EXAMINATION OF SUCCESSFUL MODAL ANALYSIS TECHNIQUES USED FOR BLADED-DISK ASSEMBLIES

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Abstract: Modal testing of bladed-disk assemblies in turbomachines is used to identify the critical natural frequencies and mode shape information used for avoiding the per-rev resonant conditions that cause high cycle fatigue (HCF) leading to premature blade and disk failures. In order to obtain the high quality modal data necessary for accurate modal identification, experience plays a major role in understanding the strengths and weaknesses associated with the variety of testing techniques. Application-specific concerns such as the blade-root disk interface connectivity, tiewire looseness and cover band design must be understood prior to test. Choices such as pre-test bladed-disk preparation, modal excitation driving point location, hammer versus shaker force excitation methods, shaker driving signal approaches, accelerometer type and location, and windowing are all important aspects that must be considered when testing specific bladed-disk configurations. Turbomachinery-specific modal analysis techniques including extraction of harmonic content and use of interference diagrams for identification of resonance conditions are also presented. The mentioned concepts are described in detail with reference to examples, which highlight the importance of the modal testing techniques implemented for a variety of applications.

Keywords: High cycle fatigue, Modal testing, Resonance, Turbomachinery

Introduction: High cycle fatigue (HCF) plays a significant role in many turbine blade failures. During operation, periodic fluctuations in the steam force occur at frequencies corresponding to the operating speed and harmonics and cause the bladed disks to vibrate. The amplitude of these vibrations depends in part on the proximity of the natural frequencies of the bladed disk to the forcing frequency. Large amplitude vibration can occur when the forcing frequency approaches or becomes resonant with a natural frequency of the bladed disk. Dynamic (alternating) stresses associated with near resonant or resonant vibration produce HCF damage and can initiate and propagate cracks very quickly [1].

Steam turbine manufacturers typically design and manufacture bladed disks with adequate margins between the forcing frequencies and the fundamental natural frequencies to avoid resonance. However, resonance with the operating speed or harmonics can occur under normal operating conditions for a variety of reasons including manufacturing variances, modified shrouding configurations, reverse engineered blades, routine wear, or any other factor that alters the mass or stiffness of the bladed disk assembly. To protect bladed disks against HCF damage from high amplitude vibration it is necessary to ensure that adequate margins exist between the natural frequencies of bladed disk assemblies and the synchronous forcing frequencies over the entire life of the bladed disk.

Modal testing can provide valuable information about the dynamic response characteristics of bladed disks at a relatively low cost. This information typically includes the natural frequencies and mode shapes of the row at room temperature and in the absence of centrifugal loading. It is important to note that the dynamic response characteristics during operation are likely to differ significantly from those measured under the test conditions due to the effects of thermal changes, fixity differences, stress stiffening, and spin softening. Thus, modal testing alone can not determine the dynamic response characteristics of turbine blade rows with sufficient accuracy to predict resonant conditions during operation. However, a variety of techniques exist for utilizing modal test results to mitigate the risk of HCF failures.

While modal testing is a fairly inexpensive and quick process, achieving meaningful results requires experience. Over the past 10 years, personnel now at Impact Technologies have developed a variety of turbomachinery specific techniques for successfully collecting, analyzing, and utilizing modal test data. This paper describes some of these techniques, and illustrates their use with examples.

<u>Turbomachinery Specific Considerations</u>: Three important considerations influence the accuracy and effectiveness of modal test data taken from bladed disk assemblies: connection stiffness within the blade-root interface and blade-tiewire interface, centrifugal loading effects, and thermal effects. When interpreting modal test results, it is critical to account for all of these conditions because an error as small as 3% in the natural frequency calculation can lead to an incorrect resonance diagnosis.

During turbine operation, turbine blade engineers rely on the centrifugal force of thousands of pounds to insure a tight fit at the blade-root interface. Centrifugal force makes the blade and disk essentially act as an integral structure, with dynamic properties distinctive from those obtained by testing the separate components. During operation, centrifugal force can also alter the effective stiffness of the turbine blades. Stress stiffening, from tensile force on the blades, causes the blade frequencies to rise as the speed increases. The modulus of elasticity of most turbine blade materials decreases as their temperature rises. Therefore, a modal test that is conducted at zero speed and room temperature requires important blade row preparation and analysis to accurately determine the dynamic characteristics of the blade row during operation.

Problems with blade fixity are best characterized by either a loose blade-root interface and/or an unsecured floating tiewire. If tested with this type of looseness, the bladed disk natural frequencies will appear lower than if they were rigidly fixed, and considerable scatter will exist between natural frequency measurements from different blades or blade groups as shown in Figure 1. The looseness will also create structural non-linearities, thus inhibiting the proper distribution of input excitation energy and leading to mode frequencies that are dependent on vibration amplitude.

A practical method for insuring fixity before performing a modal test is to apply an adhesive to the blade-root and blade-tiewire interfaces. Depending on the type of root, tiewire, and amount of looseness prescribed procedures are straightforward and easily accomplished. Blade rows that have been in service for several years since disassembly do not typically require any special treatment because deposits that form in the blade attachment region effectively lock the blade in place by filling the gaps between the blade and disk. New or recently disassembled blade rows with axial entry roots should be secured with an adhesive such as LocTite™.

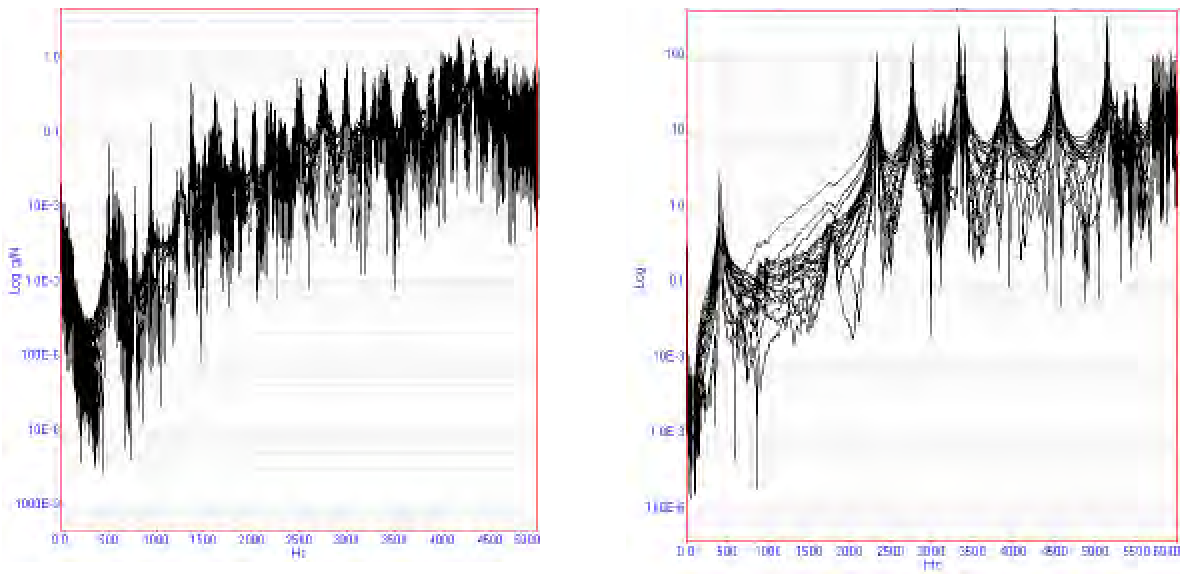

Figure 1
Frequency Response Functions from loose (Left) and tight (Right) structures

<u>Applications</u>

Quality Control: Bladed disk assemblies are particularly at risk of resonance problems following repairs or modifications that could alter the dynamic response characteristics of the structure. Procedures such as modifying shrouding configurations, or replacing blades (especially with reverse engineered blades) could create a resonance problem by changing the amount or distribution of mass or stiffness in the blade row. Gradual changes such as routine wear from erosion can also change these physical characteristics and lead to resonance problems.

Modal testing can detect changes in the dynamic response of individual blades or of an assembled row. Acceptable zero speed natural frequencies for a given blade or row are not usually available from turbine manufacturers. Turbine designers invest considerable effort in zero speed modal testing, finite element analysis, and at speed telemetry testing to ensure frequency margins of at least 5% for the fundamental modes of a bladed disk. Zero speed natural frequency specifications can be estimated by conducting modal tests on several blades, or blade groups, in their original condition (as provided by the OEM).

A resonance problem is possible when the zero speed natural frequencies of worn, modified, or replacement blades deviate from those set by the manufacturer.

Resonance Investigation: A resonance investigation is advised when evidence of HCF is discovered during a failure investigation or inspection. The most effective and economical method for identifying potentially resonant conditions in bladed disks utilizes a combination of modal test results and finite element analysis (FEA) to accurately predict the dynamic response characteristics of the structure under operating conditions. While FEA alone can predict the natural frequencies and mode shapes of a blade row during operation, experience indicates that calibration of such complex models with modal test results is necessary to accurately predict resonance. Figure 2 shows the combined testing and modeling approach.

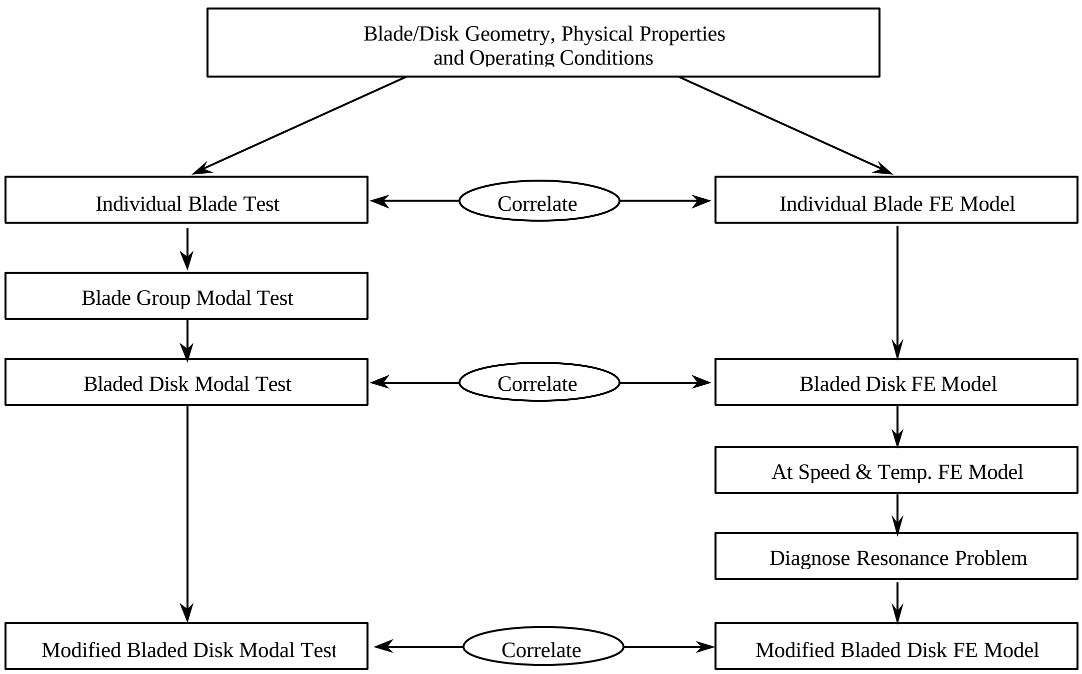

Figure 2
Block Diagram of Resonance Investigation

A resonance or failure investigation begins with the development of a BladePro [2] finite element model of the bladed disk under investigation. BladePro utilizes the FEA power of ANSYS® to significantly reduce the time required for performing structural analysis of turbine blades. The program assists the user in all aspects of turbine blade analysis: model generation, boundary condition application, analysis options, job submission, post processing, and life assessment.

A three-dimensional finite element model of a single blade is developed using templates for different components (airfoil, shroud, dovetail, and disk) provided with the BladePro software. Figure 3 shows part of the model generation process. The single blade model is eventually condensed into a “superelement,” that is replicated numerous times to form a full bladed disk assembly model. By changing a few simple input parameters in the software, the model is exercised under a variety of operating conditions and under the modal test conditions to determine the dynamic response characteristics of the blade row. Correlation of the single blade and full bladed disk models with the corresponding modal test results lends credibility to the model, and supports the calculated dynamic characteristics during operation.

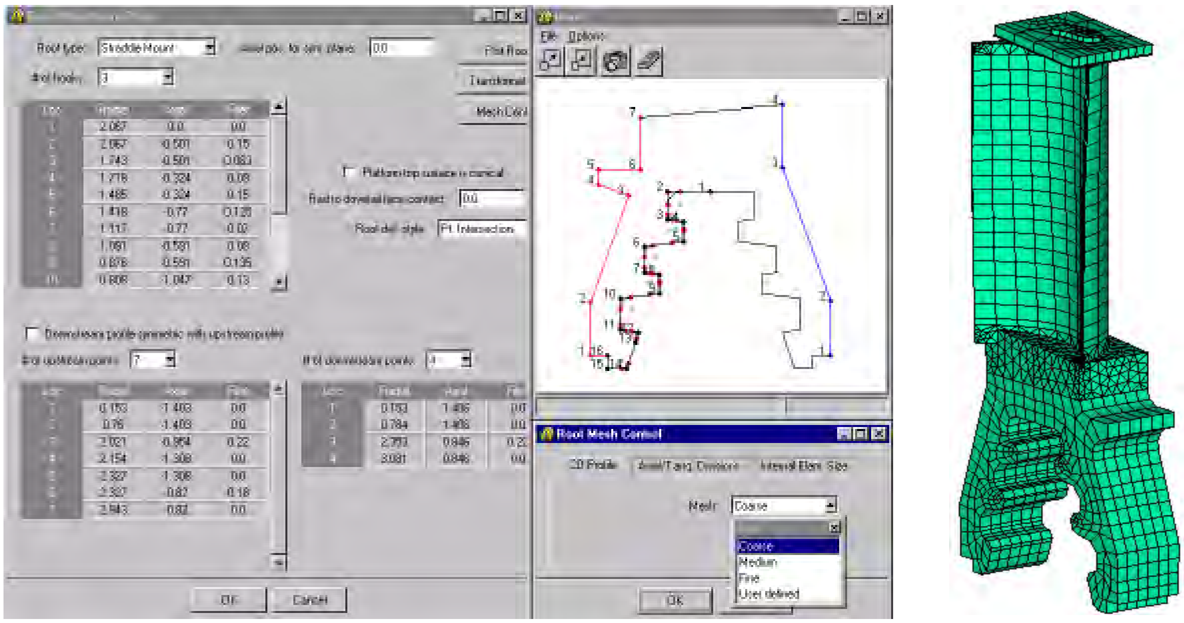

Figure 3
BladePro model generation

When a blade row resonance condition is identified, the introduction of a small change in the blade row’s natural frequencies is the most effective way to reduce the dynamic stress and increase the component fatigue life. Depending on the frequency, a frequency margin of 3% is usually enough to lower the dynamic stresses and ensure that the row remains detuned regardless of any minor changes that might occur in the future. Although it is usually more practical to modify a blade row by adding mass, mode frequencies can either be decreased or increased to achieve the necessary frequency margin. Increasing the bladed disk natural frequencies can be accomplished by stiffening the lower region of the airfoil, or by removing material from the blade tip or coverband. Decreasing the natural frequencies is likewise accomplished by removing material from the airfoil base, or by adding mass to the blade tip or coverband.

The most practical method for altering these natural frequencies to within allowable limits is by the addition of mass to the blade tip, tiewire, or coverband. This approach has the following advantages: 1) minimal changes to the airfoil that can significantly affect aerodynamic operations; 2) adding mass can usually be accomplished without removing any blades. Common techniques for adding mass to a blade row include brazing stainless steel sleeves to existing tiewires, brazing aerodynamic masses to the convex side of the airfoil tip, and increasing weld fillets around the tiewire or coverband regions. Various modification strategies are first evaluated using the finite element model. The objective is to introduce small amounts of mass to the model of the blade row that are easy to implement and result in essentially infinite component life.

Once the test engineer and relevant plant personnel choose a modification strategy, the detuning weights are temporarily attached to the blade row with an adhesive. Next, a modal test of the bladed disk is performed to assess the effect of the modification. Once satisfied with the detuning results, the weights are permanently attached to the blade row. A final modal test is conducted to verify that the frequency shift produced by the weights remains within specifications.

Test Procedures: A complete modal test of a bladed disk assembly consists of measurements to determine the dynamic response characteristics of the entire assembly and its substructures (blade groups or individual blades). Detailed substructure measurements are used to identify the fundamental mode families; i.e. tangential, axial, torsional. The motion associated with the tangential modes is primarily in the circumferential direction. For the axial modes, the major displacement component is in the rotor axial direction. The torsional modes of a single blade or blade group exhibit a twisting motion, where the leading edge is 180° out of phase with respect to the trailing edge or blade. Measurements of the entire structure's response are used to identify amplitude and phase variations in the response of the substructures.

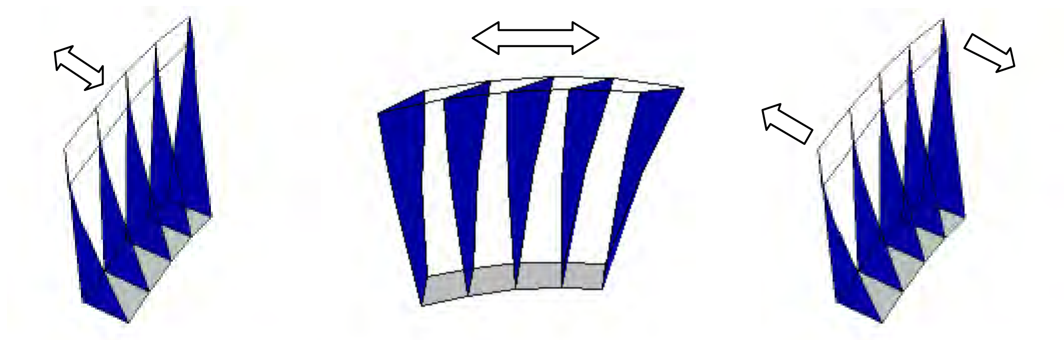

Figure 4
Fundamental mode families. Axial (left), Tangential (center), Twist (right)

The natural frequencies and associated deflection (mode) shapes of substructures are measured by "meshing" the substructure with a sufficient number of measurement points to describe the substructure's deflection. For blade groups, a mesh consisting of three points per blade (near the platform, mid span, and tip) is sufficient to document the fundamental mode shapes. Testing all of the substructures is unnecessary since, due to symmetry, all of the substructures exhibit similar dynamic response characteristics.

While all of the substructures exhibit similar mode shapes at a given natural frequency, the amplitude and phase (ideally 0 or 180°) varies from substructure to substructure. This "disk effect" is measured meshing the entire row with measurement points. Typically, one point per substructure, in a consistent location, is used. Differences in measured frequencies from substructure to substructure can be anticipated as the normal consequence of dimensional tolerances, manufacturing defects, material processing and installation procedures. These standard variances prevent any one blade from being a "perfect" copy of any other. The result of these minor differences is seen as "scatter" of natural frequencies about a mean value. Scatter of 5% is normally considered acceptable among the manufacturing community for low-pressure steam turbine blades. The full row modal test also documents the scatter associated with differences between substructures.

To excite a bladed disk, a modal hammer and a 75-pound force electromagnetic shaker have been used successfully. A modal hammer fitted with a piezoelectric force transducer can be used to excite the structure under investigation with a transient (impact) force. Hammer excitation generally involves less setup effort than the shaker, but the hammer requires more effort and skill during data collection. The shaker is connected by a stinger to a piezoelectric force transducer attached to the driving point on the structure. A cyanoacrylate or epoxy adhesive works well for bonding the force transducer to the blade row, so long as the driving point is not on a curved surface such as the leading edge of the airfoil. For testing unmounted blades, a modal hammer is preferred.

Selection of the driving point location is based largely on the objective to excite as many of the fundamental modes as possible. So long as the structure is not excited at a node (a point on the structure that is stationary for a given mode) and the structural response is linear, for all practical purposes the mode shapes will be independent of the driving point location. The applied force should act in the rotor axial and disk tangential directions to unsure coupling with both the axial and tangential modes. A driving point on the platform or mid span on the airfoil of the leading or trailing blade in a group generally excites the first and second bending modes in each direction. It is best to avoid flexible areas such as the trailing edge where significant localized deflection is likely. Access considerations, especially for shaker excitation, often necessitate selection of a driving point near the shroud.

A modal hammer is commonly used in modal analysis to provide a transient excitation of structures. A piezoelectric force transducer in the hammer tip measures the transient force applied by the hammer when it strikes an object. A brief impulse of this type in the time domain corresponds to broad band excitation in the frequency domain, and can therefore excite many modes simultaneously. Shorter impulses in the time domain, produced by harder hammer tips, excite a broader range of frequencies. A hard hammer tip such as steel is usually necessary for testing bladed disk assemblies. The response to at least five hammer impacts must be averaged to achieve stable results.

An electromagnetic shaker is commonly used in modal analysis to excite of structures with a user-selected signal. For most steam turbine bladed disks, a 75-pound electromagnetic shaker can adequately excite the structure. Broad band of forcing signal such as random (white) noise, burst random, or random multi-sine (a superposition of bin-centered sinusoids with random phase relationships) are used to simultaneously excite many modes in the frequency range of interest.

Accelerometers are used to measure the vibratory response of each measurement point on the structure to the applied excitation. A pair of roving accelerometers oriented in the axial and tangential directions to the bladed disk (as shown in Figure 5) is moved from measurement point to measurement point during the data collection process. Wax or a small magnet holds the accelerometers firmly in place during data collection. Lightweight accelerometers are used to minimize the effects of mass loading that can reduce the natural frequencies of the substructure where the accelerometer is attached. Mass loading is of greater concern when measuring the natural frequencies of unmounted blades because the total mass of the structure under investigation is much smaller.

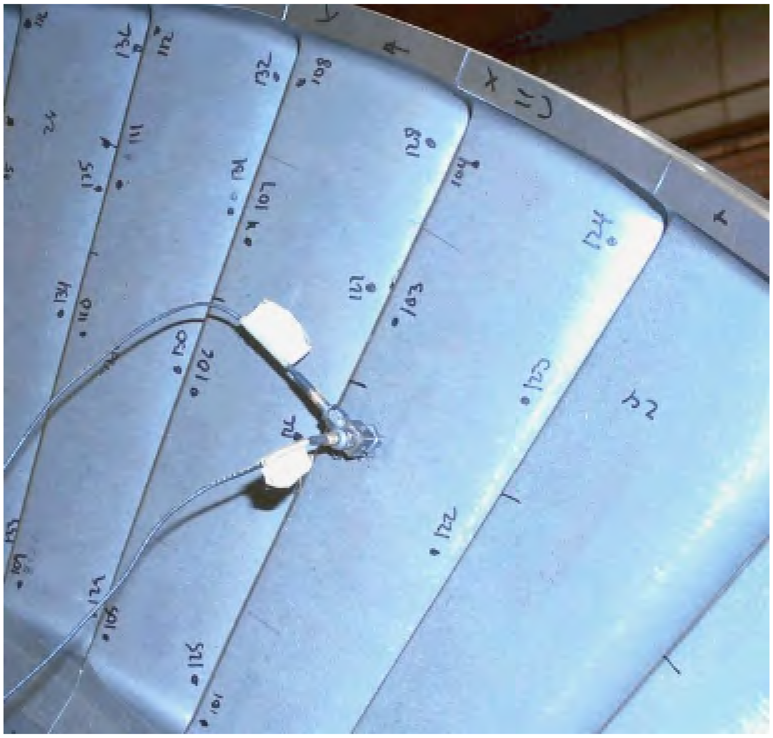

Figure 5
Accelerometer Configuration

Excitation and response signals from the test are processed using a multi-channel dynamic signal analyzer to compute a Frequency Response Function (FRF) for each measurement point. To prevent errors in the fast Fourier transformation (FFT) that forms the basis for this calculation, the measured signals must be periodic within the sampling (data collection) interval. Periodic signals will have an integer number of cycles within the sampling interval so that the signal begins and ends at the same value, while non-periodic signals exhibit a discontinuity between the beginning and end of the sampling interval. Weighting windows are used to control FFT errors by reducing the signal amplitude to zero at the beginning and end of the sampling interval.

Unfortunately, windowing introduces artificial damping that can mask the structural damping. Finite element models typically use a nominal structural damping ratio of 0.2%, but model results can be improved with more accurate damping information from a modal test. A Hanning window is commonly used when testing with a shaker driven by a continuous random noise signal. For a hammer test, a user-defined window is used to suppress the response at the end of the sampling interval. Random multi-sine excitation is unique in that windowing is not necessary because the structure is driven at discrete bin-centered (periodic within the sampling interval) frequencies.

Modal Analysis: Modal test data from a hammer or shaker test is reduced on a personal computer with modal analysis software [3]. Peaks in the acquired frequency response functions indicate the blade row natural frequencies. The mode shape and mode damping ratio associated with each peak or natural frequency is identified by curve fitting one or more peaks at a time. A polynomial equation for the FRF is fit in a least-squared-error sense to specified frequency bands of the measurement data. This information is used to build a table containing normalized mode shape amplitude and phase data for each measurement point and direction, and at each mode frequency.

Interference Diagram: For a bladed disk structure, the fundamental natural modes of vibration can be categorized as tangential, axial, or torsional. Associated with each of these fundamental mode families are a series of "nodal diameter" or "disk effect" modes. When one of these modes is excited, the amplitude of vibration can vary harmonically around the disk and the phase relationship between substructures may change as shown in Figure 6. The number of complete sinusoids around the disk (nodal diameter number) is used to describe the mode. The nodal diameter number can also be determined by dividing the number of nodes by two.

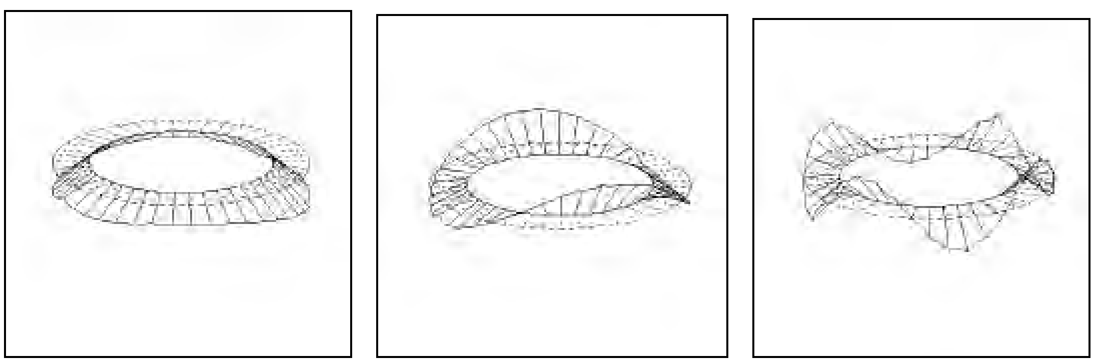

Figure 6
Disk effect examples of zero (left), two (center), and four (right) nodal diameters

In a bladed disk structure, for a mode to be strongly excited, the forcing frequency must match the natural frequency of the row, and the force distribution must match mode shape. In other words, the number of per-rev excitations must equal the number of nodal diameters. An interference diagram is a tool used to identify situations in which both of these conditions are met.

Figure 7 shows a typical at speed interference diagram with points representing nodal diameter modes belonging to one fundamental mode family. The number of nodal diameters is plotted along the X-axis while the natural frequencies are plotted along the Y-axis. The diagonal line with slope of 1 per-rev/nodal diameter is the impulse line. A strong resonant response is guaranteed if the impulse line passes through or close to a mode point. Note that such a situation satisfies both of the conditions required for a resonance response of the blade row, namely a frequency match and strong coupling between the mode shape and applied force.

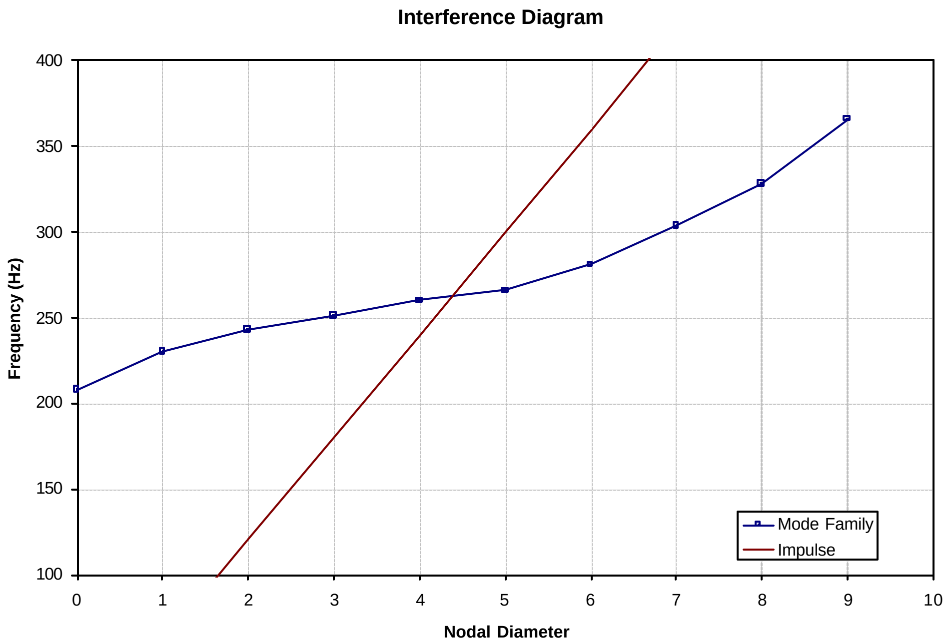

Figure 7
Interference diagram showing 4 disk effect modes belonging to one family

From the results of a modal test alone, it is only possible to construct a zero speed interference diagram. The effects of stress stiffening, stage operating temperature, and changes in operating boundary conditions must be computed with a BladePro finite element model. This combined approach makes it possible to construct an accurate at-speed interference diagram at a reasonable cost. Purely experimental at-speed interference diagrams are only obtainable through strain gauge telemetry tests.

Implementation: A brief example from the author's experience is presented to illustrate the use of modal testing in resolving a HCF problem. After replacing a row of LP turbine blades, multiple failures of the new blades occurred over a six-month period. A third party blade manufacturer provided the replacement blades based on reverse engineering of the original sample blades from the OEM. Prior to the row replacement and recent failures, the blade the row had operated without any failures in excess of 30 years. The rapid nature of the failures during the six-month period suggested that the problem was due to HCF from a strong resonant condition rather than corrosion, material defects, or low cycle fatigue.

The objective of the failure investigation was to determine the root cause of the blade row failures based on modal test data and finite element results. A single blade was analyzed to determine the geometric dimensions required for developing a finite element model. Models of the unmounted blade and the complete bladed disk were developed.

The single blade model consisted of two tiewire segments, airfoil, platform, root, and disk sector. By removing the tiewire and disk segment from the model, the natural frequencies of the freely, and cantilever supported blade can be calculated. The unmounted blade was frequency tested under both support configurations and compared with the corresponding analytical results to verify the accuracy of the single blade model. The single blade (sector) model was converted to a full row model by repeatedly replicating the sector to form a “superelement”.

Based on the small differences between the measured and calculated frequencies of the single blade, it was concluded that the finite element model would be able to reliably represent the dynamic characteristics of the blade row. Aside from the single blade model, the more important goal of the investigation was concerned with the bladed disk natural frequencies under normal operating conditions.

A complete bladed disk modal test was performed on the blade row under investigation. After reducing the data and incorporating the finite element results an at speed interference diagram was constructed. Examination of the interference diagram revealed the possibility of a near resonant condition associated with the 6th nodal diameter mode. The test engineer, plant personnel, and blade installation specialists considered various modification strategies to detune the resonant condition. It was decided to add mass at specific locations along the outer tiewire to lower the bladed disk natural frequencies. In the original blade row design, the frequency margin for the 6 nodal diameter mode was only 6 Hz, or 1.7%. A frequency margin of at least 3% is recommended to ensure resonance free operation. Calculations indicated that a minimum of 200 grams distributed along the outer tiewire would be required to achieve this frequency margin.

Thirty 9.5 gram masses were temporarily attached to the outer tiewire. The row was tested again to evaluate the resulting frequency shift in the 6 nodal diameter mode. A frequency shift of 8 Hz, yielding a frequency margin of 14 Hz or 4.1% of the 6 nodal diameter frequency was measured. The masses were permanently attached to the tiewire, and the modal test was repeated as a final verification of the detuned blade row.

Conclusions: Modal testing can provide valuable information about the dynamic response characteristics of bladed disks at a relatively low cost. This information typically includes the natural frequencies and mode shapes of the row at room temperature and in the absence of centrifugal loading. While the dynamic response characteristics of bladed disks during operation are likely to differ significantly from those measured under the modal test conditions, these “zero speed” natural frequencies can serve as a valuable quality control test. The most effective and economical method for identifying potentially resonant conditions in bladed disks utilizes a combination of modal test results and finite element analysis (FEA) to accurately predict the dynamic response characteristics of the structure under operating conditions. This natural frequency and mode shape information is plotted on an interference diagram that identifies resonant conditions with the turbine operating speed that are likely to produce a strong response. When a resonant condition exists, the finite element model is used to evaluate different blade row modification strategies. Follow-up modal tests are performed to after any modifications to verify the detuning effects.

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