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A RATIONAL METHOD FOR OPTIMIZING SHROUD DAMPING

BREAK

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ABSTRACT

A computer code, BDAMPER, was recently developed by Ohio State University researchers and can be used to predict the vibratory response of shrouded blades that contain friction interfaces. This paper discusses some modeling issues that arose in applying BDAMPER to actual blades and outlines a procedure for optimizing the shroud's design in order to minimize the blade's resonant response. A comparisons are made between BDAMPER predictions of blade response and experimental data taken from spin pit and engine tests.

1. INTRODUCTION

BDAMPER is a computer code that calculates the steady state vibratory response of blades with friction constraints, either shrouds or platform dampers. It was recently developed by Professor Chia-Hsiang Menq and his colleagues at Ohio State University. A discussion of the physical model and theoretical basis of BDAMPER are provided in the companion paper by Yang and Menq (1996). A general literature review of related research is given by Griffin (1990). The research upon which BDAMPER's development is based was funded by the GUIde Consortium of U.S. Engine Manufacturers, Richardson and Griffin (1995). In the authors' opinion, BDAMPER is a major step in the development of a rational method for optimizing shrouds. However, its successful application to actual hardware raises a number of practical issues that have not been addressed. For example, the first part of this paper discusses how to account for mistuning effects and how to estimate the magnitude of the forcing function in the test environment — an important issue since the system is nonlinear. It also identifies a practical procedure for systematically optimizing the design of these complex structures. The second part of this paper presents the first published comparison of predictions from BDAMPER with test data taken from full bladed disks.

2. REPRESENTATION OF RESULTS

Shrouded stages that contain friction interfaces respond dynamically in a nonlinear manner. This complicates both the blade's vibratory analysis and the shroud's design. This section discusses effective methods of representing their behavior that can provide a basis for optimizing their design.

BDAMPER can be used to calculate a family of *tracking plots* for different levels of excitation. Tracking plots are plots of the amplitude of the blade as a function of the excitation frequency, Figure 1. The excitation level is specified in terms of the magnitude of the generalized (modal) force that acts on the blade. The tracking plot data, in turn, can be used to generate performance, frequency, and apparent damping curves. Each of these types of curves will be described in this section.

2.1 <u>Tracking Plots</u>

Consider the tracking plots depicted in Figure 1 in which the amplitude of response is plotted as a function of engine rotational speed for a range of excitation levels. A comment is needed to clarify how this figure was generated. BDAMPER calculates the amplitude of response as a function of the excitation frequency for a specific engine order excitation, E. Thus, in a limited frequency range the engine speed or rpm is proportional to the excitation and may be calculated as

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$rpm = 60\omega/2\pi E$, where ω is the excitation frequency in radians per second. The frequency range is limited because the blade receptances used in the calculation are determined from the blade's modes calculated at a fixed operating speed.

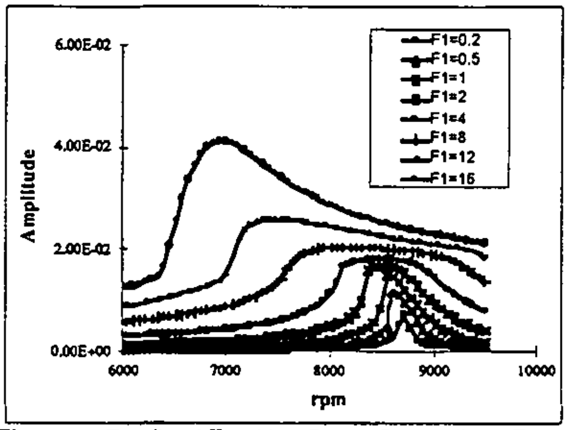

Figure 1: Family of Tracking Plots: Generalized Force Increased From 0.2 to 16.0.

The nonlinear influence of the shroud can be seen in the tracking plots. At low amplitudes the shroud is stuck and the blade is more constrained. As a result, for small levels of excitation the frequency of peak response tends to be higher than for large levels of excitation. Note also that because of friction, the peak amplitude increases only by 70% even though the magnitude of the generalized excitation force increases from one to twelve. It is clear that the shroud is a very effective nonlinear damper in this range of response.

2.2 Shroud Performance Curves

The information on the peak response contained in the tracking plots can be summarized in one of two types of shroud performance curves. In the first type, a *force performance curve*, the peak displacement or stress is plotted as a function of the generalized force, Figure 2. The advantage of the force performance curve is that it provides a robust measure of the shroud's performance that is relatively independent of the amount of baseline damping in the system. This robustness is apparent in Figure 2 in that the shrouded blade is predicted to perform in a similar manner even though the baseline damping ratio is increased by a factor of five.

Force performance curves are useful in optimizing the design of the shroud. One design parameter that can be optimized is the static pre-load or normal load that acts on the shroud interface. A careful examination of the governing equations indicates that when a Coulomb model of friction is used then a unique normalized performance curve may be derived by dividing both the amplitude of the response and the generalized force by the normal load. This approach was originally developed to optimize the normal load of under platform friction dampers by Cameron, et al. (1990). Thus, even though the system is nonlinear, the normalized force performance curve needs to be calculated only one time and the response of the system for other normal loads can be readily determined by scaling the axes.

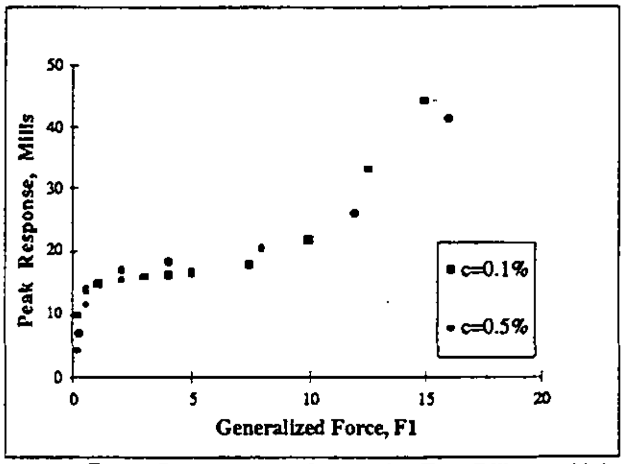

Figure 2: Force Performance Curves for Two Different Values of Baseline Damping

When performing a shroud optimization study on a new blade, the magnitude of the excitation is usually not known and the goal is to have a system that can handle as large a force as possible and not exceed some allowable level of response. From the previous discussion it is clear that the designer needs to choose the shroud normal load so as to set the flat portion of the performance curve just below the allowable stress for the material (with an appropriate safety factor). Alternatively, changing the shroud angle or the radial position of the shroud on the blade changes the shape of the performance curve. Thus, for a particular mode of response the shroud angle and radial position on the blade need to be chosen so as to insure efficient damping by the shroud in that mode, i.e., a relatively broad flat portion of the performance curve. Then, changing the normal load on the shroud sets the level at which it will efficiently damp the vibratory response.

As mentioned, the advantage of the force performance curve is that it is relatively independent of the amount of baseline damping in the system. The disadvantages are that it is difficult for engineers to have a feel for the generalized force term that is plotted on the horizontal axis and it is not clear how to compare the efficiency of the shroud in damping different modes. One way of dealing with this is to use *displacement performance curves* in which the damped amplitude is plotted as a function of the undamped amplitude, i.e., the value that the resonant amplitude would be if the shroud interface did not slip.

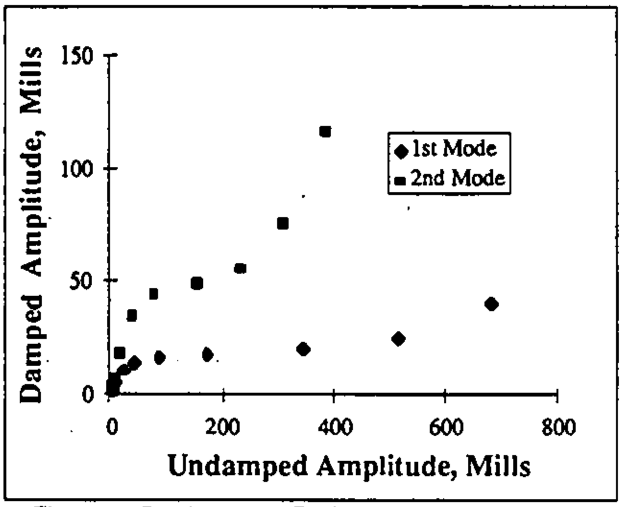

Figure 3: Displacement Performance Curves Allows Comparison of Different Modes

In order to construct a displacement performance curve the generalized force needs to be converted to undamped amplitude. This is done by running BDAMPER with a low level of excitation, an extremely high normal load, and small frequency increments in order to calculate the peak linear response of the blade without shroud slip. Since this is a linear calculation the ratio of the undamped amplitude to generalized force remains the same for all subsequent calculations. An example of a displacement performance curve is shown in Figure 3. The main disadvantage of this type of plot is that the scale of the horizontal axis depends on the amount of baseline damping used in the calculation. In this example the baseline damping was 0.5% of critical for the analyses of both the 1st and 2nd modes. It is clear that the shroud is more efficient in damping the vibratory response of the first mode than it is in damping the response of the second mode. Physically, this is because the shroud interface is more closely aligned with the direction of motion of the blade in its first mode.

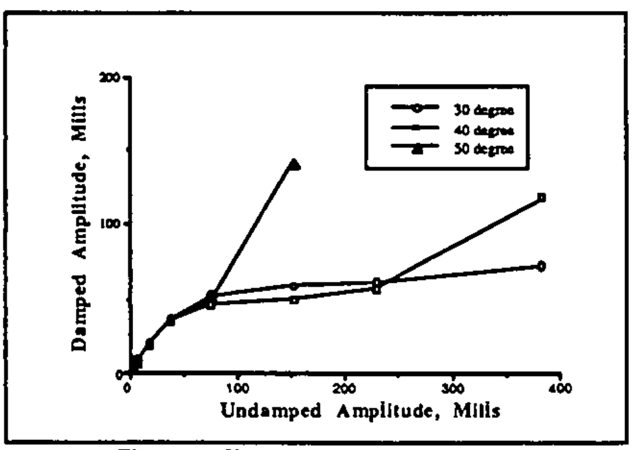

Figure 4: The Effect of Shroud Angle

Either type of performance curve can be used to optimize the contact angle of the shroud. For example, the effect of changing the shroud angle on the response of the second mode was predicted using BDAMPER and is shown in Figure 4. Clearly, in this example, lowering the angle was beneficial in that it made shroud damping more effective at high levels of excitation and broadened the performance curve.

2.3 Frequency and Damping Plots

Two additional types of curves can be used to summarize the response of the shrouded blade. From the tracking plots, information can be extracted about the frequencies over which the blade strongly responds. An example of the type of frequency plot that can be generated is shown in Figure 5.

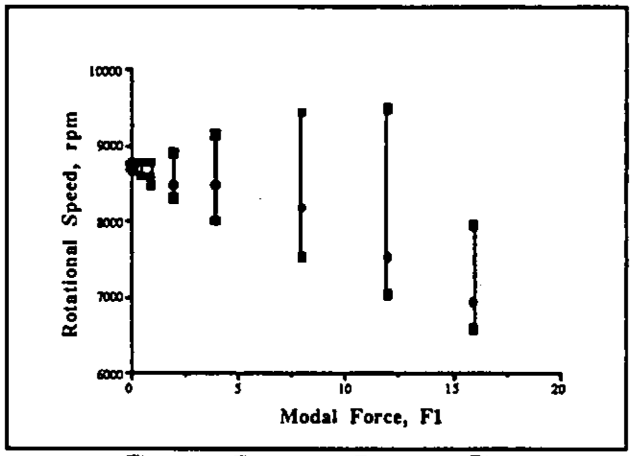

Figure 5: Frequency Response Plot

The single dark data point in the frequency response plot is the frequency at which the response was highest for a given value of generalized force. The vertical line indicates the range of frequencies over which the response was above a fixed percentage (90% in the case of Figure 5) of its maximum value. Thus, as the excitation increases the shroud slips during a larger portion of the time and causes the peak response frequency to drop and the shape of the peak to widen. In fact, if this were a linear system then the engineer would think that the amount of damping in the system was increasing because the "half bandwidth" damping increases. Clearly, the tracking plot data also can be used to generate a plot of "apparent" half bandwidth damping as a function of the generalized force. An example of a representative result is shown in Figure 6. This type of plot will prove useful for estimating the magnitude of the generalized force that was present in spin pit tests.

In summary, a sequence of tracking plots provide the fundamental data required to understand and optimize the dynamic response of shrouded blades. The presentation of the data may be simplified by the use of performance, frequency, and apparent damping curves.

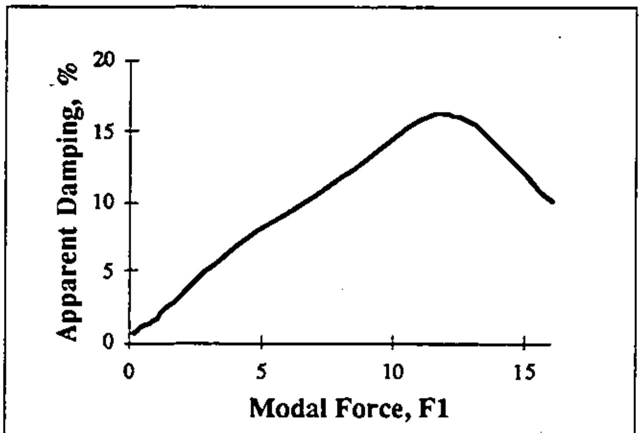

Figure 6: Apparent Damping as a Function of Generalized Force

3. COMPARISONS WITH TEST DATA

In this section BDAMPER will be used to compute the resonant response of two representative fan blades. In the first instance, the blade was extensively tested in a spin pit in order to better understand its vibratory response. The resulting data provide a broad test matrix for evaluating BDAMPER since the response in two modes were measured at different levels of excitation and at different shroud normal loads.

3.1 Simplifications in Correlating with Data

Three issues had to be addressed in order to correlate the spin pit test data with predictions using BDAMPER. These were:

1. What is the effective generalized force in the spin pit tests?
2. How to plot data from tests on stages with different shroud tightnesses?
3. How to account for mistuning, i.e., the fact that individual blades on the stage had significantly different peak amplitudes?

3.1.1 Estimating the Effective Generalized Force in Experiments. The apparent damping of the average blade's response was calculated from the spin pit test data for three different air jet excitations using the half bandwidth method. It was assumed that the air jet pressure was proportional to the generalized force acting on the blade. A proportionality constant was determined that best correlated the measured damping with the apparent damping predicted by BDAMPER. For example, the assumption that the generalized force in the first mode, F_1, is given by

$$F_1 = 0.012P \tag{1}$$

where P is the air jet pressure results in the correlation between predicted and measured damping as indicated in Figure 7.

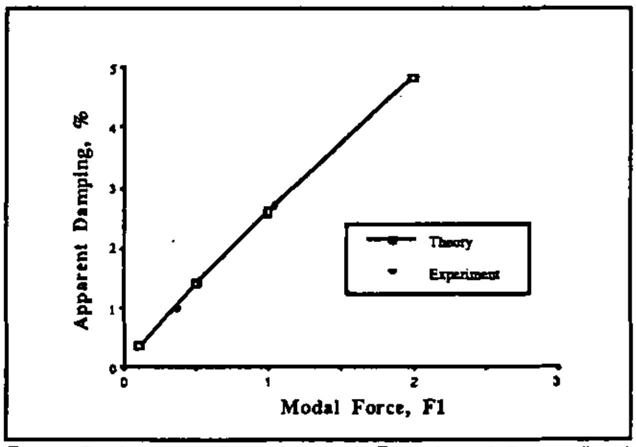

Figure 7: Apparent Damping as a Function of Generalized Force

A different generalized force and pressure relationship was developed for the second mode using the same method. Thus, the generalized force is estimated from the shape of the frequency response plots and not their magnitude. Consequently, the test of the theory is how well BDAMPER predicts the amplitude of response.

3.1.2 Accounting for Different Shroud Tightnesses. In this study, analytical results from BDAMPER are compared with spin pit tests on different wheels that have different average shroud normal loads. The average normal load in the shrouds was determined by careful measurements of the dimensions of the blades and by finite element simulations of the effect of centrifugal loading on the blade's untwist. The shroud normal load was calculated to vary by a factor of two from the loosest to the tightest set of blades tested. The issue here is how to plot experimental data from the different tests on a single curve.

Section 2 discusses the fact that the response for friction controlled systems scale with normal load. This means that the peak amplitude of response, A, divided by the normal load, N, is a unique function of the magnitude of the generalized force, F, divided by N, i.e.,

$$\frac{A}{N} = f\left(\frac{F}{N}\right) \tag{2}$$

Let N_o be the nominal normal load used in analyzing the shrouded blade with BDAMPER and let $( )^*$ quantities correspond to experimental values of the parameters. Then for purposes of plotting the data on the nominal performance curve, the effective values of generalized force and amplitude are:

$$F_e = F \cdot \frac{N_o}{N^{\bullet}} \text{ and } A_e = A \cdot \frac{N_o}{N^{\bullet}} \qquad (3)$$

Thus, all of the effective values of experimental data can be plotted on single theoretical performance curve.

3.1.3 Correlating Predictions with Mistuned Disk Response. BDAMPER assumes that the bladed disk system responds as a "tuned stage", i.e., every blade has the same amplitude and differs in the phase of its response by the same amount as the phase shift in the excitation. This is clearly not true in real systems in which the peak amplitudes of the blades may vary by almost an order of magnitude. Thus, the issue is, how can a tuned system analysis be compared with real data from a mistuned bladed disk?

Two *ad hoc* assumptions were made to deal with this difficulty. They were: 1) that the tuned system response predicted by BDAMPER is the same as the average response of the system, and that 2) the highest responding blade on the disk is subjected to an effective excitation which is λ times higher than that acting on the average blade. In effect, the motions of the neighboring blades reinforce the aerodynamic excitation acting on the highest responding blade so that it experiences a greater generalized force.

Thus, each spin pit test will provide two data points for the correlation: the stage's average response at a generalized force determined from equation (1) and the response of the highest responding blade on the disk associated with a generalized force equal to λ times the baseline value. The test data were correlated with predictions from BDAMPER for λ equal to both two and four. Both correlations displayed similar behavior. The λ equal to four correlations were slightly better and they are the ones discussed in the next section.

3.2 Correlation With Spin Pit Tests

3.2.1 Correlation with First Mode Data. It was assumed that the coefficient of friction for titanium shrouds was equal to 0.4 and that the amount of baseline damping was 0.1% of critical. Spin pit tests were performed on three different bladed disks. The blades were referred to as: "tight shrouds", "medium shrouds", and "loose shrouds". The average normal loads for each stage was calculated from measured shroud dimensions and the values are summarized in Table 1. Test data were taken at three different air jet pressures. The air jet pressures were converted to values of generalized force, F1, using equation (1). The amplitude and generalized force data were scaled using the shroud normal loads and equations (2) and (3) to give effective values of resonant amplitude and generalized force for each test. The generalized force values were then converted to values of undamped amplitude using the procedure outlined in Section 2.

Table 1: Shroud Normal Loads at First Mode Crossing

Description	*Shroud Normal Load (lbf)*
Nominal	143
Tight Shrouds	138
Medium Shrouds	94
Loose Shrouds	66

The correlation with the prediction based on the blade's first mode of response is shown in Figure 8.

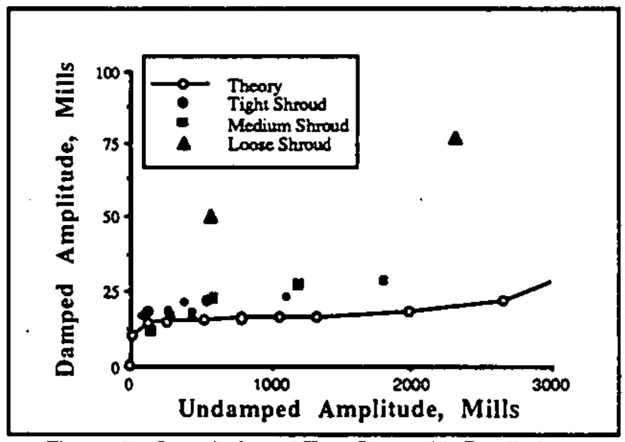

Figure 8: Correlation of Test Data with Predictions

In general the tight and medium shroud data agreed relatively well with the predicted displacement performance curves. The loose shroud data clearly did not agree with either set of predictions. Three possible reasons for this disagreement are:

1. At low values of shroud normal load, shroud misalignment may be more critical.
2. Variations in shroud normal loads will be a larger percentage of the average normal load. Thus, the more severe mistuning from shroud normal load variations may induce larger vibratory response.
3. The procedure used in estimating the generalized force (based on matching apparent damping) could be in error and the actual equivalent generalized force could be larger than estimated. Thus, the loose shroud data actually corresponds to the

nearly vertical, high response part of the performance curve. One possible reason for the error could lie in the implicit use of the assumption that BDAMPER predicts the average response of the stage.

Whatever the cause, an important result is that at low normal loads (60 lbf) BDAMPER under predicts the response of the first mode by about a factor of two. At higher normal loads (> 90 lbf) it provides a reasonable estimate of response amplitudes.

3.2.2 Correlation with Second Mode Data. For this blade, the engine speed at which the first mode and second mode crossings occur are sufficiently close that a single set of blade modes (calculated at a speed halfway between the first and second mode crossings) were used as input for both analyses. Thus, the only changes in the input for the blade and shroud models used in the first mode and second mode analyses were an increase in shroud normal load and a change in the engine order of the excitation from 4E to 8E. In addition, a new relationship had to be determined between air jet pressure and the generalized force exciting the blade in its second mode. The method of using apparent damping to estimate the generalized force acting on the stage did not work as well for the second mode as it did for the first mode in that the slopes did not agree as well, refer to Figures 7 and 9. (Note that because BDAMPER uses force inputs in terms of the modes of the blade with an unconstrained shroud, that the generalized force used to excite the second mode of the constrained blade is F_3.)

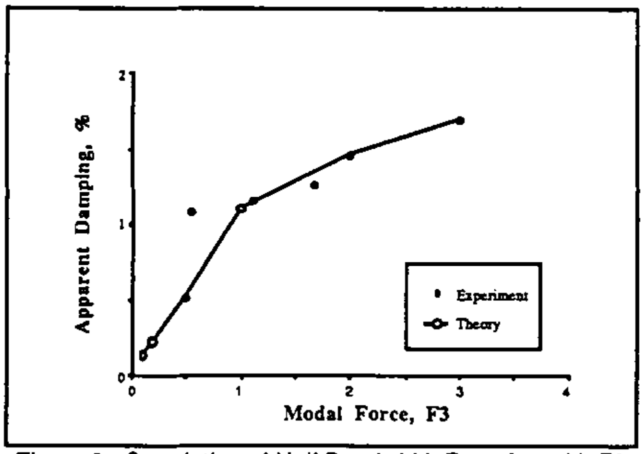

Figure 9: Correlation of Half Bandwidth Damping with F3

The correlation of BDAMPER predictions with the test data for the second mode is shown in Figure 10.

In general, the test data agreed reasonably well with the predictions for the blades second mode's response. Interestingly, BDAMPER even predicted the response of the stage with loose shrouds. One reason for this could be that at the higher rpm corresponding to the second mode, the "loose shroud" was relatively highly loaded, i.e., the normal load on the shroud at the 2nd mode crossing was calculated to be 141 lbf whereas it was only 66 lbf at the first mode crossing.

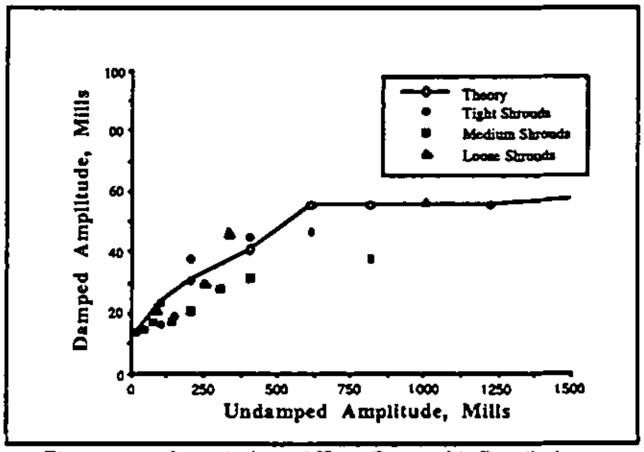

Figure 10: Correlation of Test Data with Predictions

It should be noted that since essentially the same models were used for both 1st and 2nd mode calculations that the first mode and second mode correlations should be viewed as independent tests of the same BDAMPER model and analysis. In this light one can conclude that the correlations between test data and predictions are quite good.

3.3 Correlations with Engine Tests

BDAMPER was also used to calculate the forced response of a different fan stage on which a limited amount of engine strain gage data was available. A summary of the correlation follows.

3.3.1 First Mode Response. In effect the procedure that was followed in section 3.2 was to infer the magnitude of the generalized force from the shape of the response plot (apparent damping) and then observe whether or not the amplitudes of the response correlated with BDAMPER. Consider the strain gage data taken from engine tests indicated by the solid triangles in Figure 11. The test data agrees with BDAMPER in that has a similar shape and magnitude to that predicted when the generalized force F_1 equals 500. However, the peak response occurred at an engine speed about 20% lower than that predicted by BDAMPER. One possible reason for the frequency error is that the blade finite element model that was used to calculate the input did not include the disk and it's flexibility. A second reason could be that in BDAMPER the flexibility of the shroud is accounted for in terms of the blade's receptance which is calculated by summing the modes of a blade with an unconstrained shroud. This means that a very large number of modes could be required to accurately represent shroud bending effects. If enough modes are not used then BDAMPER would tend to over predict the stiffness of the shroud and the stage's frequency would be too high. Further study is required to fully understand this issue.

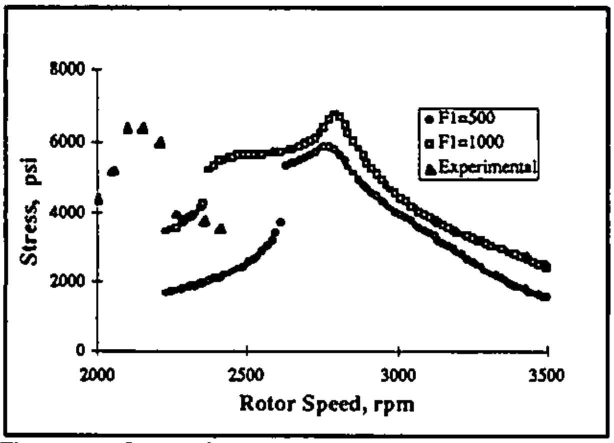

Figure 11: Comparison of Predicted Response with Strain Gage Data

3.3.2 <u>Second Mode Response</u>. The second mode on the commercial fan blade was of special interest because the history of the stage indicated that the response of the second mode seemed to be particularly sensitive to minor changes in the design. Consequently, a goal of the study was to try to develop a better general understanding of the blade's response in order to determine some possible causes of its sensitivity.

It was believed that the second mode was primarily a bending mode and test data indicated that it could be excited by a 5E excitation at about 3000 rpm, i.e., it had a natural frequency of about 250 Hz. When the blade responded in an engine test the stresses tended to be about 10,000 psi. A preliminary calculation of the blade's response to a 5E excitation came close to matching the engine data. Interestingly, the magnitude of the peak response at 3000 rpm was not strongly affected by the magnitude of the excitation.

At this point in the study, a large number of simulations were made. By starting the analyses at different frequencies and performing the calculations both from low to high frequencies and from high to low frequencies it was found that the second mode was predicted to exhibit strong nonlinear response and multiple solutions in this range. For an example, refer to Figure 12. The vertical line indicates that the solution jumped from the upper curve to the lowest curve during these calculations. Typically, when three solutions of this type are observed they are associated with a softening, nonlinear spring. In the case of the shrouded blade the softening spring is associated with shroud lift-off in the second mode. Typically, the lower solution would tend to occur during engine accels and the upper solution during decels. The middle solution is unstable and would not be observed during a test.

One interesting aspect of the response predicted in Figure 12 is that the lower solution exists over the entire range. This suggests that it is possible that the blade might never jump from the lower to the higher solution. This also implies that there is no simple linear mode in this frequency range that would be excited at lower levels of excitation. So, how does this nonlinear response develop?

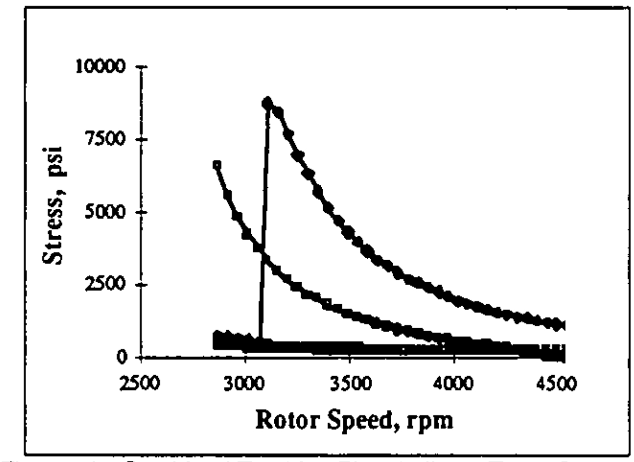

Figure 12: Combined Tracking Plots Indicate Three Solutions

Additional simulations were made at lower levels of excitation. Upon studying the predicted response it appears that a higher frequency mode, corresponding to a frequency greater than 500 Hz when the shroud is stuck, responds in an extremely nonlinear fashion and generates the observed behavior. For a value of generalized force, F_4, equal to 300, the response has a nonlinear peak of 1800 psi at a frequency of 420 Hz. When the excitation is increased by 33% then the nonlinear peak increases by a factor of five in magnitude and drops in frequency down to 260 Hz. This extremely strong nonlinear behavior appears to be associated with shroud lift-off and chatter.

It is not clear at this time whether or not the BDAMPER predictions of strong nonlinear behavior is actually representative of engine behavior. One indication that BDAMPER could be correct is that under a broad range of excitations it predicts that the peak vibratory response in the second mode occurs at about 260 Hz and has a magnitude of about 10,000 psi, for example refer to Figure 13. Thus, there appears to be some nonlinear mechanism that limits the amplitude of the response once the strength of the excitation gets above a certain level. Furthermore, the magnitude and frequency of the vibratory response are consistent with those seen in engine tests, and its on/off nature could possibly explain the observed sensitivity of the blade's response to small design changes.

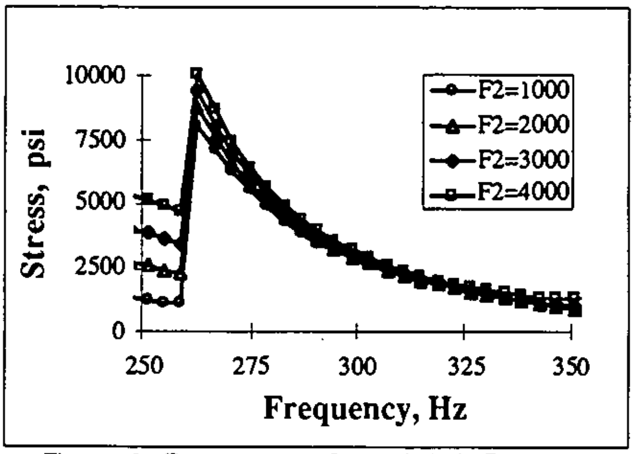

Figure 13: Response to a Second Mode Excitation

4. CONCLUSIONS AND RECOMMENDATIONS

Overall, predictions from BDAMPER appear to correlate reasonably well with the available spin pit data. An exception was the first mode response of a stage with loose shrouds. A possible explanation for this discrepancy is that the percent variations in the shroud normal loads are higher under light shroud loading and that stage mistuning effects, not considered by BDAMPER, are more important under these conditions. BDAMPER also predicted the correct shape of the tracking plot for the first mode of another fan stage, although the frequency it predicted was 20% too high. One possible reason for the frequency error is that the blade finite element model that was used to calculate the input did not include the disk and it's flexibility. A second reason could be that in BDAMPER the flexibility of the shroud is accounted for in terms of the blade's receptance which is calculated by summing the modes of a blade with an unconstrained shroud. This means that a very large number of modes could be required to accurately represent shroud bending effects. If enough modes are not used then BDAMPER would tend to over predict the stiffness of the shroud and the stage's frequency would be too high. Further study is required to fully understand this issue. Lastly, BDAMPER predicted that the second mode of the blade would behave in an extremely strong nonlinear fashion and that, as a result, it could be quite sensitive to excitation levels and direction of loading. This result is consistent with the high sensitivity that was observed in engine tests of the blade. It is sufficiently unusual, however, that it also requires further study. If the strong nonlinear behavior does, in fact, occur, then the general conditions under which it happens should be determined and avoided by improving shroud design methodology.

In general, it is concluded that BDAMPER is a very promising tool for analyzing and designing shrouded blades. An improvement to BDAMPER that is important for predicting the response of higher modes would be to include the effect of radial motion at the shroud interface. Professor Menq plans on adding this capability in the near future. It is clear that there is also a need for developing computational tools that can account for mistuning effects, i.e., blade to blade variations. The GUIde Consortium currently plans on developing this capability as part of their new, four year initiative.

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6. ACKNOWLEDGMENTS

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