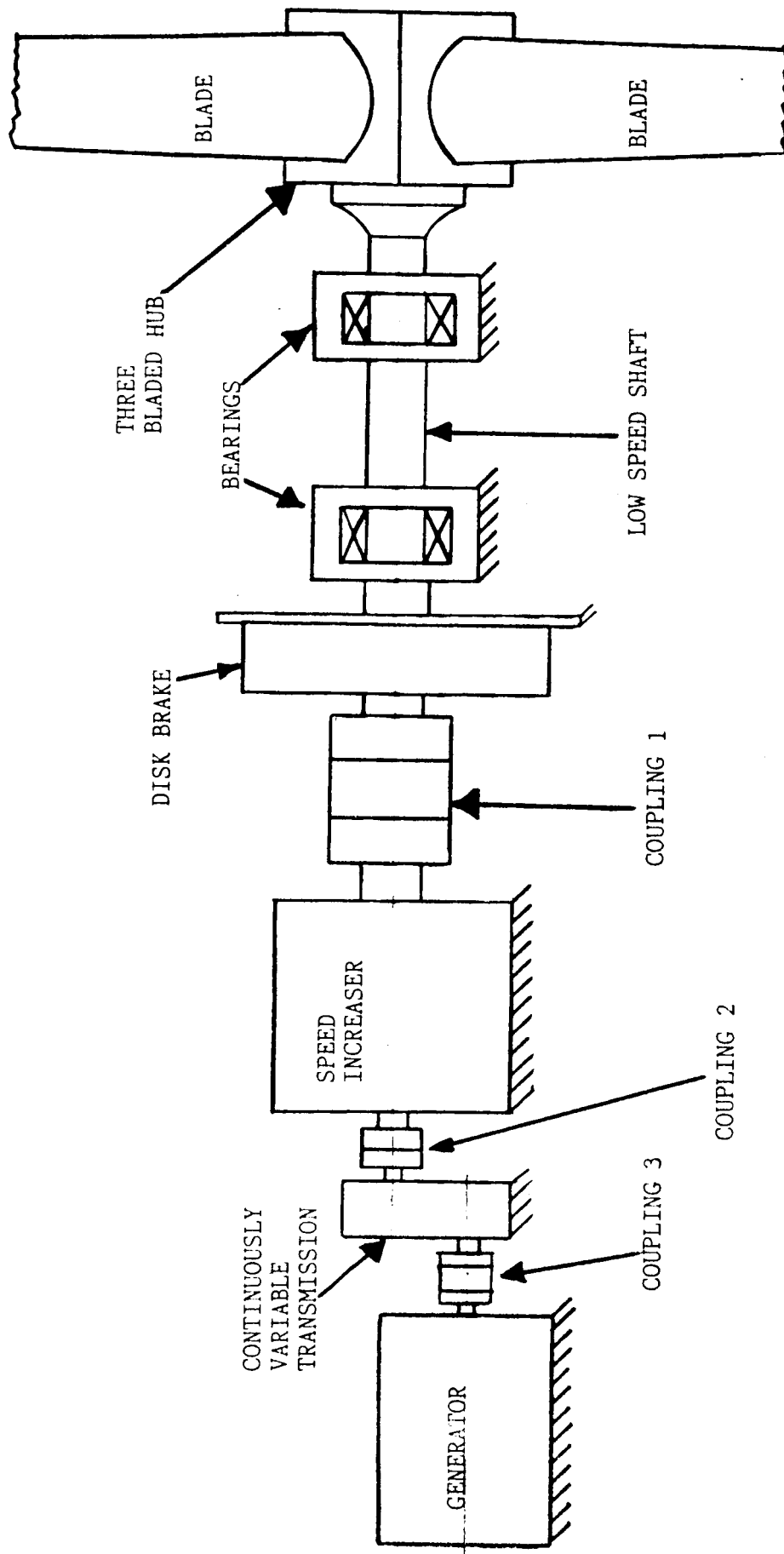
$$J5 = J5(\text{CVT Output Pulley, Shaft} + \frac{\text{Idlers} + \text{Belt}}{2})$$

$$K5 = K5(\text{Coupling 3} + \text{Generator Shaft})$$

$$J6 = J6(\text{Generator Rotor} + \text{Coupling 3})$$

$$K6 = K6(\text{Generator Field})$$

$$J7 = J7(\text{Electrical Grid})$$



Scale: 1" = 20"

Figure 2 Wind Turbine Drive Train Components

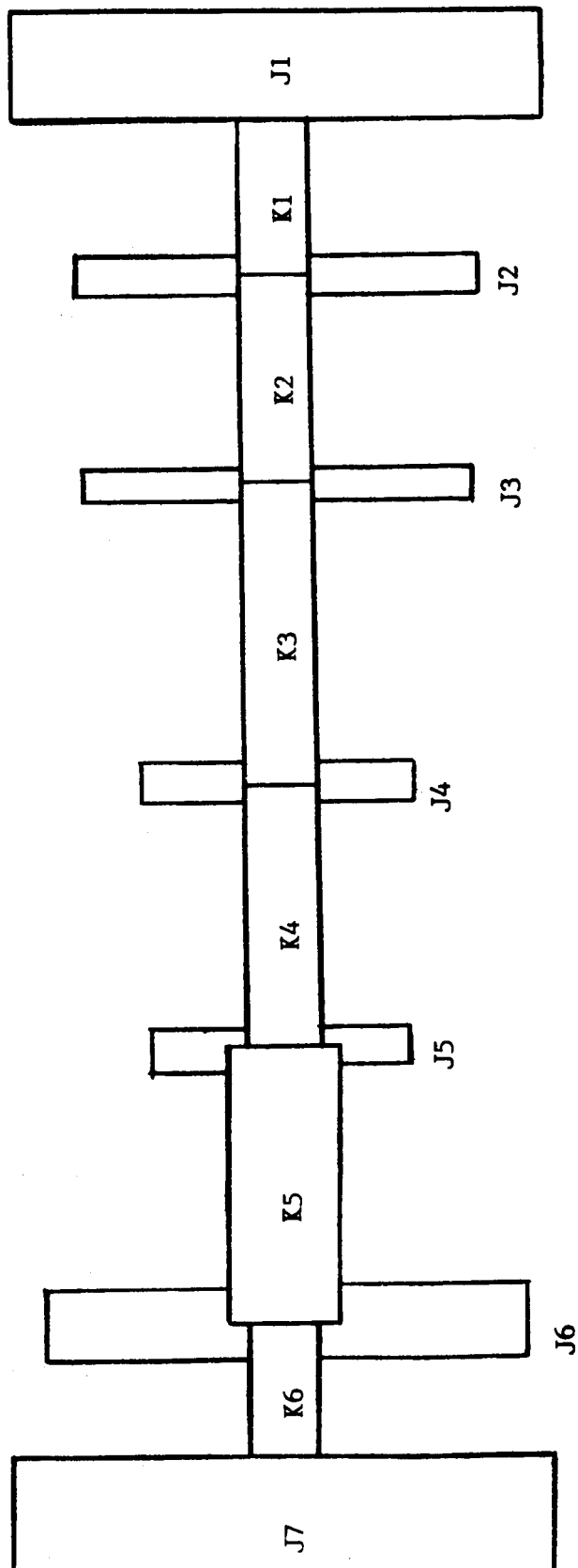


Figure 3 Inertia - Torsional Spring Schematic of WTG Drive Train

BOND GRAPH MODELING TECHNIQUES

The disk-shaft model of Figure 3 was converted to the bond graph as illustrated in Figure 4 (2,3). The inertia and damping elements are connected to constant flow "1" junctions and the stiffnesses were attached to constant effort "0" junctions. The gear ratios were represented by transformers.

The generator slip was treated as a linear relationship between torque and speed, as illustrated in Figure 5. Accordingly, the generator produces no torque at 1800 rpm (zero slip) and rated torque to produce 56 kW at 1854 rpm (full slip) (4). The effective variable damping coefficient of the generator (R_2) is defined as the ratio between the change in torque with change in speed. Equation 1 is the mathematical relationship of Figure 5 and Equation 2 is the generator damping coefficient derived from Equation 1. Ω is the generator speed in radians per second.

$$\text{Generator Torque} = -451 * \Omega + 85033 \text{ in-lb} \quad (1)$$

$$\text{Generator Damping} = \frac{85033}{\Omega} - 451 \text{ in-lb-sec} \quad (2)$$

The generator was modeled to include a relation between the mechanical shaft stiffness and rotor inertia and the electrical field stiffness and grid inertia. The generator constant (r) for this relation was obtained from Equation 2 with the generator operating at full speed to produce rated power.

The wind introduces aerodynamic damping (R_1) proportional to wind speed. The results from the PROPCODE program were used to derive the aerodynamic damping by considering the change in torque over the change in rotor speed for a constant wind speed, calculated for various wind speeds up to 50 mph. The aerodynamic damping vs. wind speed relationship is illustrated in Figure 6.

The bond graph definition of the system dictates 12 independent state variables were needed to describe the system. These first order differential, or state equations, are given in Table 4 where dP/dt is an effort variable and dQ/dt is a flow variable, following conventional bond graph definitions. The P and Q variables are momentum and displacement of each inertia and stiffness, respectively. The numbering system is consistent with the bond graph and the inertia and stiffness values from Table 3. The input torque (SE) and rotor speed (SF), as obtained from Figure 1, were applied to the first inertia (WTG rotor). Gear ratios are represented as $M1$ for the fixed ratio planetary speed increaser and $M2$ for the variable speed ratio CVT.

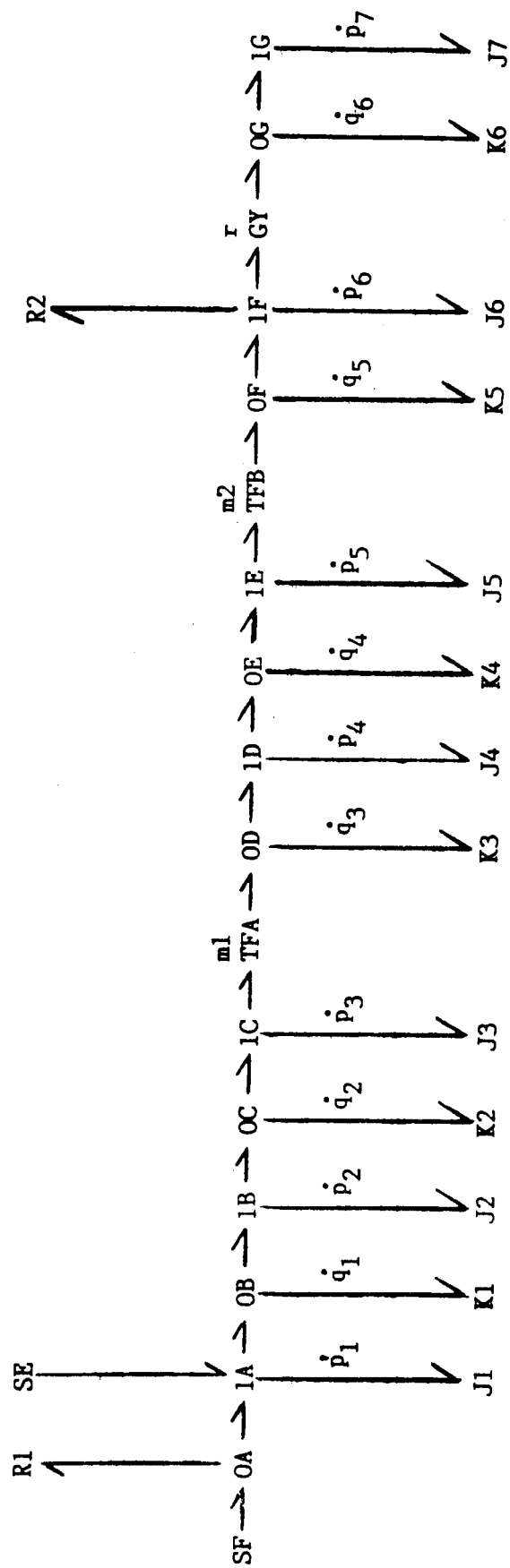


Figure 4 Bond Graph for Wind Turbine Drive Train

Figure 5 Generator Torque vs. Generator Speed for 3 Percent Slip

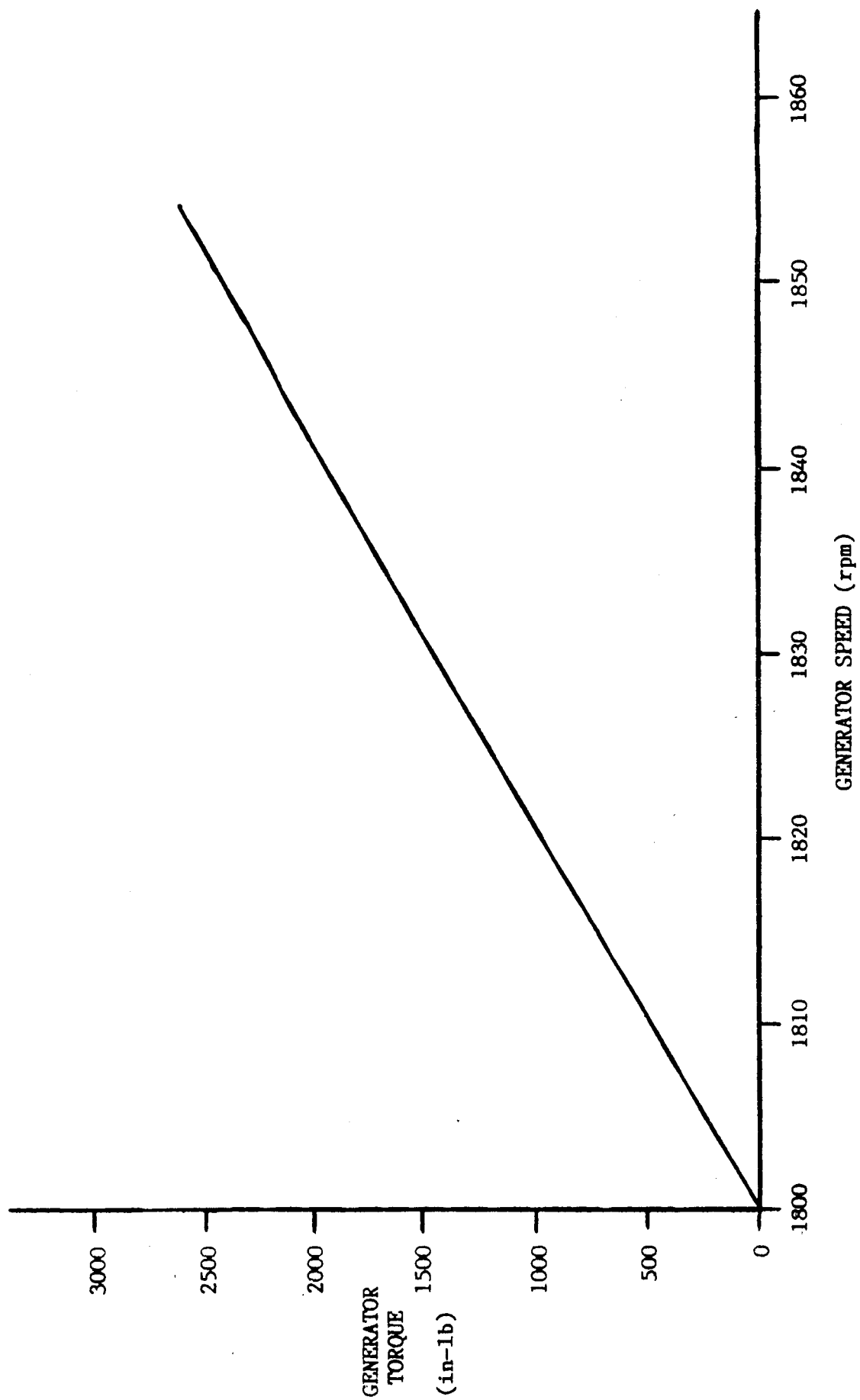


Figure 6 Aerodynamic Damping Coefficient vs. Wind Speed

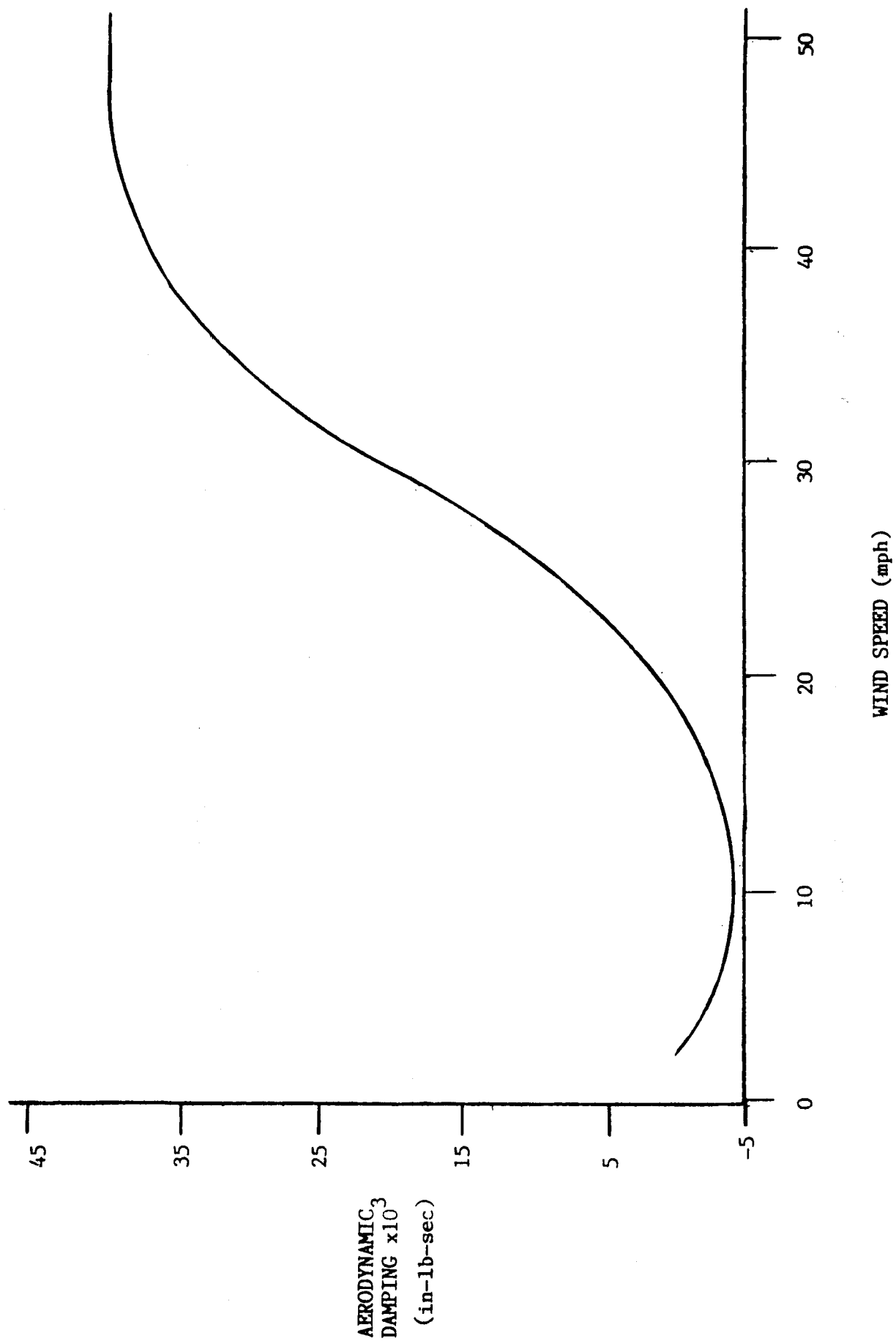


TABLE 4

State Equations for WTG Drive Train

$$\frac{dP1}{dT} = SE + R1 * (SF - \frac{P1}{J1}) - K1 * Q1$$

$$\frac{dQ1}{dT} = \frac{P1}{J1} - \frac{P2}{J2}$$

$$\frac{dP2}{dT} = K1 * Q1 - K2 * Q2$$

$$\frac{dQ2}{dT} = \frac{P2}{J2} - \frac{P3}{J3}$$

$$\frac{dP3}{dT} = K2 * Q2 - M1 * K3 * Q3$$

$$\frac{dQ3}{dT} = \frac{M1 * P3}{J3} - \frac{P4}{J4}$$

$$\frac{dP4}{dT} = K3 * Q3 - K4 * Q4$$

$$\frac{dQ4}{dT} = \frac{P4}{J4} - \frac{P5}{J5}$$

$$\frac{dP5}{dT} = K4 * Q4 - M2 * K5 * Q5$$

$$\frac{dQ5}{dT} = M2 * \frac{P5}{J5} - \frac{P6}{J6}$$

$$\frac{dP6}{dT} = \frac{J6 * K6}{J6 * K6 + r * r} * (K5 * Q5 - \frac{r * P7}{J7} - \frac{R2 * P6}{J6})$$

$$\frac{dP7}{dT} = \frac{r * P6}{J6}$$

ADVANCED CONTINUOUS SIMULATION LANGUAGE PROGRAM

The system state equations derived from the bond graph were used in the Advanced Continuous Simulation Language (ACSL) computer program for the dynamic simulation (5). The system output response over time for various input wind gust profiles was determined. The WTG is assumed to be at a nominal initial wind speed before being subjected to a wind gust.

A wind gust profile was input to the simulation. The torque and rotor speed corresponding to a particular wind speed were input to the system. The torque at the generator was obtained from the solution of the system equations. The generator speed was calculated from the slip relation presented in Figure 5 and the resulting output power was calculated. The CVT ratio was adjusted based on the system output parameters to regulate the power fluctuation from the system. Variable CVT adjustment or control times between successive CVT ratio changes were introduced to study the system output power characteristics. A block diagram of the program structure is given in Figure 7.

SYSTEM SIMULATION RESULTS

The results of the investigation indicate that the use of a continuously variable transmission of the belt-variable sheave pulley type for this 56 kW wind turbine drive train provides a practical means of controlling power output fluctuations and maximizing power production. The reasons for this conclusion are discussed in detail below.

Three different wind gust profiles were investigated. The first profile was a step increase of the wind velocity from zero to 30 mph to validate the drive train model. The second wind gust was a triangular pulse of six (6) second duration to determine the effectiveness of the CVT to control power. The third wind profile consisted of a series of uneven triangular pulses to account for the random characteristics of an actual wind. The following paragraphs present the simulation results obtained for each wind gust profile.

Power generation and control was influenced by the time duration between successive CVT speed ratio adjustments. The length of time between speed ratio adjustments was varied from very short (nearly instantaneous CVT response) to very long (constant speed WTG without a CVT). The need for adjustments is dictated by the condition of the wind. In a steady environment, the CVT speed ratio is adjusted to correspond with optimum aerodynamic performance of the rotor. In gusty winds, the degree of adjustment duration depends on the intensity and frequency of the gusts.

Initially, a wind gust step function was investigated to validate the drive train model as shown in Figure 8. The resulting mean power level should stabilize at 56 kW and remain constant for the duration of the wind function as predicted by the performance analysis. The corresponding power vs. time relation is shown in Figure 9 for variable speed operation with a CVT speed ratio adjustment time interval of 0.01 seconds. The response of the WTG due to this step input for fixed speed operation is also illustrated. A steady output of 56 kW is reached in about 6 seconds when the WTG is operating in the variable speed mode. For fixed speed operation, the WTG requires about 30 seconds to result in a constant power output condition. Thus this variable speed WTG

PROGRAM

INITIAL

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OF POOR QUALITY

Set All Constants

Calculate Initial Conditions and
Drive Train Parameters for an
Initial Operating Wind Speed

END INITIAL

DYNAMIC

Input Wind Function

Calculate Model Input Torque and
Rotor Speed Due to this Wind

DERIVATIVE

Solve State Equations

END DERIVATIVE

Calculate Generator Output Parameters

Adjust CVT Ratio

STOP

END DYNAMIC

END PROGRAM

Figure 7 ACSL Program Structure

Figure 8 Step Function Wind Profile

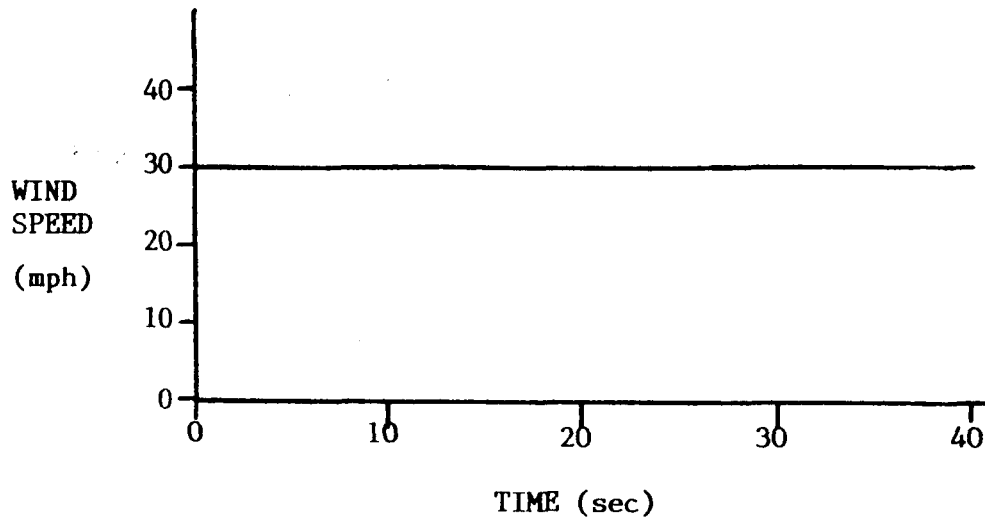
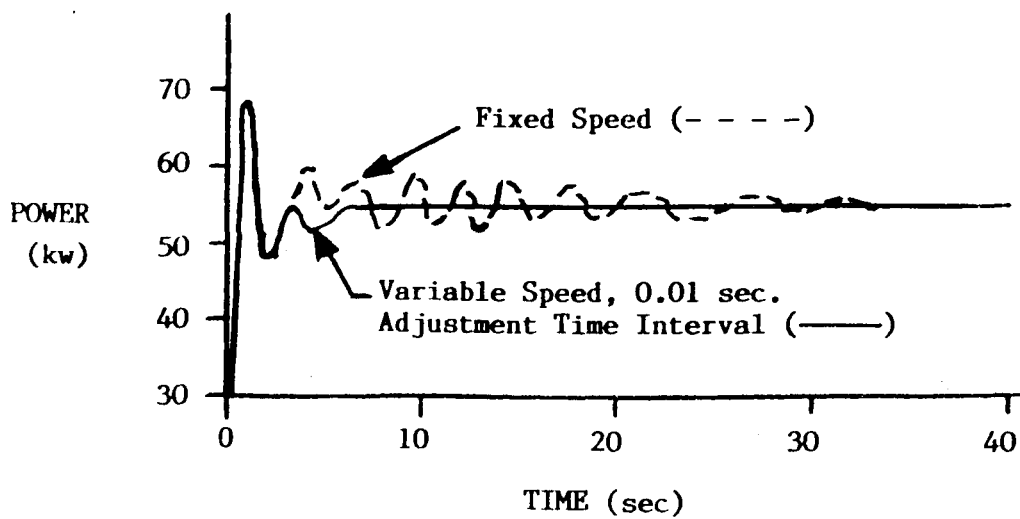


Figure 9 Power Output for Step Wind Increase From Zero to 30 mph Comparing Variable Speed Operation with a 0.01 Second Adjustment Time Interval to Fixed Speed



responds about 5 times quicker when compared to fixed speed operation for this wind profile. Both the variable and fixed speed results converge to 56 kW and thus the simulation model is correct. The fluctuation of power at the beginning of the gust is due to transients in the system which are dissipated after a few seconds.

The next profile investigated was a triangular wind gust preceded and followed by constant wind speed characteristics, as shown in Figure 10. The constant portions of the wind profile are below the rated wind speed of 24 mph and output is correspondingly below rated power which is expected. Figure 11 illustrates that a small time interval between CVT speed ratio changes results in minimal power fluctuation, desirable from a dynamic standpoint. A fixed CVT ratio results in an insignificant increase in power which fluctuates for the duration of the wind profile imposing dynamic wear on the system. The speed increaser is subjected to alternating loads resulting from the power fluctuation. The CVT must mechanically adjust for this change and the generator may overheat if power production frequently exceeds the rated value. A system with the CVT inactive produced more power but this was not constant and the objective was to optimize power production and reduce fluctuation. The results indicate a noticeable improvement in power output with the CVT controlling power as compared to a fixed speed WTG.

A more complex wind gust was input into the simulation by means of a series of uneven triangular pulses, as shown in Figure 12. The pattern for this wind gust profile was obtained from (6). The corresponding power vs. time relation is shown in Figure 13. The CVT must be adjusted frequently for minimal power fluctuation while the WTG is subjected to drops in wind speed below 24 mph. At these low wind speeds, the WTG is not able to produce rated power of 56 kW, causing power to fluctuate.

Another noticeable fluctuation occurs when wind decreases to 15 mph around the cut-in wind speed of the WTG corresponding to the speed where power is first produced. The system tends to become unstable when the CVT is adjusted at near instantaneous increments at this wind speed, as shown in Figure 13 for a 0.01 second adjustment. It is therefore necessary to operate at fixed speed around cut-in speed until wind speed increases to eliminate possible dynamic fluctuations in the system. The results indicate that variable speed operation will result in maximum power generation with minimal fluctuation for the duration of this complex wind gust profile.

As wind speed diminishes, power output decreases as was illustrated in Figure 1 representing power production from the aerodynamic analysis on the WTG rotor without any dynamic effects. Figure 14 shows a comparison of power predicted by the performance analysis and that obtained from the simulation for the wind gust profile of Figure 12. The simulation includes system dynamics resulting from energy storage and release within the drive train system. This effect is most prominent from the high inertia of the rotor resulting in energy being stored and then released through the system when operating below rated wind speed. This energy provides more power output at low wind speeds. This explains why, from Figure 14, the resulting power was higher than predicted by the performance analysis when wind speeds were reduced to 15 and 20 mph for short periods of time. Only a relatively small amount of energy is stored in the drive train components and there is a reduction in power generation at low wind speeds. Thus, decreases in output are expected when wind speed is not sufficient to produce rated power for extended periods of time.

Figure 10 Triangular Wind Pulse of Six Second Duration

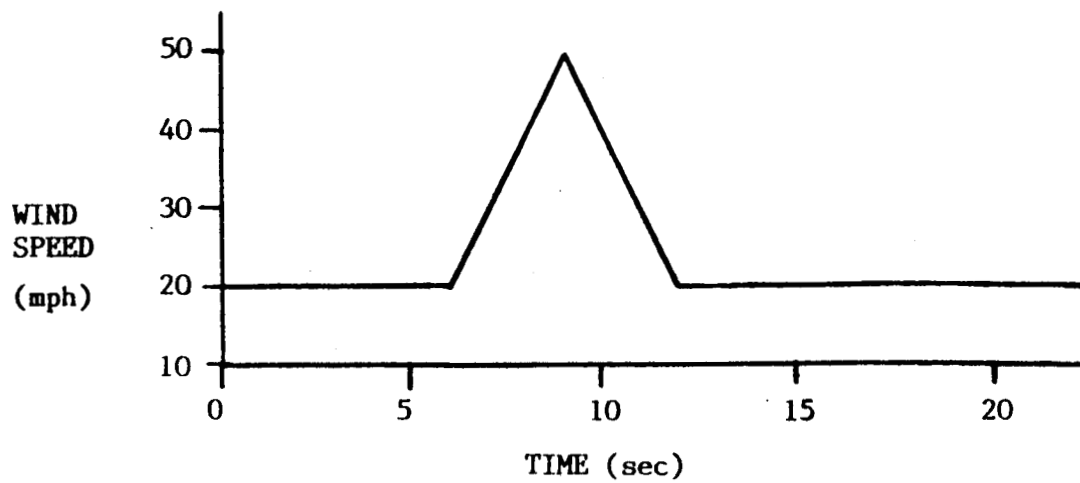


Figure 11 Comparison Between Power Response of a Fixed Speed Ratio CVT and Variable Speed with an Adjustment Time Interval of 0.01 Seconds for a Triangular Wind Pulse

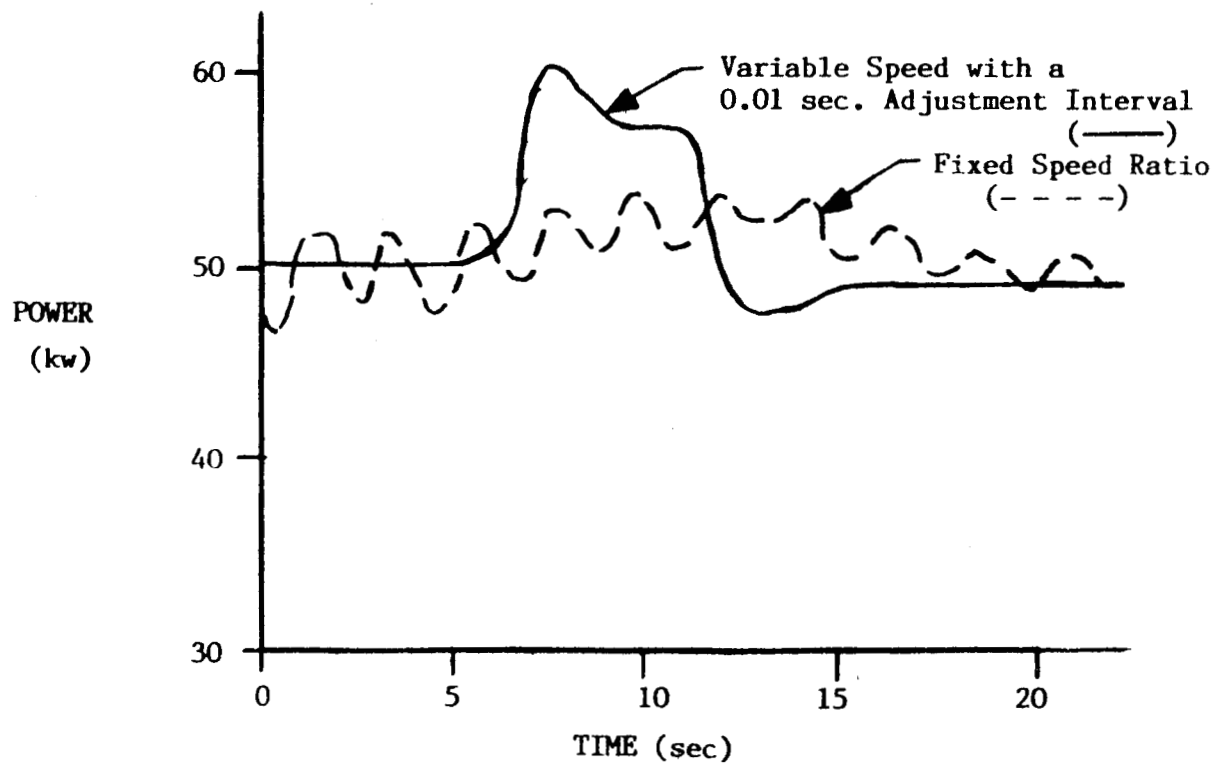


Figure 12 Complex Wind Gust Profile Simulating an Actual Wind

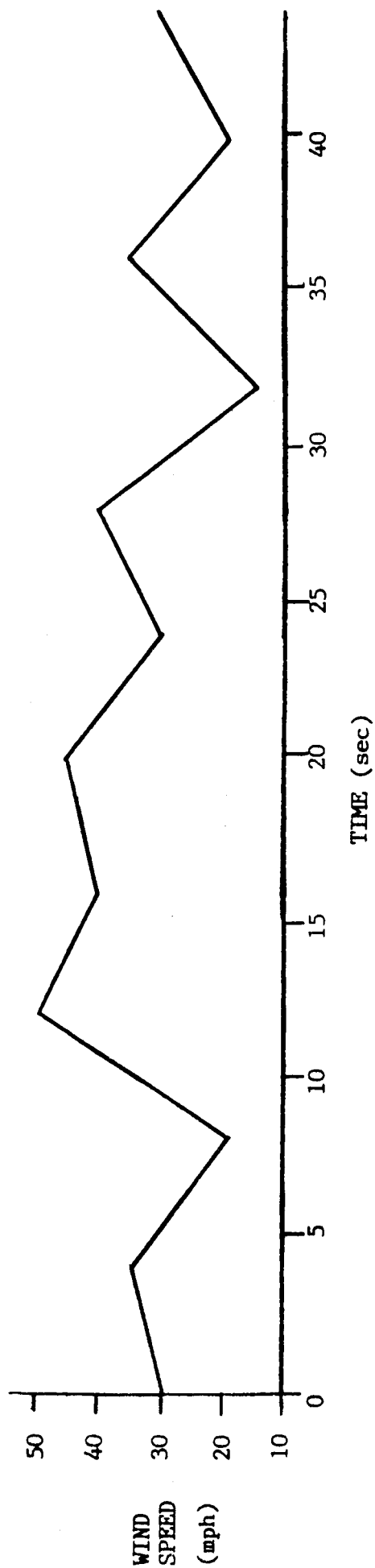


Figure 13 Power Response of Complex Wind Gust Profile Comparing Fixed Speed Operation to Variable Speed with a 0.01 sec. Adjustment Time Interval

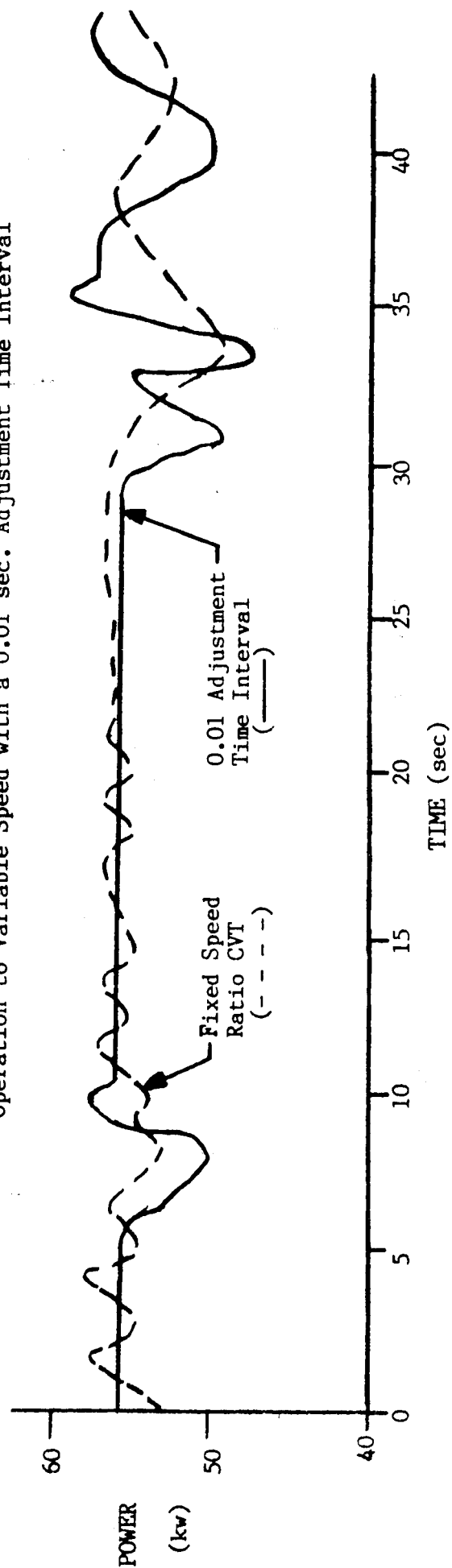
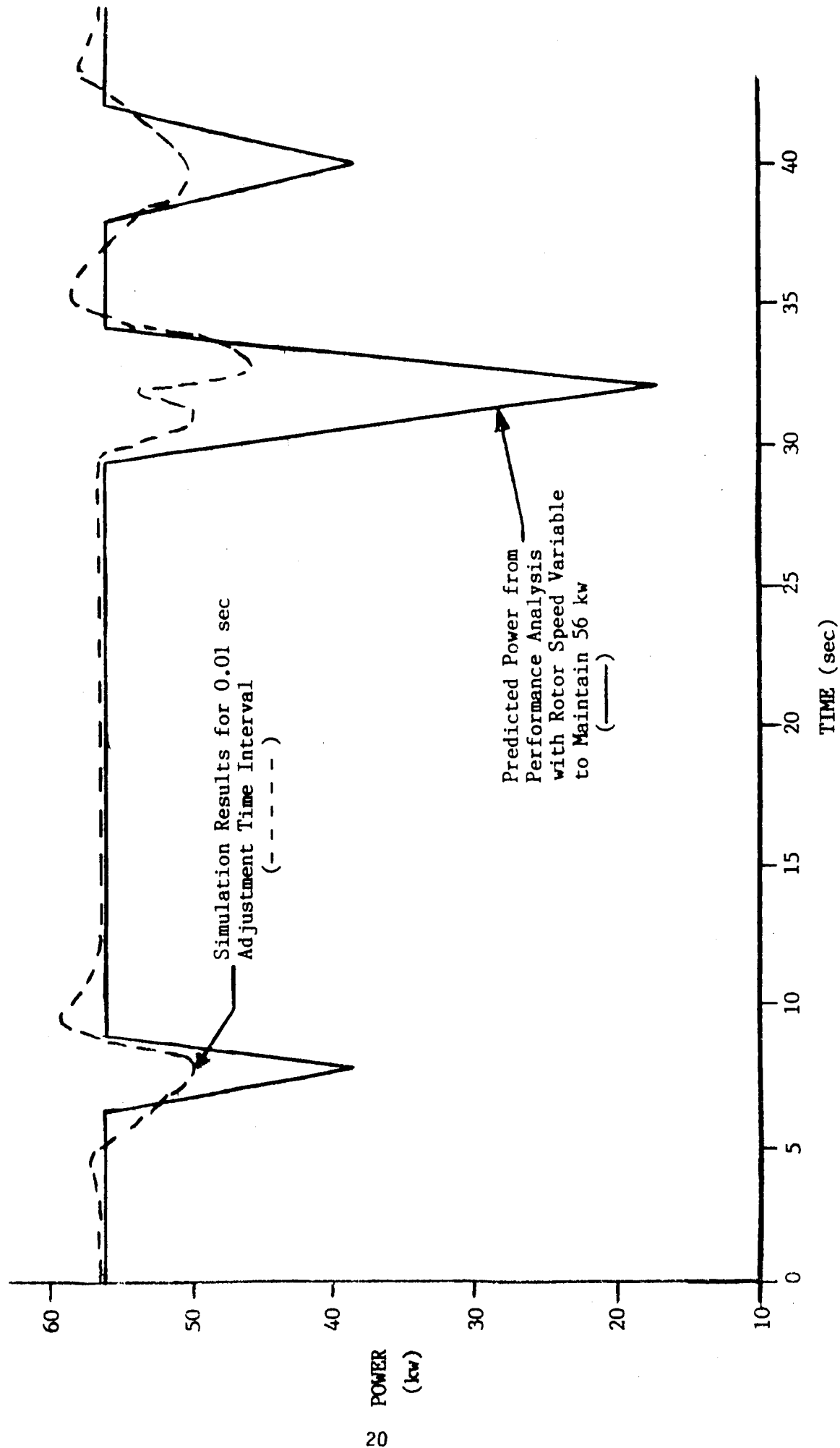


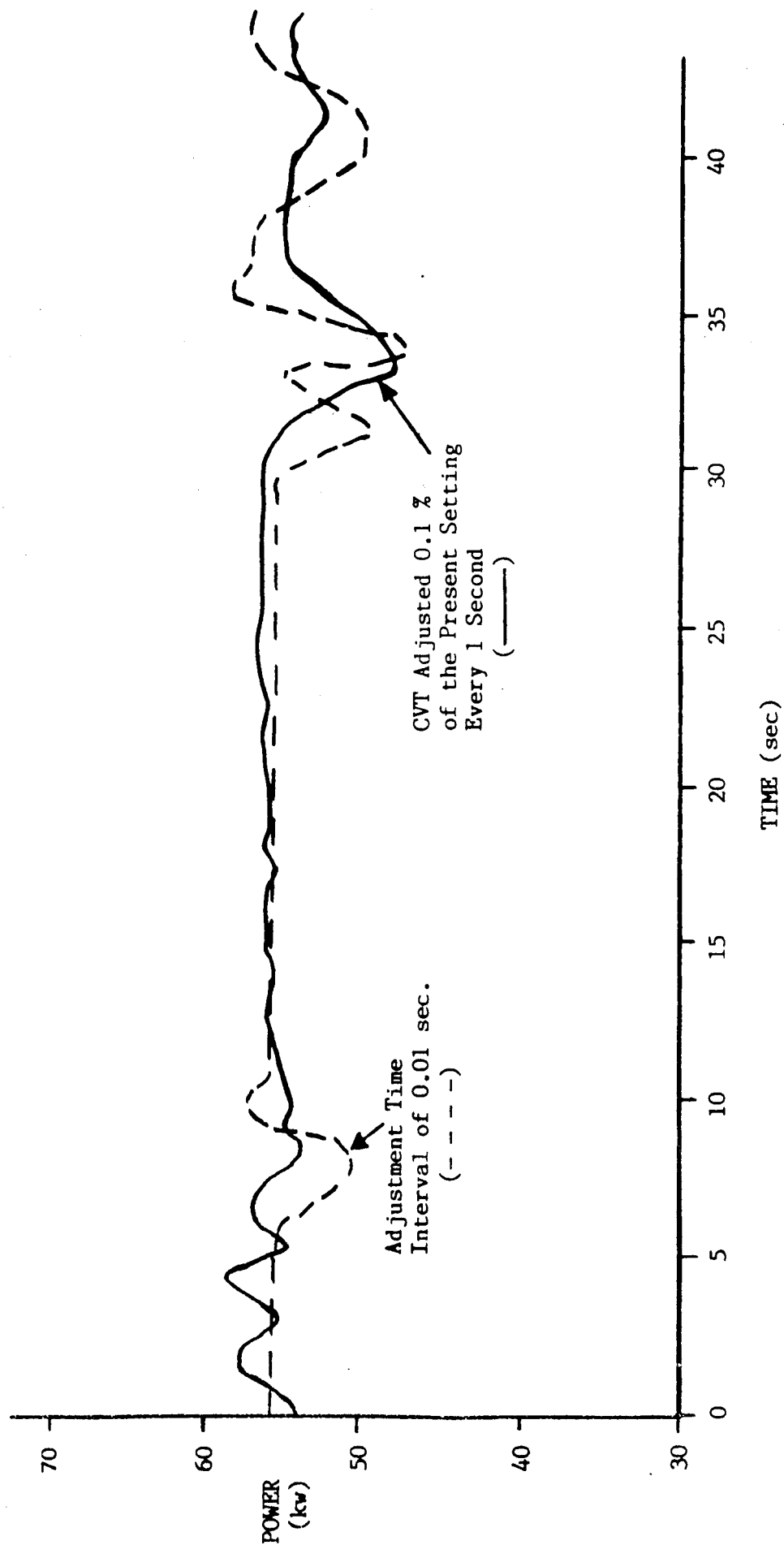
Figure 14 Comparison Between Power Predicted by the Performance Analysis
and the Simulation Results for the Complex Wind Gust Profile



Another method of control was investigated which involved reducing the CVT ratio by a constant percentage of the present setting if power is too high or increasing the ratio if power is low. Percentage changes ranging from 0.01 to 10 percent were investigated along with time spans from 0.01 to 1 second between CVT adjustments. The best power output obtained is shown in Figure 15 for the wind gust profile of Figure 12 where the CVT ratio was adjusted by 0.1 percent at 1 second intervals. This method of control is slow in reducing power fluctuation, as seen from Figure 15. For reference, the 0.01 second adjustment time interval curve of Figure 13 is repeated on this figure. Results obtained from this method are similar to fixed ratio operation of the CVT. Therefore, this type of CVT control is effective if wind speed is not changing very much since optimum power production will not result.

In summary, for actual wind profiles with changing wind speeds, power output is optimized with frequent adjustments of the CVT at a variable rate to minimize fluctuation. The belt-variable pulley CVT has the capability of improving power production from this 56 kW WIG when compared to a fixed speed ratio design.

Figure 15 Power Response for the Complex Wind Gust Profile with the CVT Adjusted 0.1 Percent of the Present Speed Ratio at 1 Second Time Intervals



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12. "Drive Train Normal Modes Analysis for the ERDA/NASA 100 Kilowatt Wind Turbine Generator", T. L. Sullivan, D. R. Miller and D. A. Spera, NASA TM-73718, July, 1977.

APPENDIX A - DESIGN METHODS FOR DRIVE TRAIN COMPONENTS

A number of appropriate components had to be designed for establishing the system parameters (inertia, stiffness and fatigue life). Subsequent sections deal with the design or selection of the rotor, low speed shaft, bearings, speed increaser, CVI, generator and couplings.

ROTOR SELECTION

A wind turbine rotor was chosen to produce rated power of 56 kW while operating within a speed range compared to that of WTG's currently in production. The limiting factors used to size a blade were a tip speed ratio of 8 and a constant of 275 which is the product of rotor speed and blade length. Previous horizontal axis wind turbine research has determined that optimum energy capture is achieved when these values are used (7). Equation 3 relates these factors together and the rated wind speed of 24 mph (34.4 fps) is determined which is the minimum wind speed needed to produce rated power.

$$\text{Tip Speed Ratio} = \frac{\text{Rotor Radius} * \text{Rotor Speed}}{\text{Wind Velocity}} \quad (3)$$

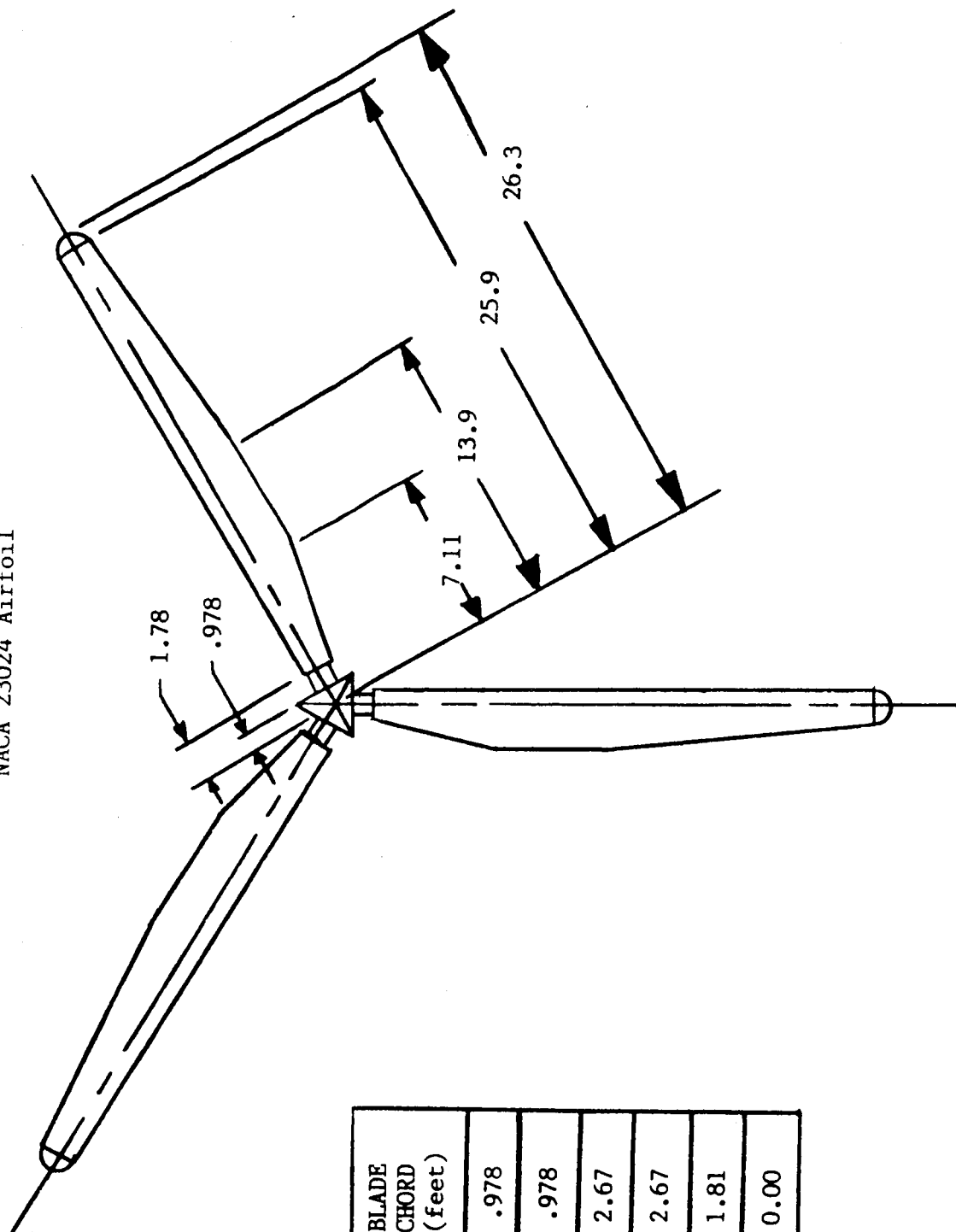
These parameters aid in developing the input to the PROPCODE performance code necessary for selection of a blade size. PROPCODE requires, as input, blade data such as airfoil lift and drag coefficients, chord length and blade radius corresponding to each chord length. This program was run at different blade radii to determine a blade that produces rated power at a wind speed of 24 mph. Figure 16 shows the rotor length and chord values at specified radial stations obtained from this analysis.

A 26 foot three bladed rotor constructed of a NACA 23024 airfoil design was used at fixed pitch. The WTG was operated in the upwind position with a zero degree coning angle since this results in the lowest dynamic loading for the upwind configuration (8).

The operating speed range of the system was determined using results obtained from PROPCODE. For a given wind speed, rotor speed was determined to extract the maximum possible power. As wind velocity increases, rotor speed is increased until rated power is reached and then the rotor is slowed down to "level off" power production and limit it to 56 kW for any further increase in wind speed. The resulting operating range of the WTG is from 36 to 60 rpm. The selected rotor speed for each wind speed and the resulting power production is presented in Figure 1.

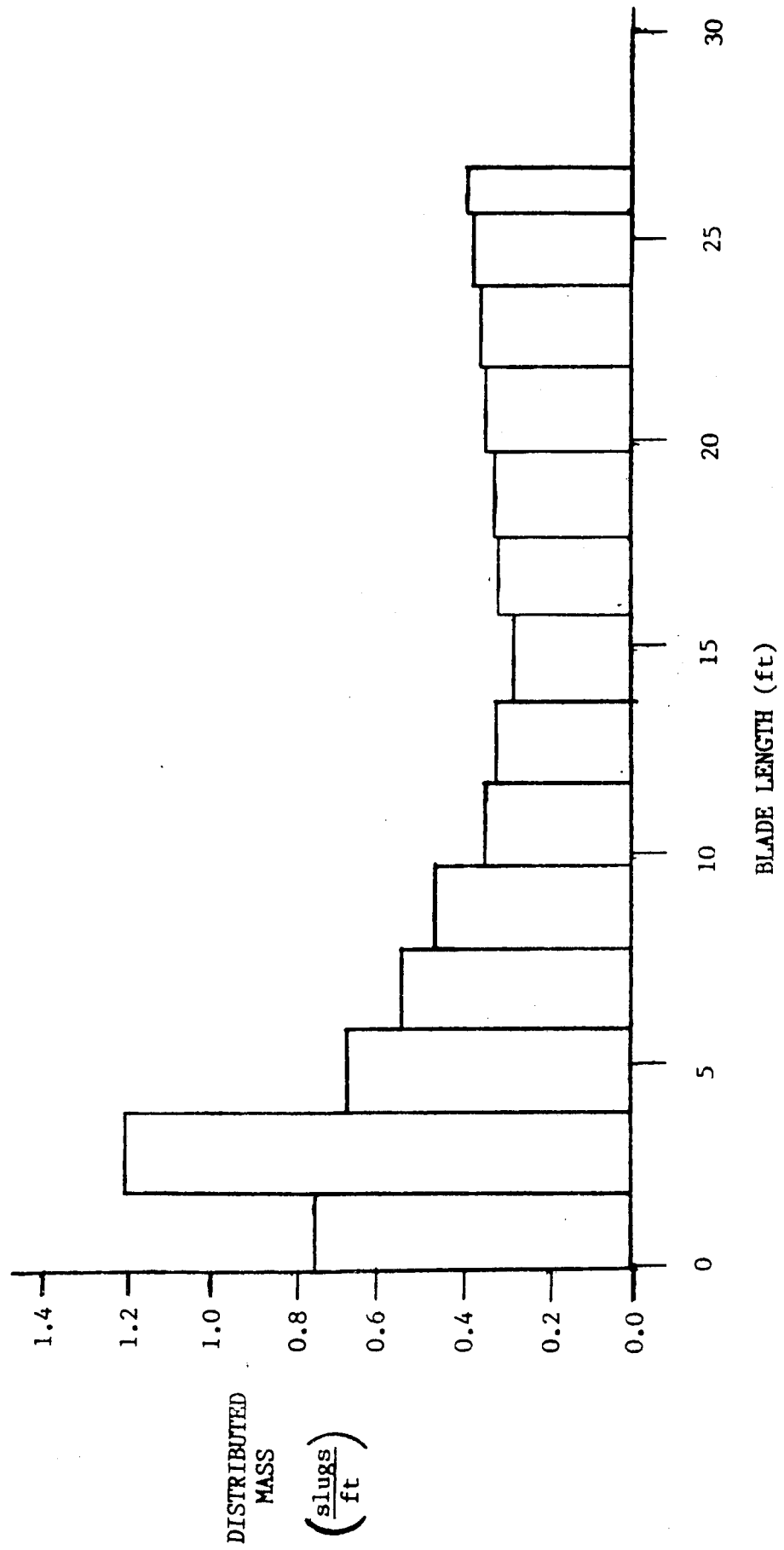
Data was available for a 50 foot blade consisting of a bar graph illustrating lumped mass distribution at specified blade locations. An equivalent graph for a 26 foot blade was obtained by reducing the 50 foot blade graph by appropriate scale factors. The radial distances were reduced by a linear ratio of the blade radii while the distributed mass values were reduced by the same ratio squared resulting from a conservative volumetric reduction. A direct determination of the inertia and stiffness values for the blade would be very involved since the type of materials and method of construction of the blade must be considered. A graph representing the lumped mass distribution vs. radial distance for the 26 foot blade is shown in Figure 17. A simple computer

Figure 16 Three Bladed Rotor
NACA 23024 Airfoil



BLADE STATION (feet)	BLADE CHORD (feet)
0.978	0.978
1.78	0.978
7.11	2.67
13.9	2.67
25.9	1.81
26.3	0.00

Figure 17 Blade Distributed Mass vs. Length



program was used to determine the weight and inertia values at each radial station based on the area and distance to each bar on the graph. A resulting single blade weight of 424 lbs. and an inertia of 28,300 in-lb-sec² was determined.

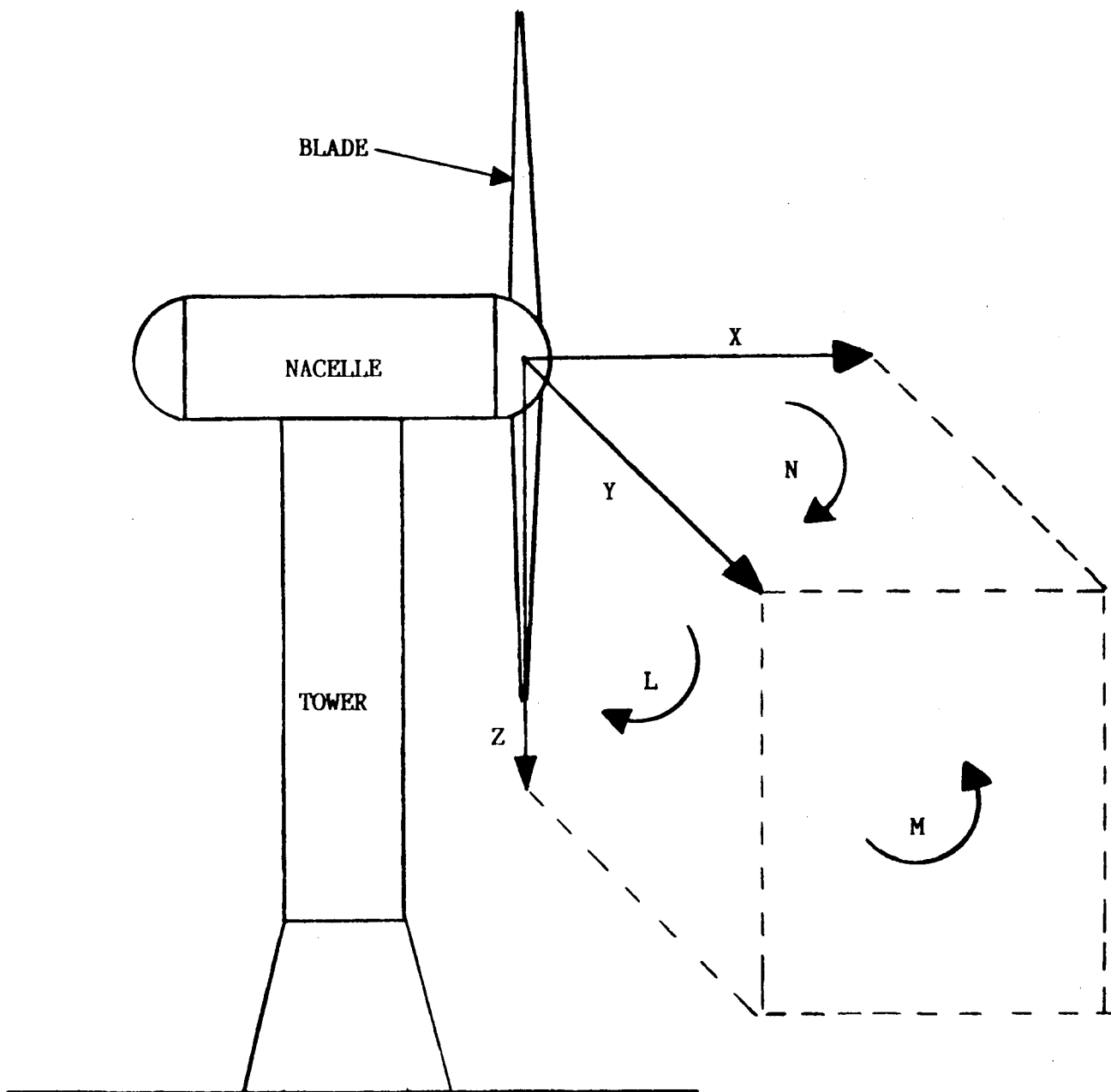
The basic shape of the MOD-0 three-bladed hub was used with dimensions reduced to accommodate for the smaller design. The hub weight was calculated along with the inertia using a method done previously for the MOD-0 hub. A detailed presentation of this method along with a breakdown of the hub sections and the inertia and weight equations for each piece is given in (9). The hub contributes 2500 lbs to the rotor weight and 2300 in-lb-sec² to the inertia.

ROTOR LOADING ANALYSIS

A dynamic loading analysis using the MOSTAB computer program was performed on the wind turbine rotor so that a low speed shaft, connecting the rotor hub to the first coupling, could be designed. The loads acting on the WTG hub, which are related to wind and rotor speed, were applied to the low speed shaft to obtain the size necessary for infinite life. The MOSTAB analysis determines these loading characteristics. The inertia and stiffness values of the low speed shaft, as determined from the dimensions, are needed as input to the dynamic simulation.

Distributed mass and radial distance necessary for input to MOSTAB are obtained from Figure 17 and the chord length comes from the PROPCODE input. The first flapping mode shape of the blade is also needed for input to MOSTAB. This involved performing a natural frequency analysis utilizing Myklestad's method for rotating beams (10). The Myklestad program requires input at each radial station consisting of the lumped mass, length between stations and a blade stiffness value scaled down from 50 foot blade data. The natural frequency of the blade as obtained from this program for an operating range of 36 to 60 rpm varies from 4.5 to 5.0 Hz.

MOSTAB was run at various rotor speeds and wind velocities to determine the magnitude of the loads on the WTG hub reacting on the low speed shaft to determine a shaft size. The design values were obtained for a rotor speed of 80 rpm and wind speed of 50 mph. It is important that the low speed shaft is over designed to carry high loads to prevent costly breakage. Since the operating range of the WTG should not exceed 60 rpm, a 33 percent margin of safety results from this design rotor speed. Table 5 presents the dynamic loads at the rotor hub obtained from MOSTAB and Figure 18 illustrates the positive sign direction and coordinate system of these loads. The most prominent loading on the WTG hub was the mean flatwise moment. Figure 19 illustrates the variation of this load vs. wind and rotor speed.



X - Axial Force

Y - Horizontal Shear Force

Z - Vertical Shear Force

L - Twist Moment

M - Flatwise Moment

N - Chordwise Moment

Figure 18 Positive Sign Direction of Blade Loads

Figure 19 Mean Flatwise Moment at Rotor Hub vs. Wind Speed
for Various Rotor Speeds

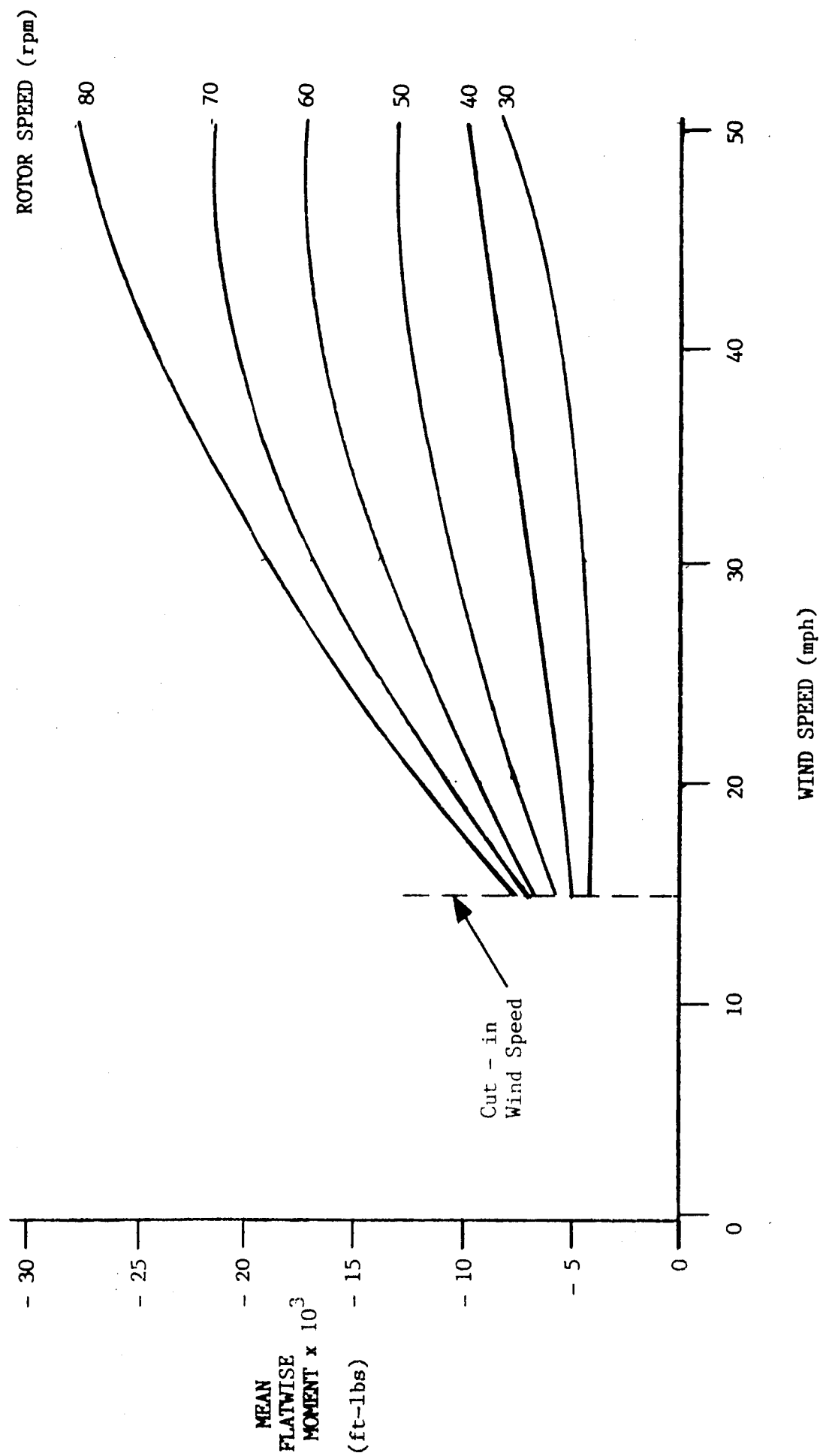


TABLE 5

Dynamic Loading at Rotor Hub Obtained from MOSTAB for a
Design Rotor Speed of 80 rpm and Wind Speed of 50 mph

Axial Force in X-Direction:	Mean = - 9700 lbs
	Cyclic = 410 lbs
Horizontal Force in Y-Direction:	Mean = 390 lbs
	Cyclic = 410 lbs
Vertical Shear in Z-Direction:	Mean = - 1600 lbs
	Cyclic = 80 lbs
Flatwise Moment in X-Z Plane:	Mean = - 327000 in-lbs
	Cyclic = 17600 in-lbs
Chordwise Moment X-Y Plane:	Mean = - 89000 in-lbs
	Cyclic = 53800 in-lbs
Twist Torque in Y-Z Plane:	Mean = 940 in-lbs
	Cyclic = 460 in-lbs

LOW SPEED SHAFT DESIGN

A low speed shaft was designed using the loads presented in Table 5 along with a design torque of 264,000 in-lbs due to a 50 mph wind and 80 rpm rotor acting in the Y-Z plane as determined by PROPCODE. A force of 4000 lbs in the Z-direction resulting from the weight of the blades and hub was also considered. These forces, moments and torques were converted to stresses which were combined by addition if acting in the same direction. Resulting values were obtained for stresses in the same plane. As expected, the moment and torque values dominate and therefore represent the constraint for the shaft size. An equivalent mean and alternating stress was determined from the resulting stress relations and these final values are shown in Equations 4 and 5 with diameter as a variable. For a hollow shaft, the diameter term is equal to Equation 6.

$$\text{Equivalent Mean Stress} = \frac{4,200,000}{\text{Diameter}^3} \text{ psi} \quad (4)$$

$$\text{Equivalent Alternating Stress} = \frac{581,000}{\text{Diameter}^3} \text{ psi} \quad (5)$$

$$\text{Diameter}^3 = \frac{\text{Outside Diameter}^4 - \text{Inside Diameter}^4}{\text{Outside Diameter}} \text{ in}^3 \quad (6)$$

AISI 4340 steel with a yield strength of 99 ksi and an ultimate tensile strength of 111 ksi was chosen as a material for the low speed shaft because of the high strength characteristics. A Goodman Diagram was plotted based on shear strengths since torsional stress was the limiting factor due to the high torque on the shaft. The shear yield strength is 58 percent and the shear ultimate strength is 80 percent of the normal values. The 10 million cycle strength for infinite life was taken from Equation 7 with a surface factor (CS) of 0.73, for machining, a load factor (CL) of 0.58 for torsion and a diameter factor (CD) of 0.9.

$$\text{Infinite Life Strength} = \frac{\text{Ultimate Strength} * \text{CS} * \text{CL} * \text{CD}}{2} \text{ psi} \quad (7)$$

Equations 4 through 6 were used to determine stress values for various shaft diameters. The results were compared on the Goodman Diagram shown in Figure 20 to determine the factor of safety and also to check for an infinite life expectancy (11). Outer diameters of 5, 6, 7 and 8 inches were investigated corresponding to safety values of 1.3, 2.3, 3.6 and 5.4 respectively. A shaft with outside diameter of 7 inches and inside diameter of 3 inches was selected to provide a factor of safety of 3.6. This value is very conservative since the shaft was designed for wind and rotor speeds above the operating range of the WTG.

The inertia and stiffness of the low speed shaft were calculated using Equations 8 and 9 with a shaft length of 65 inches where G is the torsional modulus of rigidity. Inertia and stiffness values for all the components are included in Table 7.

$$\text{Stiffness} = \frac{G * \pi}{32 * \text{Length}} * (\text{OD}^4 - \text{ID}^4) \text{ in-lb/rad} \quad (8)$$

$$\text{Inertia} = \frac{\pi * \text{Length} * \text{Density}}{32} * (\text{OD}^4 - \text{ID}^4) \text{ in-lb-sec}^2 \quad (9)$$

LOW SPEED SHAFT BEARINGS

Bearings were selected to support the low speed shaft so that all components are connected to the bedplate. Double row tapered spherical roller bearings were used because of their high load carrying capability. Two bearings were chosen which introduce two variable distances: the measurement from the hub center to the first bearing and the distance between bearings. The first was chosen to be 32 inches allowing for half the hub and the low speed shaft flange which fastens to the hub. The second was determined based on load and size restrictions. The rear bearing was subjected to both a radial and thrust load and the front bearing was required to carry only a radial load. This arrangement adjusts mainly for shaft expansion due to temperature changes and also to provide an allowance for misalignment.

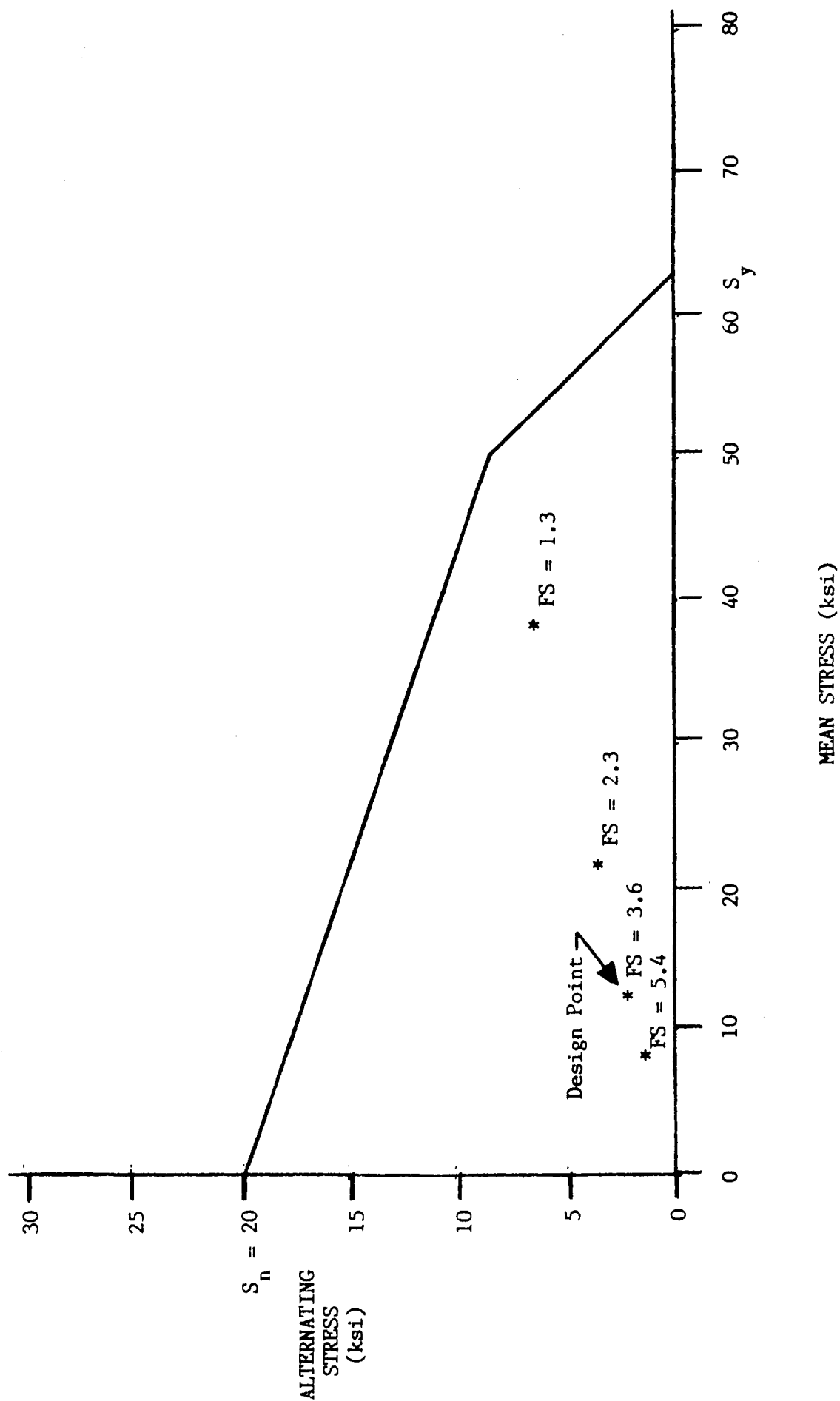
Moments and shear forces acting on the shaft in both the radial Y and Z directions were taken into account in addition to an axial thrust force. These loads were converted into equivalent design values as given in Equations 10 through 13 with "B" representing the unknown distance between bearings.

Outboard Bearing:

$$\text{Vertical Force} = \frac{76,300}{B} - 400 \text{ lbs} \quad (11)$$

$$\text{Horizontal Force} = \frac{569,600}{B} + 7600 \text{ lbs} \quad (10)$$

Figure 20 Goodman Diagram for Low Speed Shaft
 AISI 4340 Steel
 Based on Shear Strengths



Inboard Bearing:

$$\text{Vertical Force} = \frac{76,300}{B} + 30,000 \text{ lbs} \quad (13)$$

$$\text{Horizontal Force} = \frac{569,600}{B} + 30,000 \text{ lbs} \quad (12)$$

A distance of 25 inches was chosen for "B" because this provided a compromise between bearing loads and shaft length. Both the front and rear bearings are SKF-SDAF 22638 complete pillow block assemblies. The front bearing will have provisions in the pillow block to allow for the floating capability.

DISK BRAKE

A disk brake is connected to the low speed shaft to include the inertia effects from the disk in the simulation. The brake is placed before the speed increaser to reduce the resulting rotational dynamic effects as an alternative to a disk on the high speed end. The disadvantage of this is that a larger brake is needed to overcome the large torque resulting from the wind on the blades.

A Wichita ATD-130-H air disk brake was chosen which is rated at 193,000 in-lbs of torque. This is double the maximum torque the WTG generates within the normal operating range. It is assumed that the brake is engaged if rotor speed exceeds 60 rpm and thus should theoretically never be subjected to a torque near the rated value. Dimensions were obtained for the brake hub and disk from manufacturer's literature to determine an inertia value. The hollow feature of the disk necessary for cooling and low inertia characteristics was considered in this calculation also.

SPEED INCREASER

A gearbox size was chosen based on the operating speed range and torque requirements. The rotor operates from 36 to 60 rpm and the CVT ratio ranges from 2:1 to 3.33:1. In order to produce a speed range at the generator from 1800 to 1854 rpm for 3 percent slip, the gearbox must supply a gear ratio of about 15:1. A design torque of 153,000 in-lbs was obtained from PROPCODE for a 60 rpm rotor and 50 mph wind speed. This value was doubled to implement a load safety factor of 2 for continuous operation. A Crichton 24,000 series speed increaser capable of 323,000 in-lbs of torque was selected with a 15.15:1 ratio. The planetary speed increaser is preferred for this application because the drive train centerline is not altered due to off center input and output shafts.

The inertia and stiffness of the speed increaser was determined by approximating the dimensions of the gears and carriers within the planetary speed increaser. This was accomplished by the use of the gearbox outer dimensions and a pictorial exploded view of the internal components. This was necessary since detailed information was not available for this component.

The gearbox is a two stage device including two separate planetary speed increases. The stiffness and inertia of the separate internal pieces were combined into three distinct groups determined by the magnitude of the speed increase each goes through. These were then combined into equivalent single values for the entire speed increaser. This follows the methodology for the gearbox parameters as presented in (12). The total stiffness for the gearbox was reduced by about 25 percent to adjust for flexibility in the gear teeth.

CONTINUOUSLY VARIABLE TRANSMISSION

This component is the most important in the system because the implementation of a CVT in a wind turbine drive train is the purpose of this project. A CVT with a simple method of ratio control was used here to keep cost to a minimum. The 75 horsepower CVT manufactured by Kumm Industries has a speed ratio variation from 2:1 to 3.33:1. The input pulley of the transmission is of a fixed radius while the output pulley diameter is varied by means of hydraulic control producing a near instantaneous adjustment of the speed ratio. The CVT belt is manufactured of neoprene and includes chords consisting of Kevlar resulting in a stiff plastic-rubber composition. Figure 21 is a sketch of the internal parts of the CVT.

The inertia and stiffness of the CVT was determined in a manner similar to that of the speed increaser except a scaled blueprint drawing and a sketch of the internal parts of the CVT were available. Pulley and shaft values were determined using Equations 8 and 9 with dimensions obtained from the drawings. The belt inertia was represented as added weight on the pulleys and the stiffness was determined from Young's modulus, the length of the belt in tension and the input pulley radius, squared.

INDUCTION GENERATOR

The generator chosen for this application is a three phase Reliance-XT 3651S producing an output of 56 kW (75 hp) at 460 volts. This generator includes a fan for cooling and a space heater for low temperature operation. This component is an extreme heavy duty model since it is subjected to a near constant mode of operation.

The generator contributes an inertia and stiffness due to both the mechanical and electrical aspects. The mechanical inertia and stiffness of the generator shaft were obtained by approximating the internal rotor size from external dimensions. The electrical characteristics were determined by using an equivalent field stiffness and grid inertia. An estimated value for the field stiffness was obtained from (12). This final stiffness value was connected to an electrical grid of infinite inertia where a large finite value was used for computational purposes.

DRIVE TRAIN COUPLINGS

Three couplings are necessary to connect the low speed shaft, speed increaser, CVT and generator together in series. Double engagement gear couplings were selected for each location to adjust for misalignment between

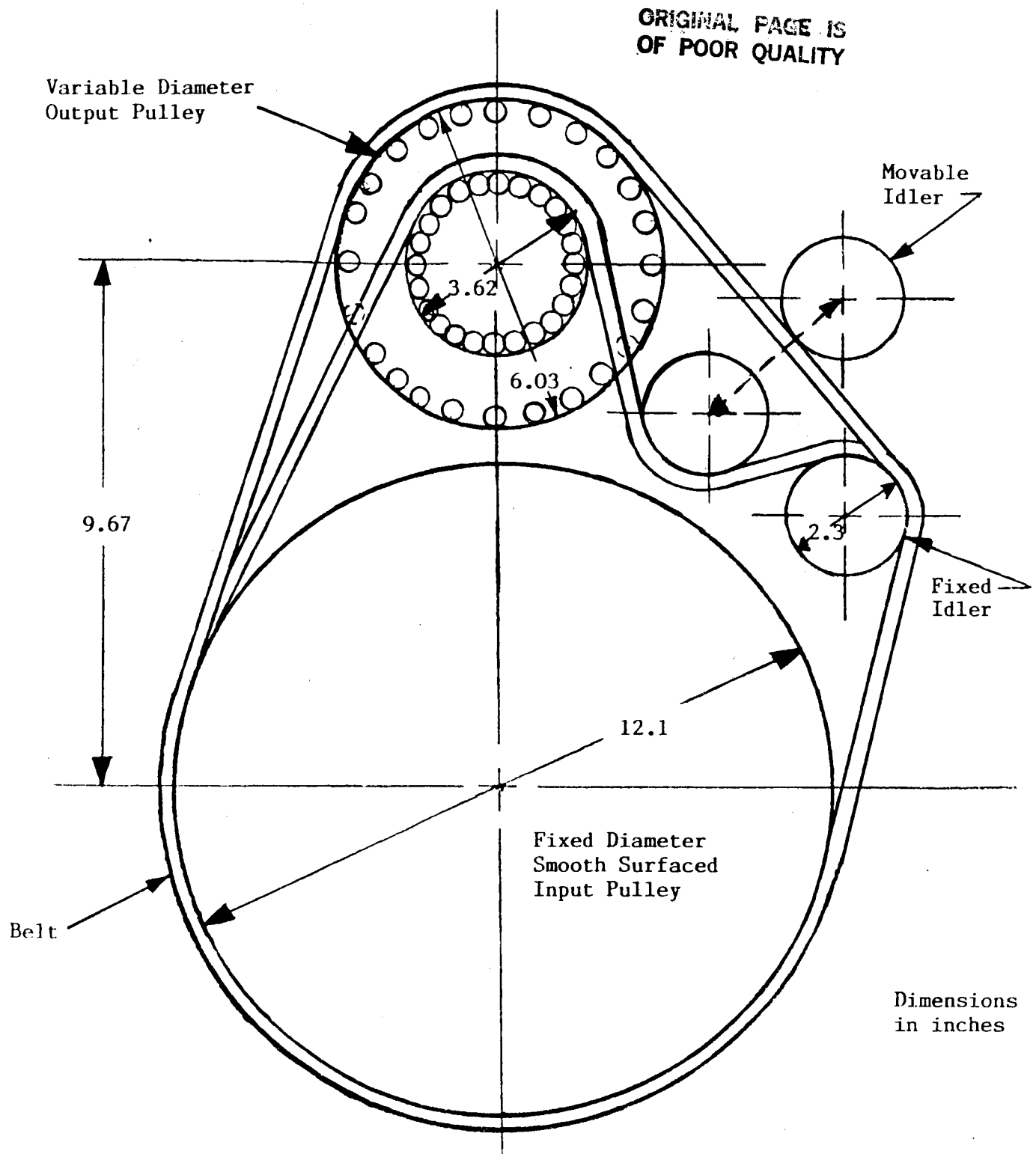


Figure 21 Diagram of CVT

components or for expansion and contraction of shafts due to temperature changes. The design constraints necessary to determine a coupling size are the amount of torque each can withstand and the maximum allowable bore based on coupling strength. In some cases, a spacer is needed to facilitate assembly and to assure a secure attachment. Information on power and size restrictions obtained from catalogues are used to select these components.

The coupling between the low speed shaft and speed increaser is a Falk G20-1050G including a 6 inch spacer. A Falk G20-1030G coupling connects the gearbox and CVT. This is essentially the same as the previous coupling except in a smaller version since transmitted torque is lower due to the speed increase. Appropriate inertia and stiffness values were obtained from tables supplied by the manufacturer.

The final coupling connecting the CVT to the generator is a Zurn F101.5 with a 3 inch spacer necessary for assembly. The inertia and stiffness values supplied by the manufacturer are shown in Table 7.

Table 6 represents a summary of the drive train components and Table 7 lists the inertia and stiffness data calculated for each.

TABLE 6

Component Specifications in 56 kW WTG Drive Train

Rotor -
Three - Bladed with a Blade Length of 26.3 feet, Zero Coning, Upwind Position, NACA 23024 Airfoil
Low Speed Shaft -
7 in. Outside, 3 in. Inside Diameter, AISI 4340 Steel
Bearings -
SKF SDAF 22638 Pillow Block Assembly
Coupling 1 -
Falk G20 - 1050G Double Engagement Gear, 6 in. Spacer
Speed Increaser -
Crichton 24,000 Model Two Stage Planetary, 15.15:1 Overall Ratio
Coupling 2 -
Falk G20 - 1030G Double Engagement Gear Type
Continuously Variable Transmission -
Kumm Industries, 75 HP, 2:1 - 3.33:1 Speed Ratio
Coupling 3 -
Zurn F101.5 Gear Coupling, 3 in. Spacer
Generator -
Reliance XT-365TS, 56 kW, 460 Volt Output, Three Phase, Cooling Fan, Space Heater

TABLE 7

Component Inertia and Stiffness Values

COMPONENT	ROTATIONAL INERTIA (in-lb-sec ²)	TORSIONAL STIFFNESS (in-lb/radian)
3 Blades	8.49×10^4	-
Hub	2.30×10^3	-
Low Speed Shaft	1.04×10^1	3.9×10^7
Brake	4.50×10^1	-
Coupling 1	3.20×10^1	6.1×10^8
Gearbox	1.61×10^3	9.5×10^7
Coupling 2	3.83×10^{-1}	8.1×10^7
CVT Input Shaft	7.50×10^{-3}	1.7×10^6
CVT Input Pulley	4.80×10^{-1}	-
CVT Idlers	2.70×10^{-3}	-
CVT Belt (on input)	3.90×10^{-4}	1.8×10^6
(on output)	1.30×10^{-4}	1.8×10^6
CVT Output Pulley	2.85×10^{-1}	-
CVT Output Shaft	7.50×10^{-3}	1.3×10^6
Coupling 3	3.10×10^{-1}	2.5×10^6
Generator Rotor/Shaft	7.80×10^0	7.0×10^6
Generator Field	-	1.2×10^4
Electrical Grid	1.0×10^{20}	-

APPENDIX B - ACSL PROGRAM INPUT LISTING

```
//VM#R0000 JOB (R0000,0000000000,5,300),ACSL,CLASS=L
// EXEC ACSLCLG
//ACSL.SYSIN DD *
PROGRAM 56KW WIND TURBINE DRIVE TRAIN DYNAMIC SIMULATION
INITIAL
'----- DEFINE ALL THE PRESET VARIABLES-----'
  CONSTANT J1 = 87200.0 , J2 = 864.0 , J3 = 804.0
  CONSTANT J4 = .872 , J5 = .296 , J6 = 8.11 , J7 = 1.0E+15
  CONSTANT K1 = 3.66E+07 , K2 = 9.54E+07 , K3 = 1.26E+06
  CONSTANT K4 = 7.48E+05 , K5 = 5.46E+05 , K6 = 1.2E+04
  CONSTANT R = -13.14 , M1 = 15.15 , PI = 3.14159265
  CONSTANT CVADJ = 100. , WNOT = 30. , SLIP = .03 , TSTP = 43.999
  CINTERVAL CINT = .01
  TICNT = 0.0
'--AT THE END OF EACH CINT, GENERATOR TORQUE AND RPM IS CALCULATED--'
'-- AT THE END OF EACH CVADJ, THE CVT RATIO IS ADJUSTED --'
'----INPUT FUNCTION FOR AERODYNAMIC DAMPING BASED ON WIND SPEED----'
  R1A = 18417.0-3896.4*WNOT+139.88*WNOT**2+3.1024*WNOT**3
  R1B = -0.12048*WNOT**4+0.24939E-04*WNOT**5+0.14284E-04*WNOT**6
  R1 = R1A + R1B
'-----SET INITIAL CONDITIONS FOR ROTOR SPEED OMEGA---'
  TNOT1 = -43686. + 12700.*WNOT - 1265.7*WNOT**2
  TNOT2 = 65.165*WNOT**3 - 1.0652*WNOT**4
  TNOT = TNOT1 + TNOT2
  IF (WNOT .GT. 30.) TNOT = 97509.
  IF (WNOT .LE. 6.0) TNOT = 0.0
'-----CALCULATE ROTOR SPEED FOR INITIAL WIND SPEED-----'
  IF (WNOT .LE. 10.) OMRPM = 36.
  IF (WNOT .GT. 10. .AND. WNOT .LE. 18.) OMRPM = 3.0*WNOT + 6.0
  IF (WNOT .GT. 18. .AND. WNOT .LE. 22.) OMRPM = 60.
  IF (WNOT .GT. 22. .AND. WNOT .LE. 30.) OMRPM = -1.4375*WNOT+91.625
  IF (WNOT .GT. 30.) OMRPM = 48.5
  OMEGA = OMRPM*PI/30.
'-----CALCULATE VALUES FOR SYSTEM BASED ON INITIAL CONDITIONS-----'
  POW = TNOT*OMEGA/8843.6
  GENRAD = ((POW*SLIP/56.) + 1.)*188.49556
  GENRPM = GENRAD*30./PI
  M2 = GENRAD/(15.15*OMEGA)
  R2 = -1.0*R
'----- CALCULATE INTEGRATION INITIAL CONDITIONS -----'
  Q1IC = TNOT/K1
  Q2IC = TNOT/K2
  Q3IC = TNOT/K3
  Q4IC = TNOT/K4
  Q5IC = TNOT/K5
  P1IC = J1*OMEGA
  P2IC = J2*OMEGA
  P3IC = J3*OMEGA
  P4IC = J4*OMEGA
  P5IC = J5*OMEGA
  P6IC = J6*OMEGA
  P7IC = J7*OMEGA
END $'OF INITIAL'
DYNAMIC
```

```

'-----INPUT WIND FUNCTION OVER TIME-----'
'---TRIANGULAR PULSE STARTING AT T=0 FOR INITIAL WIND OF 20 MPH---'
  IF (T .GE. 0.0 .AND. T .LT. 10.0) WIND = 20.
  IF (T .GE. 10.0 .AND. T .LT. 13.0) WIND = 10.0*T - 80.
  IF (T .GE. 13.0 .AND. T .LT. 16.0) WIND = -10.0*T + 180.
  IF (T .GE. 16.0) WIND = 20.
'-----CHANGE ROTOR SPEED DUE TO CHANGING WIND-----'
  IF (WIND .LE. 10.) OMRPM = 36.
  IF (WIND .GT. 10. .AND. WIND .LE. 18.) OMRPM = 3.0*WIND + 6.0
  IF (WIND .GT. 18. .AND. WIND .LE. 22.) OMRPM = 60.
  IF (WIND .GT. 22. .AND. WIND .LE. 30.) OMRPM = -1.4375*WIND+91.625
  IF (WIND .GT. 30.) OMRPM = 48.5
  OMEGA = OMRPM*PI/30.
'-----CALCULATE AERODYNAMIC DAMPING BASED ON NEW WIND SPEED-----'
  R1A = 18417.0-3896.4*WIND+139.88*WIND**2+3.1024*WIND**3
  R1B = -0.12048*WIND**4+0.24939E-04*WIND**5+0.14284E-04*WIND**6
  R1 = R1A + R1B
'-----CALCULATE TORQUE FROM WIND FOR MAXIMUM POWER EXTRACTION-----'
  TORQ1 = 38456. - 11028.*WIND + 951.78*WIND**2 - 11.554*WIND**3
  TORQ2 = -.59013*WIND**4 + .16891E-01*WIND**5 - .12249E-03*WIND**6
  TORQ = TORQ1 + TORQ2
'-----'
DERIVATIVE
  MAXTERVAL MAXT = .0005
'--- MAXT IS THE INTEGRATION TIME INCREMENT ----'
'--- NUMBER OF INTEGRATION STEPS TO NEXT CINT==> NSTP=CINT/MAXT ----'
'-----STATE EQUATIONS -----'
  PDOT1 = TORQ + R1*(OMEGA - P1/J1) - K1*Q1
  QDOT1 = P1/J1 - P2/J2
  PDOT2 = K1*Q1 - K2*Q2
  QDOT2 = P2/J2 - P3/J3
  PDOT3 = K2*Q2 - M1*K3*Q3
  QDOT3 = (M1/J3)*P3 - P4/J4
  PDOT4 = K3*Q3 - K4*Q4
  QDOT4 = P4/J4 - P5/J5
  PDOT5 = K4*Q4 - M2*K5*Q5
  QDOT5 = (M2/J5)*P5 - P6/J6
  PDOT6 = (J6/(J6 + R*R/K6))*(K5*Q5 - (R/J7)*P7 - (R2/J6)*P6)
  PDOT7 = (R/J6)*P6
'----- INTEGRATE STATE EQUATIONS -----'
  P1 = INTEG(PDOT1 , P1IC)
  Q1 = INTEG(QDOT1 , Q1IC)
  P2 = INTEG(PDOT2 , P2IC)
  Q2 = INTEG(QDOT2 , Q2IC)
  P3 = INTEG(PDOT3 , P3IC)
  Q3 = INTEG(QDOT3 , Q3IC)
  P4 = INTEG(PDOT4 , P4IC)
  Q4 = INTEG(QDOT4 , Q4IC)
  P5 = INTEG(PDOT5 , P5IC)
  Q5 = INTEG(QDOT5 , Q5IC)
  P6 = INTEG(PDOT6 , P6IC)
  P7 = INTEG(PDOT7 , P7IC)
END $'OF DERIVATIVE--'
'----- CALCULATE OUTPUT PARAMETERS AT GENERATOR -----'

```

```

GENRAD      = 188.49556*(1. - PDOT7*SLIP/2551.)
GENRPM      = GENRAD*30./PI
'-----ADJUST CVT AT INTERVALS EVERY CVADJ SECONDS-----'
CONSM       = TICNT + CVADJ - CINT/2.
CONSP       = TICNT + CVADJ + CINT/2.
IF (T .GT. CONSM .AND. T .LT. CONSP) TICNT = T
IF (T .GT. CONSM .AND. T .LT. CONSP) M2 = GENRAD/(15.15*OMEGA)
'----- CALCULATE POWER AND UPDATE PARAMETERS FOR INTEGRATION -----'
POW         = PDOT7*GENRPM/84450.
PPS         = POW*(-1.0)
R2          = (2551./SLIP)*(1./188.49556 - 1./GENRAD)
'-----SPECIFY TERMINATION CONDITION-----'
TERMT(T.GE.TSTP)
END $'OF DYNAMIC'
END $'OF PROGRAM'
//GO.SYSIN DD *
SET TITLE = 'DYNAMIC SIMULATION OF A 56KW WIND TURBINE DRIVE TRAIN '
SET DIS=99 $' FORCE 3 COL FORMAT FOR OUTPUT '
SET NDBUG = 0
OUTPUT T,      OMRPM, CVADJ, QDOT2, P5, ...
WIND, R1, PDOT1, QDOT3, P6, ...
TORQ, R2, PDOT2, QDOT4, P7, ...
PDOT7, M2, PDOT3, QDOT5, Q1, ...
GENRPM, M1, PDOT4, P1, Q2, ...
POW, R, PDOT5, P2, Q3, ...
TNOT, K6, PDOT6, P3, Q4, ...
WNOT, TICNT, QDOT1, P4, Q5, 'NCIOUT' = 50
PREPAR T,PDOT7,GENRPM,POW,WIND
SET NGXPPL=50,NGYPPL=20,NPXPPL=100,NPYPPL=100
START
PLOT WIND
PLOT POW
STOP
//GO.FT99F001 DD SYSOUT=A
/*
//

```

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16. Abstract This report covers the dynamic analysis of a horizontal axis fixed pitch wind turbine generator (WTG) that is rated at 56 kW. A mechanical Continuously Variable Transmission (CVT) was incorporated in the drive train to provide variable speed operation capability. One goal of the dynamic analysis was to determine if variable speed operation, by means of a mechanical CVT, is capable of capturing the transient power in the WTG/wind environment. Another goal was to determine the extent of power regulation possible with CVT operation.					
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