GUY CABLE DESIGN AND DAMPING FOR VERTICAL AXIS WIND TURBINES

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ABSTRACT

Guy cables are frequently used to support vertical axis wind turbines since guying the turbine reduces some of the structural requirements on the tower. The guys must be designed to provide both the required strength and the required stiffness at the top of the turbine. The axial load which the guys apply to the tower, bearings, and foundations is an undesirable consequence of using guys to support the turbine. Limiting the axial load so that it does not significantly affect the cost of the turbine is an important objective of the cable design. The lateral vibrations of the cables is another feature of the cable design which needs to be considered. These aspects of the cable design are discussed in this paper, and a technique for damping cable vibrations is mathematically analyzed and demonstrated with experimental data.

INTRODUCTION

Most vertical axis wind turbines use guy cable to support the top of a single, fully rotating central tower. Other designs, such a cantilevered tower or two concentric towers (one stationary to react cable loads) have been found to be, generally, less cost effective. The guy cables serve two primary functions while supporting the turbine. They provide the strength necessary to hold the turbine during hurricane winds, and they provide the stiffness at the top of the rotor. The strength and stiffness requirements are not competing design objectives, however, one or the other may "drive" the various parameters involved in the cable design.

There are two important consequences of using guy cables to support the rotor. First, since guys have an initial tension, the vertical component of the tension is reacted through the tower, the foundation and, in this case, the bearings. This axial load imposed by the guys impacts the cost of these components. The second consequence of the guy support is lateral vibrations of the cables. Cable vibration will be excited by the motion of the top of the turbine while it is operating. There is always excitation to the cables as long as the turbine is operating in wind. If the excitation frequency is near one of the cable natural frequencies, then the amplitude of vibration can be quite large. Large amplitude cable vibrations need to be avoided in order to reduce fatigue in the cables and their terminations and also to maintain a reasonable blade/cable clearance distance. Other aspects of the cable design, which include the cable sag, the required blade/cable clearance distance, thermal expansion effects, and cable anchors, are discussed in some detail in Ref. 1-5.

DESIGN GUIDELINES

There are a number of cable parameters which have to be chosen in order to establish the design for the guys. These parameters are the cross-sectional area of a cable, the number of cables, the length of the cables, the pretension of the cables, the cable elevation angle, and the cable material (density and modulus). All of these parameters affect the two key design requirements on the guys, their support strength and the stiffness they create at the top of the turbine [Ref. 3]. These cable parameters also affect the consequences of the cable design which include lateral cable vibrations, cable sag, and the resulting axial load on the turbine due to the tension in the guys [Ref. 3].

The strength requirement on the guys is the easiest to specify. The cross-sectional area of the cables must be sufficiently large to support the turbine during parked survival in hurricane winds without the tension in the cables exceeding their ultimate strength divided by a factor of safety. A factor of safety of at least three is recommended for cables with an expected life of thirty years. This factor of safety is not conservative when compared to other cable applications and should be increased if the turbine will be placed in an environment which is particularly harsh for corrosion. The tension in cables during the parked survival condition will be the initial tension plus the increase (or decrease) due to the wind drag on the turbine. Thus the value of the initial tension will affect the cable tension during parked survival unless the initial tension is sufficiently low that the downwind cable goes slack during the hurricane winds.

The stiffness requirement for the guys is impossible to generalize for all turbine designs. The required stiffness will have to be chosen interactively with the rest of the turbine design. This is due to the fact that the stiffness at the top of the turbine affects the frequencies of the natural modes of vibration. However, these frequencies are also controlled by the whole turbine structure especially the blades and the tower. Consequently, depending on the desirable values for these modal frequencies, the operational frequency of the turbine, and the structural properties of the turbine, the guy cable stiffness will have to be chosen accordingly.

Figure 1 shows the variation in the modal frequencies with the guy stiffness for the 17-Meter Low Cost Turbine. Examining this figure, one can see that some of the modes are quite sensitive to the guy stiffness. Further, we can see that, in order to keep the modal frequencies away from the excitation frequencies, the guy stiffness is forced to be greater than three k.

Another consideration for choosing the cable stiffness is the allowable angle change for the bearings at the bottom of the rotor. If the bearing can only allow a small angle change of the rotor, then the cable stiffness will have to be sufficiently high to restrain the rotor from leaning. However, if a universal joint is used to protect the bearing or some more tolerant bearing design is used, then this constraint on the stiffness will not apply.

The frequencies of lateral vibrations of the guy cables are a consequence of the guy design, and they can place a constraint on the design. The natural frequencies of vibration of the cables are $f_n = n(T/\rho A)^{\frac{1}{2}}/2L$ [Ref. 3] where n is the mode number, L is the cable length, T is the guy tension, A is the cable cross-sectional area, and ρ is the mass density. In many designs these parameters cannot be chosen so that the first cable frequency is above the excitation frequency of 2.0/rev (for a two bladed turbine) since that would force the tension to be larger than would otherwise be desirable. Further, since the tension in the cables vary with the wind velocity and temperature, it may not be possible to completely avoid all cable resonances

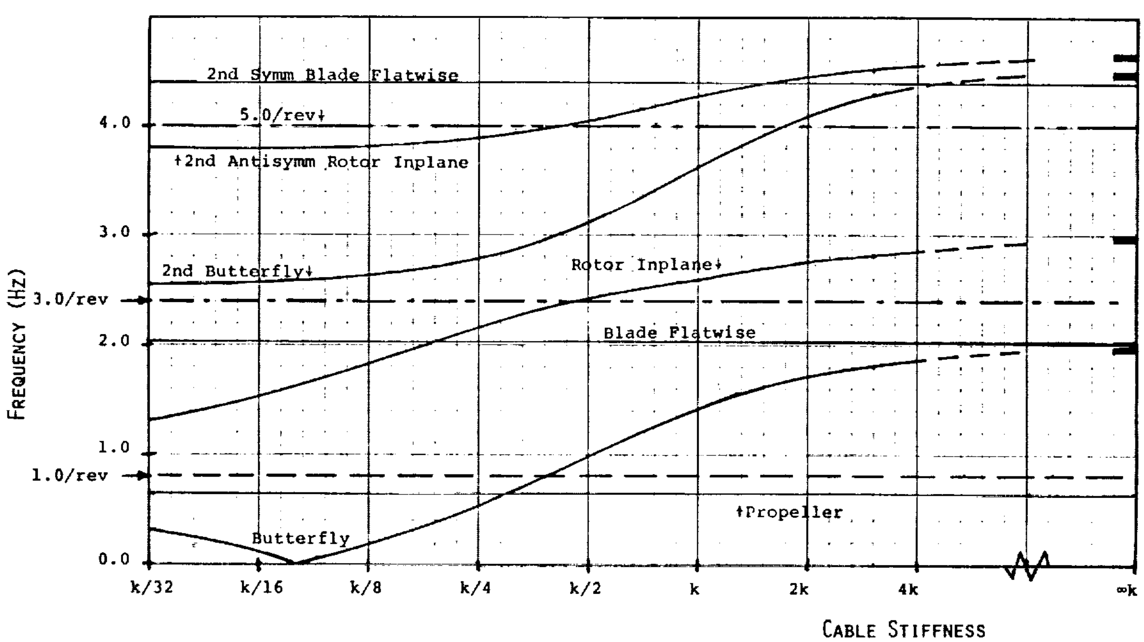

Figure 1 - Modal Frequencies Versus Cable Stiffness

k = 4440 lbs/in, speed = 48.1 rpm

in any case. The cable vibration problem is discussed in the last section of this paper and a technique for restraining the vibrations is analyzed and demonstrated.

The sag of the guy cables is another consequence of the guy design, and it must be evaluated and compared with the blades/cable clearance distance. If the sag is too large for available clearance, then the cable design will have to be changed. More discussion of the clearance requirements can be found in Ref. 3.

The axial load which is applied to the turbine by the guys can be the most serious consequence of the guy design. Reducing the axial load within the constraints of the design will reduce the cost of the turbine. Nellums in Ref. 6 has shown that reductions in the costs of the bearings and tower result from reducing the cable imposed axial load for a 150 foot tall turbine. Foundation costs would also be reduced.

The axial load is simply the total of the guy tensions times the sine of the cable elevation angle at the top of the turbine. Reducing either the tension or the elevation angle will reduce the axial load. However, both of these changes also decrease the guy stiffness, so the required stiffness will restrict the design choices. It can be shown that to obtain a minimum axial load, for a given stiffness, the elevation angle should be 35.3 degrees.

The axial load on the turbine will change as the guy tensions change, so during the hurricane survival the axial load may increase. Depending on the tower and bearings designs, an increase in the axial load during the parked survival condition may or may not affect the cost of these components.

In view of the requirements and consequences of the cable design, it is clear that the design process must be iterative and interactive with the total turbine design. The starting point for the cable design must be the strength requirement. Then, using an elevation angle of thirty-five degrees, determine a tension and an area for the stiffness requirement. This new area and tension will then be factored back into the strength requirment, and then the stiffness reevaluated, and during this design iteration the axial load, cable sag, and cable natural frequencies need to be computed and evaluated for acceptability.

CABLE VIBRATIONS

If the design of the guys results in the first cable frequency being less than n per rev where n is the number of blades, then lateral vibrations of the cables can result. Excessive cable vibration could cause a blade strike or fatigue the cable terminations or anchors. Excitation of the cables always exists since the top of the rotor moves while it is operating, and the cables have exceptionally low inherent damping (less than 0.2 percent of critical), so very high resonant responses can result. Experience has shown that if the first cable frequency is above the primary excitation frequency, n per rev, then there is no problem with cable vibrations. However, keeping cable frequencies that high can be costly, particularly for large turbines.

There are two direct solutions to alleviate the cable vibration problem. One is to constrain the cable so that the cable modes are shifted to a higher frequency. This can be done by forcing one or more nodal points along the span of the cable. The other solution would be to add damping to the cable so that resonant responses, when they occur, would be limited in magnitude.

The rest of this paper will discuss a cable damping system which was developed for this purpose.

The concept for the dampers utilizes Coulomb friction to dissipate the energy. It is simply a pair of weights which are suspended from the cable and slide on two inclined surfaces whenever the cable moves. The two surfaces are at right angles to each other and at right angles to the cable, thus they can damp the motion of any lateral cable vibration. The dampers can be placed near the anchors so that they are out-of-the-way. They can be built inexpensively and require no power to operate them. They could be designed to require very little maintenance and be very reliable. Figure 2 shows a diagram of the damping concept with just one friction surface shown, omitting the other for clarity purposes.

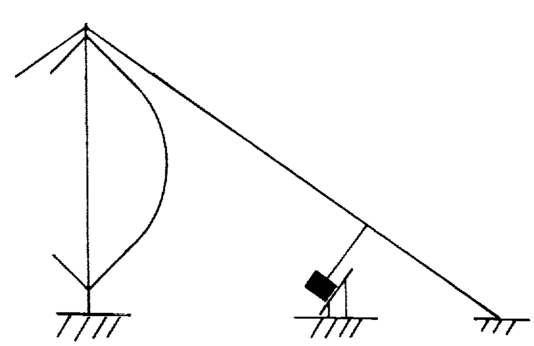

FIGURE 2 - DIAGRAM OF CABLE DAMPING CONCEPT

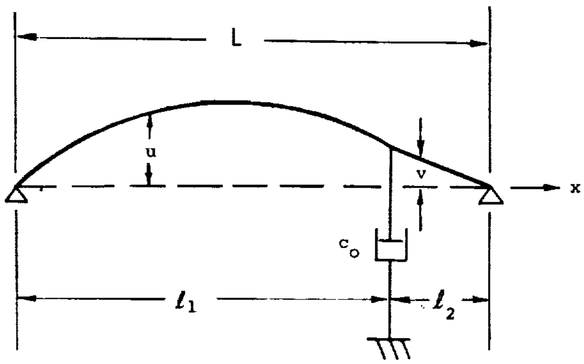

FIGURE 3 - DIAGRAM OF CABLE WITH DAMPER

In order to determine the size, weights, and spacing of this damping concept, an analysis for the damping was performed and is described very briefly below. Figure 3 shows a diagram of the physical system that will be anaylzed with the indicated notation. We will consider motion in only one plane since the out-of-plane motion is uncoupled. Our interest is the modal damping in the cable, so we can consider the homogeneous differential equation and boundary conditions.

$$\rho\ddot{u}(x, t) - T u''(x, t) = 0 \quad \text{for } 0 < x < \ell_1 \text{ and} \tag{1}$$

$$\rho\ddot{v}(x, t) - T v''(x, t) = 0 \quad \text{for } \ell_1 < x < L , \tag{2}$$

where ρ is the mass per unit length, T is the tension, u and v are the displacments, and the dots represent differentiation with respect to time and the primes with respect to x. The boundary conditions and continuity conditions are

$$u(0, t) = v(L, t) = 0 , \tag{3}$$

$$u(\ell_1, t) = v(\ell_1, t) , \tag{4}$$

and

$$T[v'(\ell_1, t) - u'(\ell_1, t)] = c_o\dot{u}(\ell_1, t), \tag{5}$$

where c_o is the viscous damping coefficient. If we take ℓ_2 to be much less than ℓ_1, then v" can be approximated by zero; and the coupled differential equations can be reduced to one differential equation with an inhomogeneous boundary condition. Using separation of variables and assuming sinusoidal motion in the standard way leads to a transcendental equation for the eigenfrequencies, except in this case, the frequencies are complex because of the damping.

$$\frac{\ell_1}{\ell_2}\frac{\tan\beta}{\beta} + i\, c \tan\beta + 1 = 0 , \tag{6}$$

where

$$\beta = \frac{\omega}{\frac{1}{\ell_1}\sqrt{T/\rho}} ,$$

$$c = c_o/\sqrt{T\rho} ,$$

and ω is the natural frequency of vibration. Equation (6) can be solved approximately for the damping coefficient in the first mode, and we find

$$\zeta = \frac{\pi c \lambda^2}{1 + \pi^2 c^2 \lambda^2} \tag{7}$$

where

$$\lambda = \ell_2/L .$$

Equation (7) reveals some reassuring behavior. When c is zero or when c is infinity, the damping coefficient is zero as one would expect physically. Consequently ζ is a maximum at some intermediate value of c. This is easily computed and

$$\zeta_{max} = \lambda/2 \text{ at } c = 1/\pi\lambda \tag{8}$$

Figure 4 shows a series of plots of the damping coefficient ζ as a function of c for various values of λ. One can see the maxima of ζ shift as λ is increased. Also note that the curves become steeper near the maximum ζ with increasing λ.

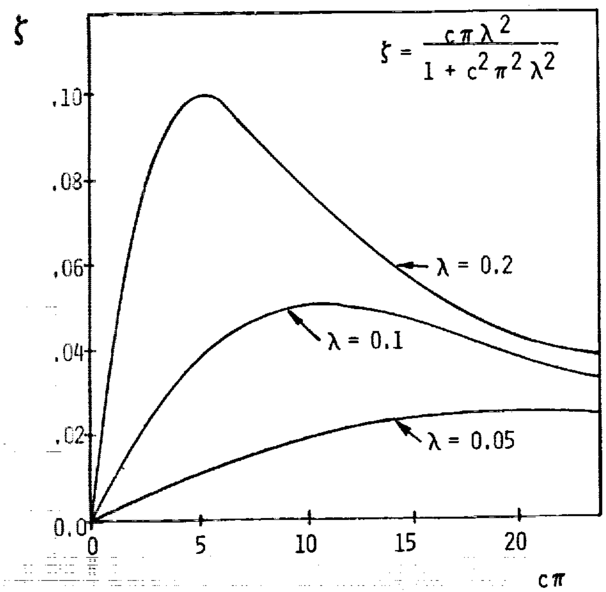

FIGURE 4 - THE VARIATION IN DAMPING FACTOR WITH VISCOUS DAMPING COEFFICIENT

The magnitude of the damping coefficient that can be obtained with this system appears quite attractive. With the damping system connected to the cable one-tenth of the distance along the cable ($\lambda = 0.1$), damping values as high as five percent of critical can be obtained. Even if only two percent damping were achieved, this still represents an increase by a factor of ten over the damping inherently in the cables.

The analysis of the cable damping was performed assuming viscous damping, but the actual damping mechanism is Coulomb friction. By equating the energy dissipated in one cycle of the viscous damper to that of the friction damper, a relationship between the viscous coefficient and friction coefficient can be obtained. The energy dissipated in one cycle is

$$U_D = \int_0^{2\pi/\omega} F_D \cdot \dot{u}\, dt$$

where F_D is the damping force. Evaluating U_D for both the viscous damping and the friction damping and equating the results, we find that the sliding weight W which must be suspended by the cable for the same energy dissipation is

$$W = \frac{c_o\ \pi\ \omega\ u(x_o)}{4\ \mu\ \cos\ \gamma} \qquad (9)$$

where μ is the friction coefficient, γ is the elevation angle of the friction surface, and $u(x_o)$ is the displacement of the cable at the attachment point. Equation (9) reveals the basic nonlinearity of friction damping as opposed to viscous damping; the weight required for equivalent damping is proportional to the displacement $u(x_o)$. Thus, the equivalent c_o goes down with increasing amplitude and goes up with decreasing amplitude. Consequently, an anticipated displacement $u(x_o)$ must be known before an equivalent weight W can be calculated. This nonlinearity also reveals itself when the cable motion is small, then the equivalent c_o is large, and the damper creates a large force relative to the elastic force at the connection. This force just drives a node point and the energy dissipation goes to zero.

This damping scheme was tested on the 17-Meter Research Turbine at Sandia National Laboratories in order to determine the effect of the damping on the cable vibrations. A single pair of friction dampers at right angles were connected to just one of the four guy cables four meters from the cable end ($\lambda = 0.1$). A pair of 18 kg weights were used as the sliding elements. The surfaces were inclined at thirty-five degree from the horizontal and had a friction coefficient $\mu = 0.20$. The size, mass, and location of these dampers were quite reasonable and should have yielded between two to five percent damping depending on the amplitude of the cable motion.

Acceleration measurements were taken on the cable in the horizontal and vertical directions while the turbine was operating in a variety of wind speeds. The acceleration data was analyzed using the Method of Bins [Ref. 7] so that the rms acceleration amplitude could be plotted as a function of wind speed. One would expect the cable vibrations to increase with wind speed since the excitation increases. Measurements were taken with and without the damping system connected and at three different cable tensions, T_1, T_2, and T_3.

The highest tension, T_1 = 80 kN, caused the first cable frequency to be about twenty percent higher than two per rev; the second tension, T_2 = 54 kN, produced a cable frequency very near two per rev; and the third tension, T_3 = 27 kN, created a cable frequency at about seventy percent of two per rev. The Bins data for the horizontal and vertical accelerations with the three tensions are shown in Figures 5-14. The first five figures are the horizontal acceleration plots:

Tension 1 without damping, Tension 2 without damping, Tension 2 with damping, Tension 3 without damping, and Tension 3 with damping. The last five figures are the vertical acceleration plots in that same order. Note the tremendous difference in the acceleration levels between Figs. 5 and 6, and then adding the damper in Fig. 7 brings the acceleration levels back down to the levels of Fig. 5. Figs. 8 and 9 are the undamped and damped cases for tension T_3. There is hardly any difference between the plots. This is the expected result since this case is not a resonant response, but merely a forced response, and damping does very little unless a resonance exists. Figs 10-14 show the same data but for the vertical direction. The same trends are exhibited in these figures.

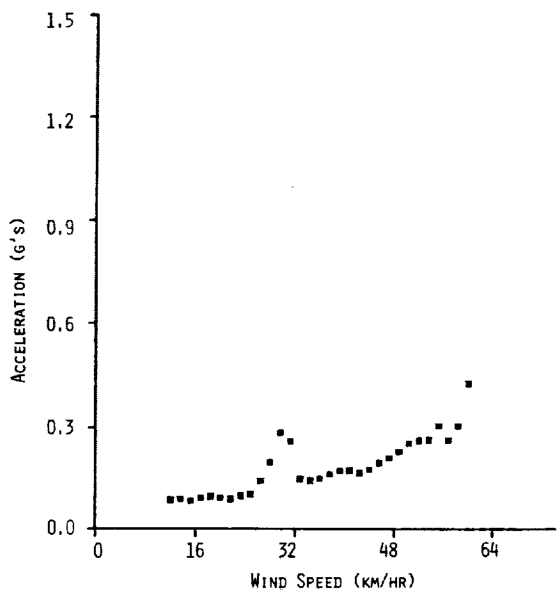

Figure 5 - Cable Horizontal Acceleration Versus Wind Speed for Tension 1, Undamped

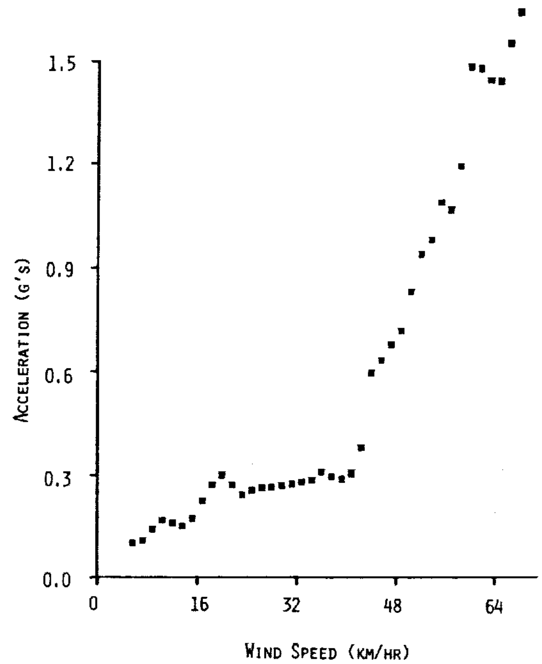

Figure 6 - Cable Horizontal Acceleration Versus Wind Speed for Tension 2, Undamped

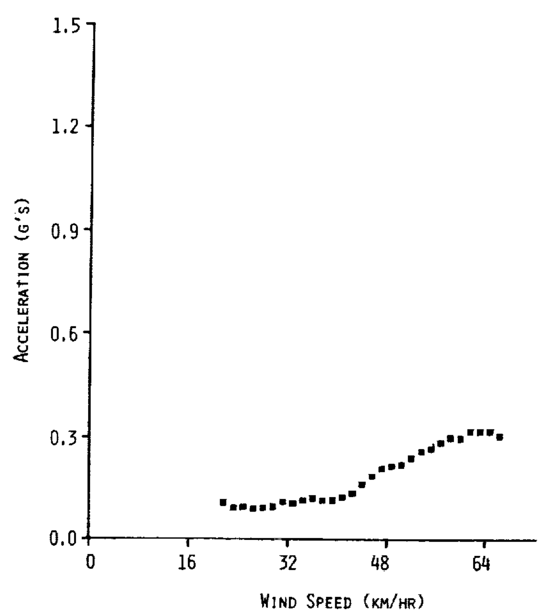

Figure 7 - Cable Horizontal Acceleration Versus Wind Speed for Tension 2, Damped

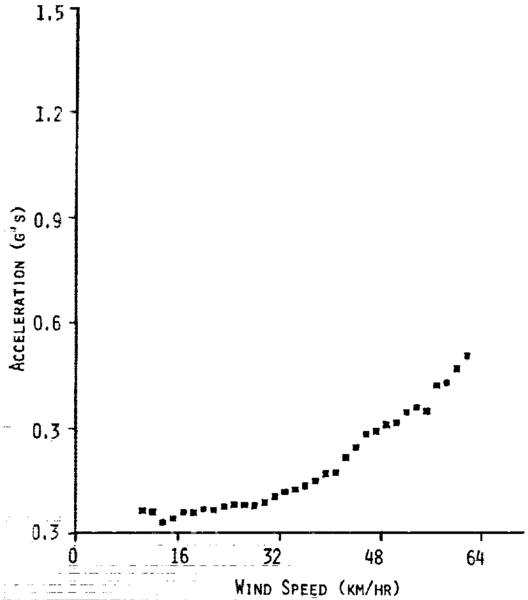

Figure 8 - Cable Horizontal Acceleration Versus Wind Speed for Tension 3, Undamped

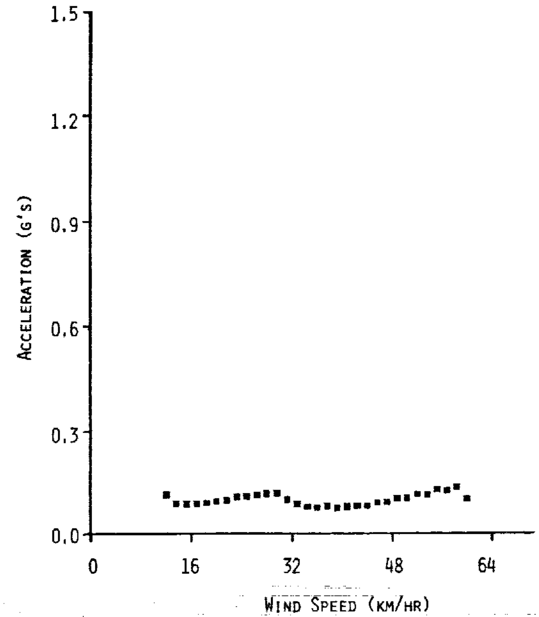

Figure 10 - Cable Vertical Acceleration Versus Wind Speed for Tension 1, Undamped

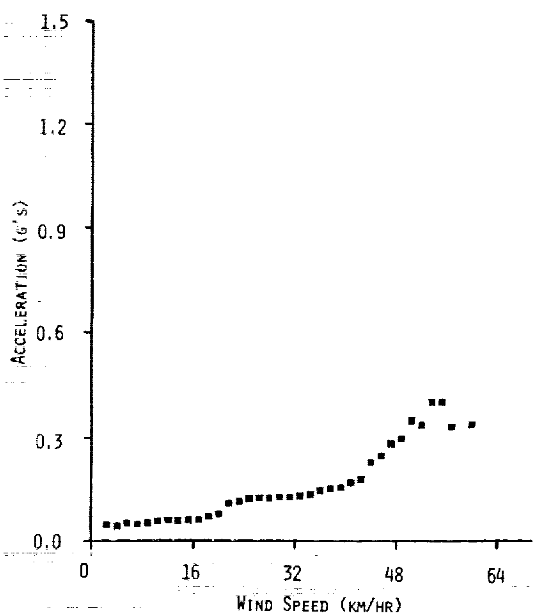

Figure 9 - Cable Horizontal Acceleration Versus Wind Speed for Tension 3, Damped

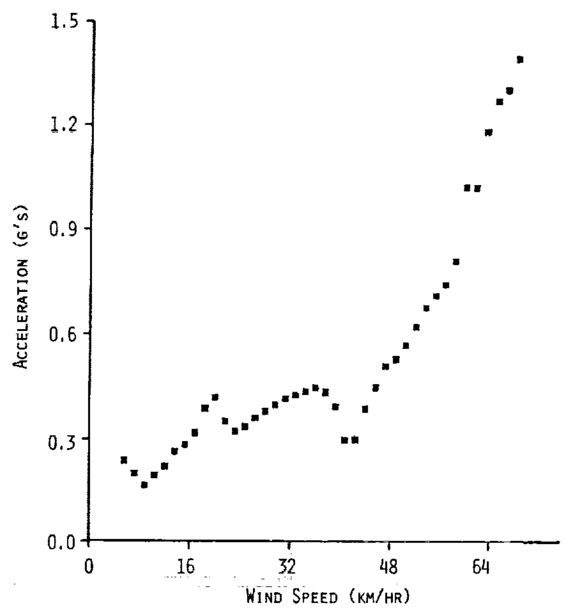

Figure 11 - Cable Vertical Acceleration Versus Wind Speed for Tension 2, Undamped

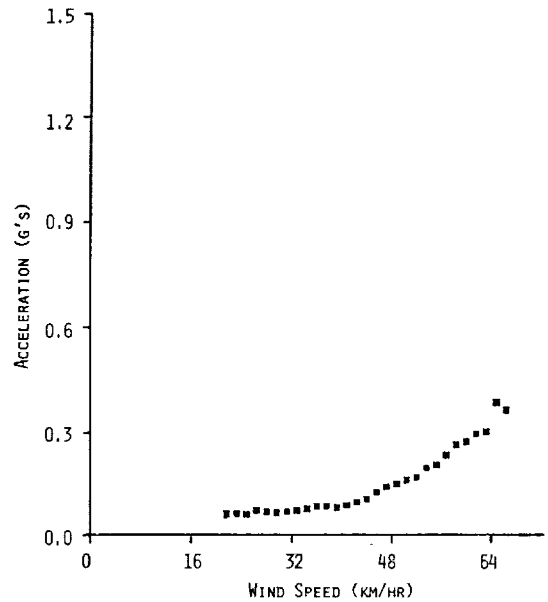

Figure 12 - Cable Vertical Acceleration versus Wind Speed for Tension 2, Damped

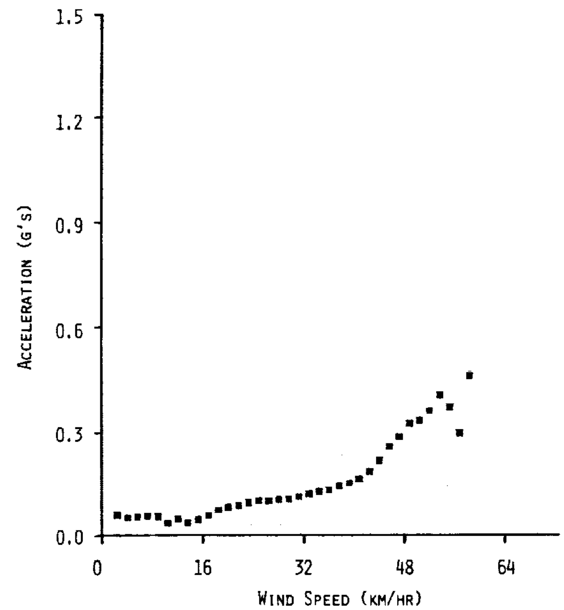

Figure 14 - Cable Vertical Acceleration Versus Wind Speed for Tension 3, Damped

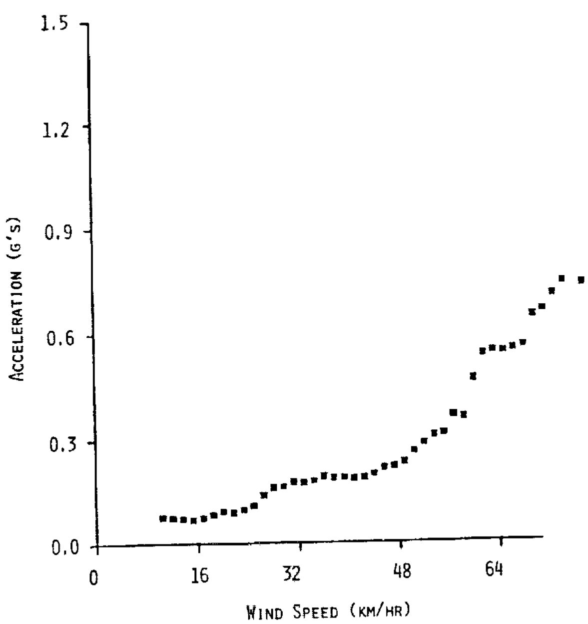

Figure 13 - Cable Vertical Acceleration Versus Wind Speed for Tension 3, Undamped

CONCLUSIONS

The design of the guy cables is not a straightforward process. It must be iterative since there are multiple requirements and constraints, and it must be interactive with the rest of the structural design since the cable stiffness affects the modal frequencies.

The cable damping concept demonstrated here appears to work quite well in reducing resonant vibrations of the cables. The system appears simple to design and install. Consequently, this system or something similar can be used to eliminate the constraint of cable vibrations on the design of the guys.

ACKNOWLEDGEMENTS

G. M. McNerney collected and processed all of the acceleration data for the Bins plots, and his contributions were essential to this work. R. O. Nellums' discussions with me were most helpful and are acknowledged.

REFERENCES

1. Reuter, R. C., Jr., Tie-Down Cable Selection and Initial Tensioning for the Sandia 17-Meter Vertical Axis Wind Turbine, SAND 76-0616 (Albuquerque, NM: Sandia National Laboratories, 1977).

2. Reuter, R. C., Jr., Vertical Axis Wind Turbine Tie-Down Design with an Example, SAND77-1919 (Albuquerque, NM: Sandia National Laboratories, 1977).

3. Carne, T. G., "Guy Cable and Foundation Design Techniques," SAND80-0984, Proceedings of the Vertical Axis Wind Turbine (VAWT) Design Technology Seminar for Industry, (Albuquerque, NM: Sandia National Laboratories, 1980).

4. Auld, H. E. and Lodde, P. F., A Study of Foundation/Anchor Requirements for Prototype Vertical-Axis Wind Turbines, SAND78-7046 (Albuquerque, NM: Sandia National Laboratories, 1979).

5. Lodde, P. F., Wind Turbine Foundation Parameter Study, SAND80-7015 (Albuquerque, NM: Sandia National Laboratories, 1980).

6. Nellums, R. O., "Economic Assessment of the Darrieus Wind Turbine," AIAA Paper 80-0614, Proceedings of the AIAA/SERI Wind Energy Conference, April 9-11, 1980, Boulder. Colorado.

7. Sullivan, W. N., "5-Meter Turbine Field Testing," SAND76-5586, Proceedings of the Vertical Axis Wind Turbine Technology Workshop, May, 1976 (Albuquerque, NM: Sandia National Laboratories, 1976).

QUESTIONS AND ANSWERS

T.G. Carne

From: D.J. Malcolm

Q: Why the odd effect of cable stiffness on first Butterfly frequency (first decrease, then increase)?

A: *There are actually two Butterfly modal frequencies. One is positive and one is negative. We have only shown the positive half of the frequency plane because the negative half is a mirror image. The cable stiffness for which the Butterfly mode has a zero frequency is the point where the frequencies are changing their signs. This feature is due to the fact that we are observing the modes in the rotating frame and is caused by the centrifugal softening and the coriollis effects.*

From: M.S. Chappell

Q: Will the dampers change characteristics with time, wear, weather conditions?

A: *We have seen the coefficient of friction change by ±50% for extreme weather conditions, but that magnitude of change does not alter the modal damping very much.*

From: Anonymous

Q: What is the maximum damping coefficient that you have measured experimentally? How do you include the effect of the weight (mass, used for friction) in your model?

A: *We have measured a five percent damping coefficient with the damping 4m from the end of a 40m cable. The mass can be included in the force equilibrium condition at the connection point.*

From: G.R. Frederick

Q: 1) Is cable stiffness due to cable only or some combination of cable and soil anchor stiffness?

2) What is the factor of safety against cable breakage?

3) What cable pretension to you use? Why cables instead of rods?

A: 1) *It is due only to the cable. but both the elastic deformation and changing the sag (see Ref. 3).*

2) *One should use at least a factor of safety of three on the highest tensions imposed on the cables, more if the turbine is in a corrosive environment.*

3) *The pretension is one of the cable parameters which must be determined from the stiffness requirement, and considerations of the axial load, sag, and cable frequencies. Cables have a high breaking strength and are much easier to work with.*

From: Anonymous

Q: How do you include time related (creep) changes in cable tension?

A: *By occasionally checking the tension in the cables and increasing it, if necessary, however, we have seen very little, if any, creep in the cables after installation.*