S&V OBSERVER

Interference Diagrams – When Campbell Diagrams Aren't Enough

Eric Olson, **Steve Hogg**, **Maki Onari** and **William Marscher**
Mechanical Solutions Inc., Whippany, New Jersey

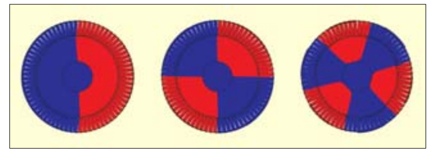

Figure 1. One, two, and three nodal diameter/ zero nodal circle ("umbrella mode shapes") for a steam turbine bladed disk.

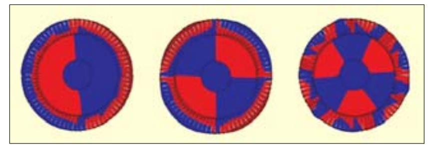

Figure 2. One, two, and three nodal diameter/ one nodal circle for a steam turbine bladed disk.

The intent of this article is to help readers understand more about interference diagrams, how they are used, and when an equipment user or buyer should specify the use of an interference diagram in assessing the potential for resonance problems in a centrifugal impeller or axial-bladed disk. This information is of relevance to pumps, compressors, and turbines that are undergoing a design audit or that have experienced unexpected fatigue failures of the blades, shrouds, hub outer diameter (OD), or disk rim.

Interference diagrams can be prepared using information from either numerical analyses or from modal testing. Sometimes we combine both analysis and test techniques so that the effect of fit-up interferences and/or casting tolerances can be crosschecked by the test. Also, the analytical model can be calibrated by these real-life effects before being used in downstream calculations. An example is forced-response stresses for plotting on fatigue Goodman Diagrams. The numerical results produced using finite-element analysis (FEA) are used to help identify potential fatigue and cracking problems in bladed disk elements such as turbine rotors and compressor impellers. One outcome of a typical FEA analysis is the identification of natural frequencies that coincide with machine running speeds multiplied by select excitation sources such as vane pass frequency, blade pass frequency, etc., producing so-called "excitation orders" (EOs). For example, an EO of 5 is a frequency equal to five times the running speed. This information alone, communicated on a Campbell diagram of frequency vs. running speed, is not enough to make design decisions. Typically, a Campbell diagram that includes all of a bladed disk's vibration modes indicates more potential problems than actually exist, as will be explained below. Interference diagrams help the experienced engineer sort out which modes are most likely to actually cause problems.

An interference diagram is a plot of natural frequencies versus number of "nodal diameters" (circumferential "zig-zag" convolutions of the mode shape for a given natural frequency) distributed in groups or families, each characterized by a certain number of "nodal circles" (radial "zig-zag" convolutions). The convolutions include combined distortion of the blades at the disk rim as well as the disk itself and of the shroud (if any) as well as the hub. Some modes are dominated by blade motion and some by disk or hub/shroud motion, but all modes exhibit the nodal diameter/nodal circle character.

Figure 1 displays typical mode shapes for one, two, and three nodal diameter/ zero nodal circle (umbrella modes) of a steam turbine bladed disk. The nodal circle is a circumferential line inside the rim of the disk/impeller. Figure 2 displays the same turbine bladed disk modes for one, two, and three nodal diameter/ one nodal circle mode shapes. In both figures, the red color indicates motion out toward the reader, and the blue color shows the motion away from the reader (in the axial direction). There are also modes where the nodal diameters involve tangential motion clockwise vs. counterclockwise, and/or radial motion outward from the center, or inward toward the center.

The mode shapes are roughly on the order of increasingly complex patterns (1st, 2nd, 3rd, etc), analogous to bending modes in a beam, and depend on the distributed stiffness and mass of the structure. Figure 3 is an example of a bladed disk mode shape plotted from onsite test data. A node in the terms "nodal diameter" and "nodal circle," is a boundary line on the structure that is stationary, while the rest of the structure is vibrating in one direction on one side of the line and in the opposite direction on the other side of the line. Interference diagrams are particularly useful for evaluating such patterns vs. the patterns of probable excitation pressure fields. Therefore they are useful in predicting the strength of the ensuing vibration and attendant cracking problems even when a large number of complex modes are potentially involved.

An interference diagram simultaneously takes into account natural frequency and mode shape, and nodal diameters are used to sort out which specific modes will cause a problem or (in an existing machine) are causing a problem. If identified during the machine design, the component design can be modified (i.e. changing the mass by modifying the material of the shroud for turbine disks, or changing the number of the diaphragm nozzle blades to increase or decrease the difference between this number and the number of blades, etc.). If discovered after a machine is installed, the operation of the machine can be changed to avoid certain running speeds, or a retrofit may modify components so that they pass interference diagram inspection.

The following describes how an interference diagram is interpreted. The example is based on analysis and redesign work performed within the last couple of years on a steam turbine installed in 1996. The machine had 72 blades, includ-

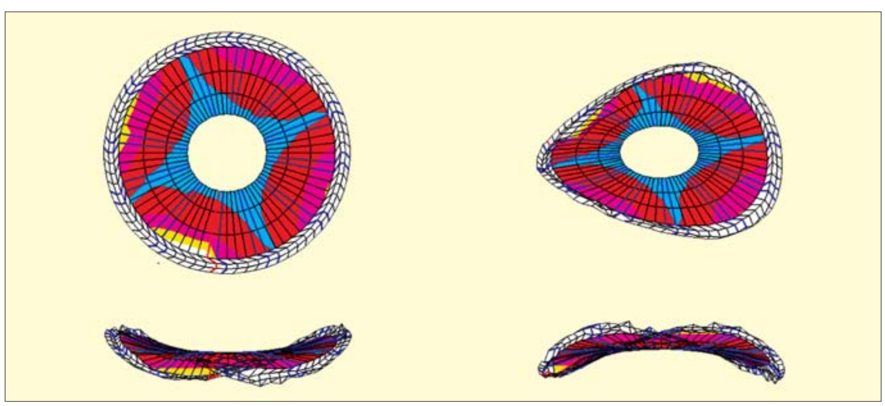

Figure 3. An example of a single turbine bladed disk mode shape obtained from data collected by the authors with on-site modal testing. A single disk will have hundreds of modes, most of them benign. In this case, the field testing and interference diagram were used to help resolve a problem. The modal test data was used to create the mode shape of the disk at specific natural frequency. An accurate FEA model of the disk and blades reproduces exactly which mode was causing problems, since testing during operation and a forced response analysis can determine whether the alternating stresses from a given force (e.g. nozzle force producing torque) exciting this natural frequency excitation would be enough to cause a failure. A design modification was made to resolve the problem. Had an interference diagram been properly used before the machine was built in 1996, the problem might have been caught during the design phase.

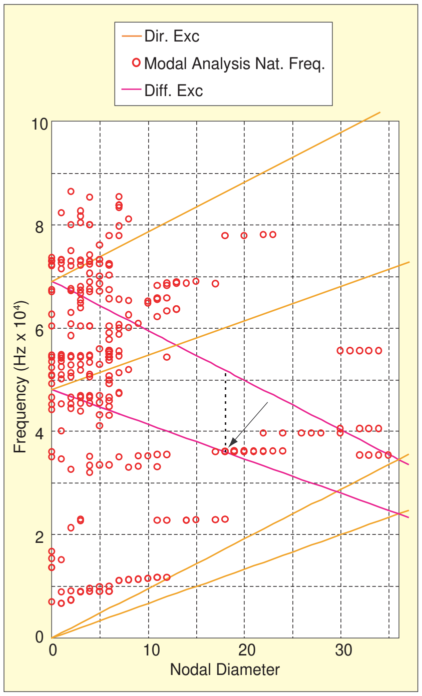

Figure 4. Potential problem mode was identified by using an interference diagram. As a result, the number of nozzles on the turbine was recommended to be reduced from 90 to 88 or less to avoid resonances. The pairs of diagonal lines represent the lower and upper speed ranges, respectively.

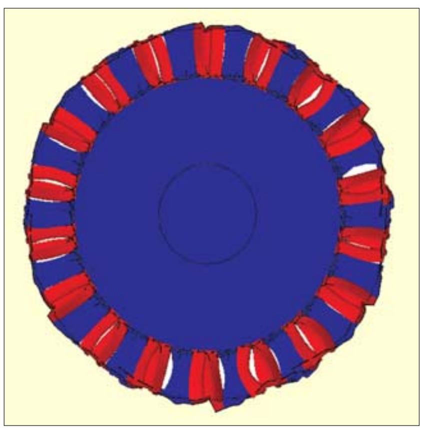

Figure 5. Potential problem mode identified as 18 nodal diameter/zero nodal circle at 3597 Hz.

ing a locking blade, grouped in 12 groups of 6 blades, and there were 90 nozzles on the adjacent diaphragm. In this case, a problem was discovered after years of operation of the machine due to low cycle fatigue where the failure took place at the upper neck of the fir tree of the first blade of one packet, as viewed from the admission end (CW rotation). "First blade" in a shrouded group means the first blade encountered by a given nozzle as the rotor spins. The blade was thrown from the wheel, causing severe damage to the downstream stages.

A modal analysis of the complete bladed disk was performed to determine a potential problematic mode in the interference diagram. With reference to Figure 4, a resonance problem is indicated when any natural frequencies fall within the range of the intersection between the purple backward-slanted (i.e., gets lower as the plot progresses to the right) "blade/vane difference" straight lines and the vertical dashed nodal diameter line, or if they cross the forward slanted "direct excitation" yellow straight lines at a nodal diameter equal to the number of nozzles. The pairs of diagonal lines on this plot represent the lower and upper speed ranges, respectively. A resonance is predicted at minimum speed at the location where the blue arrow is pointing.

The reason for being concerned with the "difference excitation" involving nodal diameters equal to the number of blades minus the number of nozzles is that blade/nozzle interactions are nonlinear and result in pressure fields that include a strong response distributed in space with lobes equal to the nozzle number *n*, a usually weaker "reflection" component with spatial lobes equal to blade number *m* and strong responses with lobes at *m*–*n* and *m*+*n*. The *m*+*n* involves very high nodal diameters that involve equally high frequencies that are outside the range of typical turbomachinery excitations. However, the *m*–*n* nodal diameter frequencies are very often in the range of expected excitations and have been observed to cause failures that were unexpected by groups that were not yet using the interference diagram method. This was first reported in the literature around 1980 by Jay (Allison Engine Company), who subsequently avoided future failures by applying the interference diagram method.

From Figure 4, a natural frequency at 3597 Hz is identified as a potential problematic mode, since the relatively simple 18 nodal diameter mode shape matched the spatial excitation patterns of the anticipated blade/vane difference pattern of 18 (Figure 5) and is close to the minimum speed nozzle-passing excitation frequency. Even with relatively low energy due to the intersection being only at minimum speed, the mode shape could have near maximum available oscillating energy fed to it, depending on the nozzle spray pattern details. As a matter of principle, we consider this mode a good candidate for strong resonant response because of the relatively simple shape of the blade displacements and the consistent structure of the blade shapes over the nodal diameter segments. This facilitates a match-up with the expected excitation force pattern. In this instance, we recommended that the number the adjacent nozzles upstream of the bladed disk be reduced from 90 to 88 to avoid resonance problems. This reduced the difference between the nozzle and the blades from 18 to 16, where no natural frequencies were found according to the interference diagram.

The author can be contacted at: eric.olson@mechsol.com.