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A REVIEW OF RESONANCE RESPONSE IN LARGE, HORIZONTAL-AXIS WIND TURBINES

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ABSTRACT

Field operation of the Mod-0 and Mod-1 wind turbines has provided valuable information concerning resonance response in large, two-bladed, horizontal axis wind turbines. Operational experience has shown that 1 per rev excitation exists in the drive train, high aerodynamic damping prevents resonance response of the blade flatwise modes and teetering the hub substantially reduces the chordwise blade response to odd harmonic excitation. These results can be used by the designer as a guide to system frequency placement. In addition it has been found that present analytical techniques can accurately predict wind turbine natural frequencies.

INTRODUCTION

In any mechanical system, vibrations are undesirable and possibly dangerous if the vibratory motion becomes excessive. However, in a rotating system, like a wind turbine, vibrations are unavoidable. Hence, one of the keys to good design of wind turbines is to minimize vibrations. This is done by avoiding resonance.

Resonance is a phenomenon which occurs in a structure when an exciting or forcing frequency equals or nearly equals one of the natural frequencies of the system. It is characterized by a large increase in displacements and internal loads. In a rotating system the exciting frequencies are integer multiples of the rotational speed. In a wind turbine the important natural frequencies that must be considered are those associated with the blades, the tower (including yaw drive) and the drive train.

The purpose of this report is to review what has been learned about resonance response from operation of the Mod-0 and Mod-1 wind turbines. Based on this field experience, conclusions will be made concerning when resonance will occur in a two-bladed, horizontal-axis wind turbine. In addition analysis and design approaches to avoid resonance will be discussed.

ANALYTICAL APPROACH TO RESONANCE

A useful tool in the analysis of resonance potential of complex rotating systems is the frequency plot or Campbell diagram. A Campbell diagram for a hypothetical wind turbine, typical of wind turbines like the Mod-0 and Mod-1, is shown in Figure 1. In this diagram system natural frequencies are plotted versus rotational frequency. The radial lines are plots of integer multiples of the rotational frequency and represent exciting frequencies. Therefore, a potential for resonance exists whenever lines cross. This type of diagram is particularly useful for analyzing wind turbines designed to operate at variable rotational speeds.

System natural frequencies like those shown in Figure 1 can be calculated using finite element computer codes such as NASTRAN. Modeling considerations and natural frequency results for the Mod-0 tower, bed plate and blades are presented in reference 1 and for the Mod-0 drive train in reference 2. Experimental methods using vibration analyzers (ref. 3) are available for verifying calculations. Agreement between measured and calculated frequencies within 5 percent are not uncommon for the lower frequency modes that are of primary interest in wind turbines.

For a wind turbine with a single operating speed, another type of frequency diagram can be constructed and this is illustrated in Figure 2. This diagram was presented at the Mod-2 preliminary design review. The radial "per rev" lines of Figure 1 are now vertical lines with an avoid range applied. The required width of the avoid ranges is not precisely known. Here they were conservatively set at ±0.5 per rev for the even harmonics and ±0.25 per rev for the odd harmonics. Calculated system frequencies are also shown. All the system frequencies are clear of the avoid ranges except the off-line drive train frequency. Because the rotor is a node point for this mode, the rotor cannot excite it. Therefore it is permissible to be within the avoid range for this case.

The avoid ranges shown in Figure 2 are relatively wide, resulting in rather small windows for the designer to place system frequencies. Another point to note in this figure is that for the drive train no avoid range is shown for the odd harmonics. The significance of this will be discussed in the following section.

A similar diagram for the Mod-1 wind turbine for 35 rpm operation is shown in Figure 3(a). Here a slightly different approach to the avoid ranges was taken. The avoid ranges are not as wide, allowing the designer greater flexibility in frequency placement. The justification for the very narrow avoid range for the flatwise blade mode is the high aerodynamic damping associated with this mode. Only odd harmonic avoid ranges are shown for the blade chordwise mode because no even harmonic excitation is expected. The theoretical basis for this can be found in reference 4. The blade torsion/pitch change mechanism (PCM) mode requires high stiffness to avoid flutter. The natural frequency of this mode should be designed to be so high that resonance is not a consideration.

Measured and calculated system frequencies are compared in Figure 3(a), showing the degree of correlation which can be expected. The calculated blade frequencies include the effect of centrifugal stiffening. The measured blade frequencies were obtained by applying an analytically derived correction factor to the measured non-rotating frequencies. In general the agreement between measured and calculated values is good, which is typical of experience to date. When differences do occur, they can usually be explained by modeling assumptions that were not realized in the as-built wind turbine. For instance, the difference in the off-line drive train frequencies comes from the assumption of no backlash in the drive train. The difference in the tower bending frequencies is related in part to the assumption of a flexible foundation, when the actual wind turbine was attached to bed rock.

A condensed version of Figure 3(a) is shown in Figure 3(b) for the present operating speed of the Mod-1, 23 rpm. The measured natural frequencies are clear of the avoid ranges except for the blade flatwise mode. Because of high aerodynamic damping, this is no concern. Note that the calculated drive train mode now is very close to 1 per rev. The implications of this are discussed in the following section.

FIELD EXPERIENCE WITH RESONANCE

Because it is a research machine the Mod-0 wind turbine has provided much experience with resonance. The Mod-1 has provided additional experience. In this section this experience will be reviewed for the drive train, rotor support (tower, yaw drive) and blades.

Drive Train

For a perfectly balanced rotor, only even harmonic excitation would be expected in the drive train of a two bladed rotor. For this reason avoid ranges in Figures 2 and 3 are shown only on the even harmonics. However, mass, stiffness or aerodynamic unbalance from blade to blade can cause odd harmonic excitation as well. This was vividly demonstrated when the Mod-1 rotor speed was reduced from 35 to 23 rpm which placed its natural frequency close to 1 per rev. The output power of a wind turbine can vary about a mean power level. Figure 4 compares the amplitudes of cyclic power variation for 23 and 35 rpm operations. Examination of the real time data for 23 rpm operation showed the cyclic frequency was close to 1 per rev. The source of the 1 per rev excitation in the drive train is presently being investigated. However, the response seen here is not peculiar to the Mod-1. Synchronous operation of the Mod-0 at 35 rpm placed its drive train natural frequency close to 1 per rev. A time history of output power is shown in Figure 5. The response here is very similar to that seen with the Mod-1.

The field experience described above provides strong evidence that wind turbine drive trains contain 1 per rev excitation. In addition, Mod-0 drive train resonance caused by even harmonic excitation is documented in reference 2. Therefore, drive train natural frequencies must avoid both even and odd harmonics until the source of the 1 per rev excitation is better understood and reduced or, hopefully, eliminated.

Rotor Support Structure

In its initial configuration the Mod-0 had a relatively flexible yaw drive. The natural frequency of the nacelle and rotor about the yaw drive was close to 2/rev at the operational speed of 40 rpm. This resonance resulted in large yaw oscillations which in turn resulted in an amplification of blade loads. This amplification was of major concern because, if allowed to continue, it would significantly decrease blade life. Modifications were, therefore, made to Mod-0 to increase its yaw stiffness. Figure 6 shows cyclic blade loads as a function of yaw drive stiffness. The measured data are compared to loads calculated using the MOSTAS-B computer code (ref. 5). The near 2 per rev resonance in the yaw drive caused both edgewise and flatwise blade load amplification. This amplification has been reduced by either softening the yaw drive (free yaw) or by stiffening it by means of a dual yaw drive (ref. 6) and by locking the bed plate to the tower with brakes.

Because it is a research machine, the Mod-0 has been modified in several ways primarily to support advanced wind turbine projects like the Mod-2. Among these modifications are reduction of the tower natural frequency by placing it on a spring fixture (ref. 7) and replacement of the rigid hub with a teetered hub (ref. 8). The effect of these modifications on resonance response will be described next.

The original Mod-0 tower had a first bending frequency of about 2.1 Hz and was characterized as being "stiff" because this frequency was greater than the blade passing frequency of 2/rev when operating at 40 rpm. Because of interest in more flexible towers, primarily for economic reasons, the frequency of the Mod-0 truss tower was reduced by placing the tower on a spring fixture. The details of this fixture are given in reference 7. With the fixture the tower bending frequency was reduced to about 0.8 Hz or 1.5/rev at the present operational speed of 31 rpm. Towers with bending frequencies between 1/rev and 2/rev are classified as "firm" towers. One of the operational characteristics with firm towers is that the machine passes through a 2/rev tower resonance going up to and coming down from operation speed. The effect of this resonance on blade and tower loads was a concern.

Figure 7 shows the envelope of peaks from a time history of the response of the Mod-0 with a firm tower during a typical start-up, shut-down cycle. In summary, this test showed that when passing through the tower resonance, blade loads were not adversely affected and tower deflections were not excessive. More details of this test can be found in reference 9. Initial operation of the first Mod-2 wind turbine at Goldendale, Washington, confirms the results of the Mod-0 firm tower tests.

Blades

Blade resonance response can be determined by running a speed survey in which the wind turbine is operated over a wide range of rotor speeds. The results of a speed survey between 12 and 40 rpm for Mod-0 in the stiff tower configuration are shown in Figure 8. In general the chordwise loads respond to the odd harmonics as expected. Peak response was expected to occur very close to the odd per revs. However, the 5 per rev peak response occurs about 2 rpm higher than expected. The reason for this displaced response is not presently understood. The response to harmonics greater than 5/rev does not appear to be significant. Examination of flatwise blade loads during this same speed survey showed no load amplification at any rotor speed. High aerodynamic damping prevents the blades from responding in the flatwise direction.

The primary reasons for teetering a rotor is to reduce flatwise blade loads and torsional tower loads. As a result hub motions are reduced, which in turn should reduce the blade resonance response. This indeed is the case as shown in Figure 8 for a 5/rev resonance. The load amplification for the teetered hub was about half that for the rigid hub.

The data shown in Figure 8 have certain implications with respect to multispeed operation of a wind turbine. Operation on 5/rev and above for a teetered rotor and 7/rev and above for a rigid rotor could probably be tolerated. This is based on the fact that the load magnification for these resonances is less than 20 percent.

CONCLUSIONS

Based on the operation of the Mod-0 and Mod-1 wind turbines, the following conclusions concerning resonance response in large, two-bladed, horizontal axis wind turbines can be made:

1. The important natural frequencies of wind turbines can be calculated with reasonable accuracy.

2. Odd harmonic content is present in the drive train and can cause significant resonance response.

3. Resonance associated with yaw drive flexibility causes blade load amplification; resonance associated with tower bending flexibility does not cause blade load amplification.

4. Odd harmonic excitations up to and including 5/rev can cause significant blade chordwise resonance response; teetering the rotor will reduce this response substantially.

5. High aerodynamic damping prevents resonance response in the blade flatwise direction at all frequencies.

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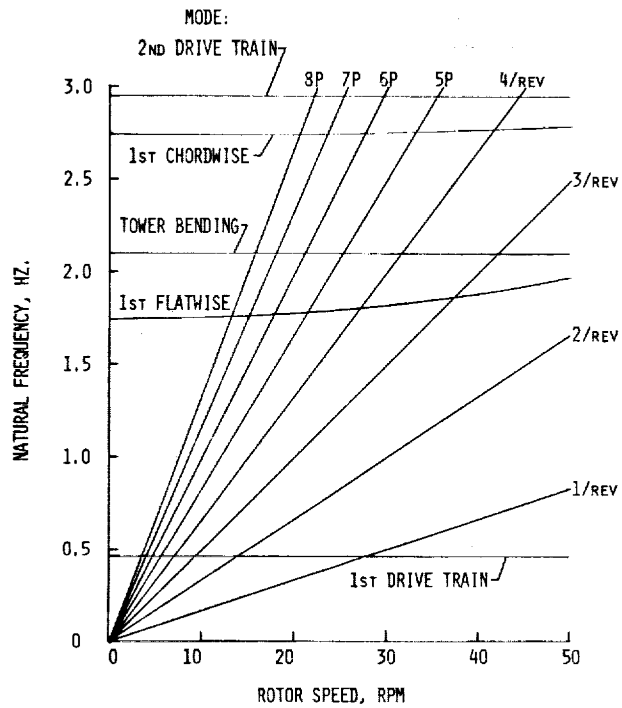


Figure 1 - Campbell diagram for a hypothetical wind turbine.

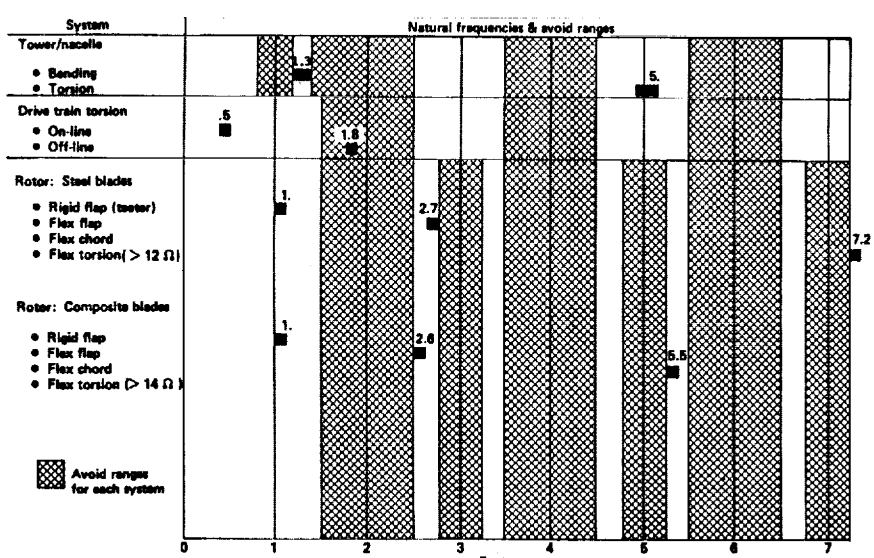
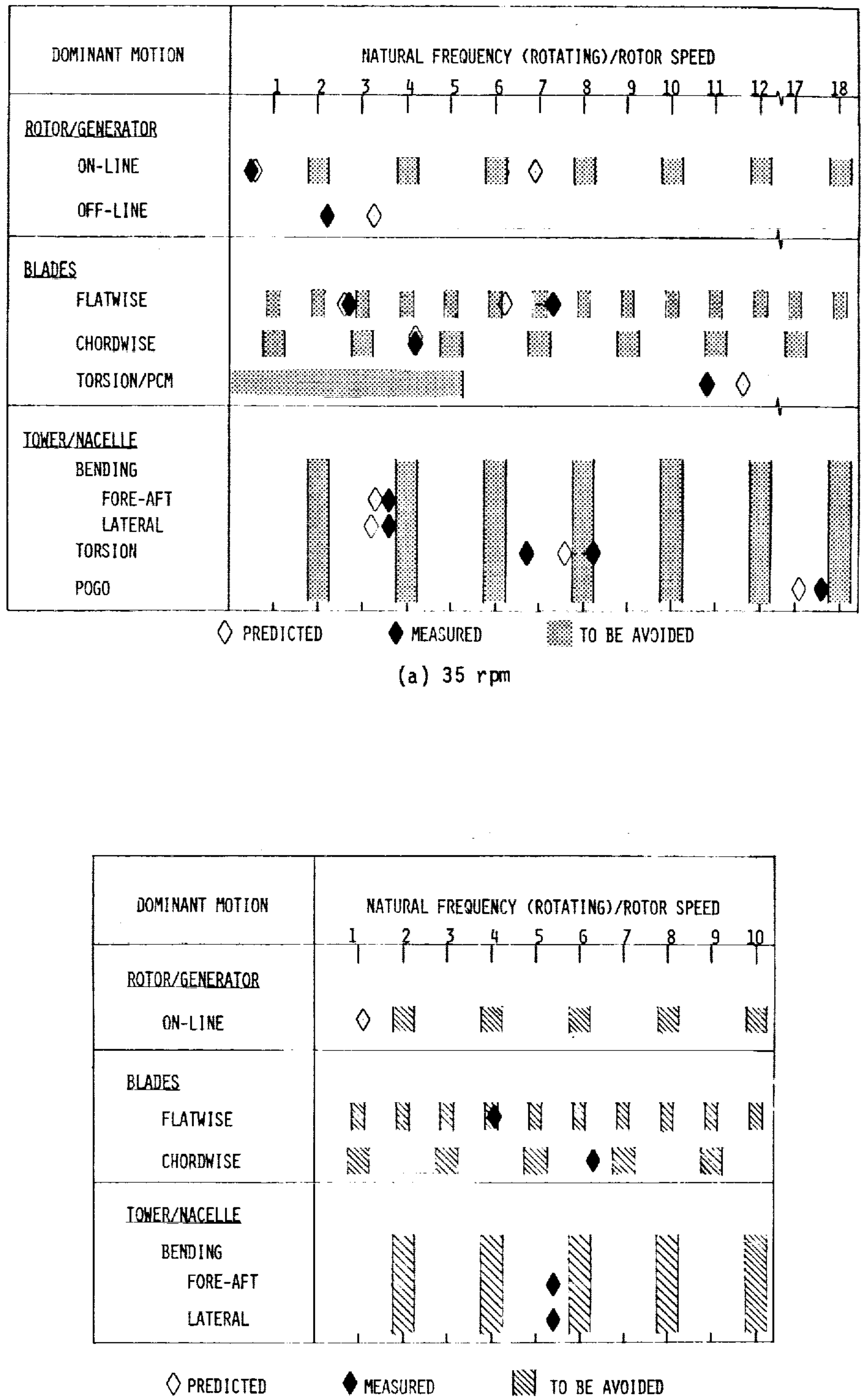


Figure 2 - System resonance avoidance -- the Mod-2 approach.



(a) 35 rpm

(b) 23 rpm

Figure 3 - Mod-1 system natural frequencies.

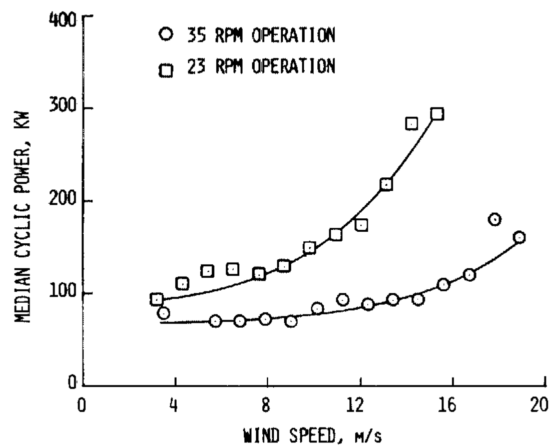


Figure 4 - Effect of drive train resonance on the amplitude of alternator power fluctions in the Mod-1.

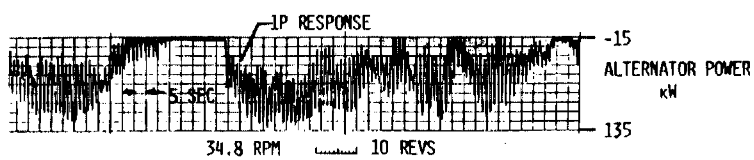


Figure 5 - Time history of a 1/rev drive train resonance in the Mod-0.

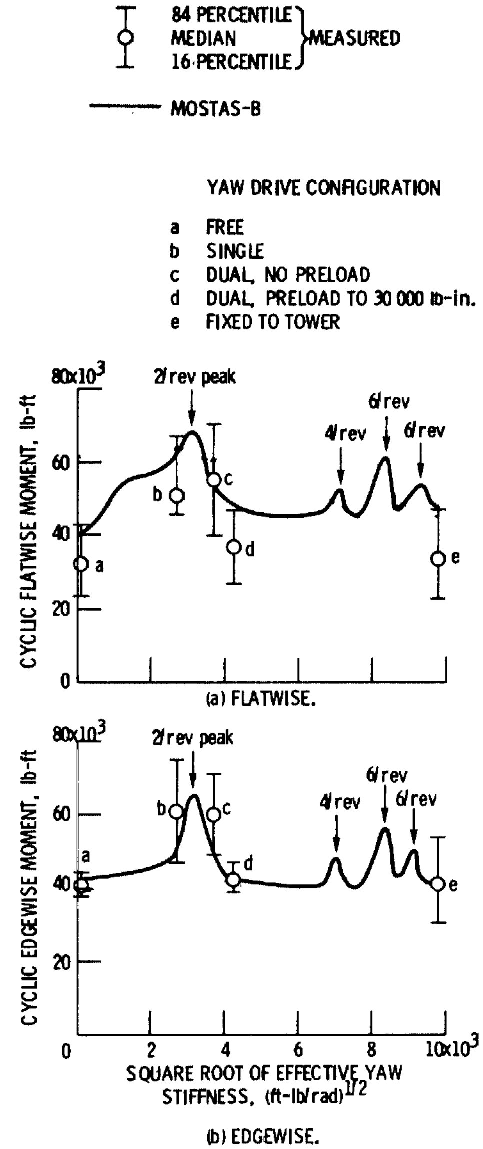


Figure 6 - Blade response to a yaw drive resonance in the Mod-0 (5% span).

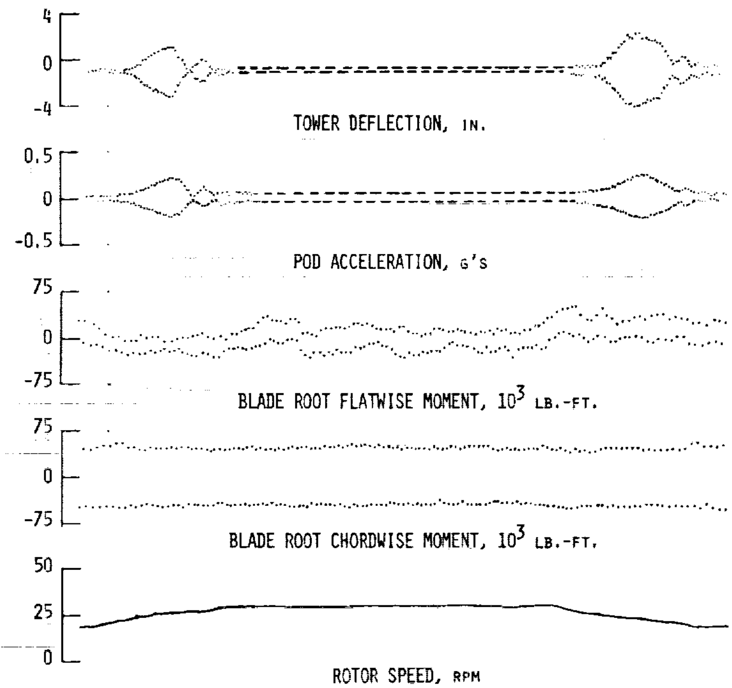


Figure 7 - Response of the Mod-0 when passing through a 2/rev tower resonance (envelope of cyclic peaks).

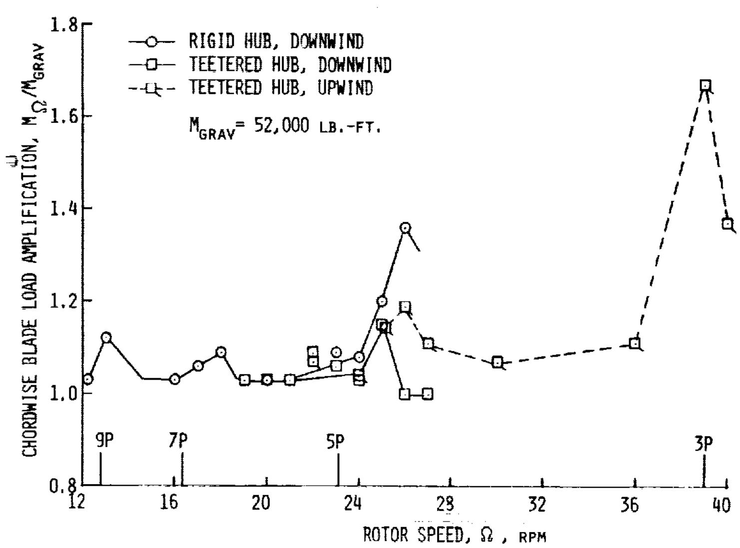


Figure 8 - Blade root chordwise response as a function of rotor speed for the Mod-0 tip-controlled blade.

QUESTIONS AND ANSWERS

T.L. Sullivan

From: K. Hohenemser

Q: Why did you not show the chordwise 3P for the rigid hub?

A: *We were afraid of damaging the blades if we ran the rigid hub up to 3P (40 rpm).*

From: W.C. Walton

Q: 1) Do blade dynamic characteristics (stiffness; natural frequencies) play any role in the 1P drive train resonance (i.e., do the blades just provide the rotational inertia)?

2) In the blooming of tower response (with no accompanying increase in blade loads) were the tower loads of any concern?

A: *1) For the fundamental drive train mode (which is the one of most concern) only the rotor inertia plays a role in the drive train natural frequency.*

2) Tower loads were not of concern. However, the loads in the spring fixture used to soften the tower were of concern. Hence, the spring fixture was strain gaged. These strains were within acceptable limits.

From: Anonymous

Q: What is the predominant frequency of the cyclic component of MOD-1 power output (at 23 rpm rotor speed)?

A: *Close to 1/rev.*

From: R. Perley

Q: What is the basis of establishing the avoidance ranges?

A: *Presently they are set rather arbitrarily. Heavily damped modes have narrow bands. Lightly damped modes have wider bands. Additional field experience and computer studies will allow us to set the avoid ranges more precisely.*

From: W. Sullivan

Q: What methods were used to compute MOD-1 and 2 resonant frequencies--are all rotating coordinate system effects included?

A: *The only rotational effect taken into account was centrifugal stiffening. This approach gives good agreement with measured data.*