

Interference Diagrams: When Campbell Diagrams Aren't Enough

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The intent of this article is to help readers understand more about Interference Diagrams, how they are used, and when an equipment user/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 article is of relevance to pumps, compressors, and turbines that are under-going a design audit, or that have experienced unexpected fatigue failures of the blades, shrouds, hub O.D., or disk rim.

Interference Diagrams can be prepared using information from either numerical analysis or from modal testing. Sometimes the authors combine both the analysis and test techniques so that the effect of fit-up interferences and/ or casting tolerances can be cross-checked by the test, and the analytical model can be calibrated by these real-life effects before being used in downstream calculations, for example, of forced response stresses for plotting on fatigue Goodman Diagrams. The numerical results are produced using Finite Element Analysis (FEA) results, and are used to help identify potential fatigue and cracking problems in bladed disk elements such as turbine rotors and compressor impellers. One outcome of the typical FEA analysis is the identification of natural frequencies in resonance with machine running speeds multiplied by select excitation sources such as vane pass frequency, blade pass frequency, etc., producing so-called “excitation orders” EO (an EO of 5 is a frequency equal to 5 times 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 of the paper and the blue color shows the motion into the paper (in the axial direction). There are also modes where the nodal diameters involve tangential motion clockwise vs. counter-clockwise, and/ or radial motion outward from the center, or inward toward the center.

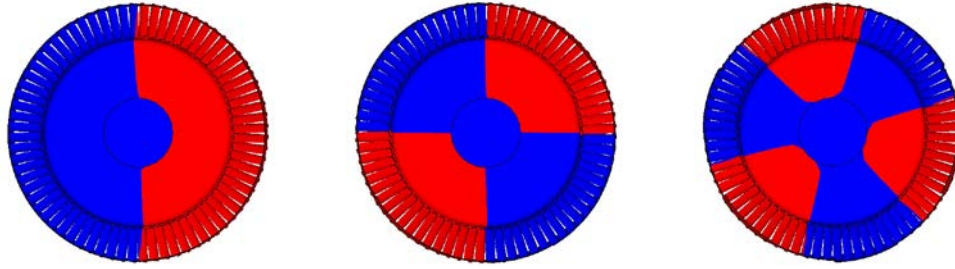


Figure 1. One, two and three nodal diameter/zero nodal circle (“umbrella mode shapes”) for a steam turbine bladed disk analyzed by the authors.

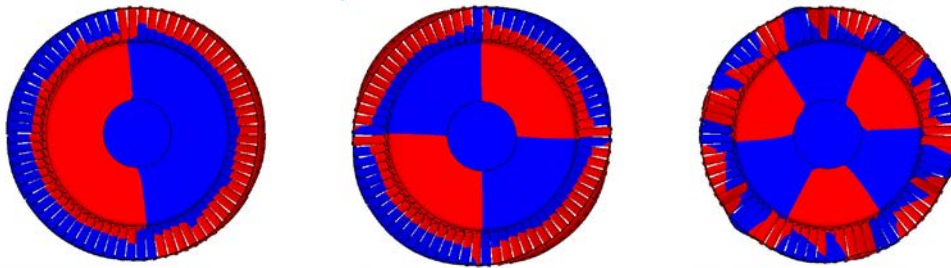


Figure 2. One, two and three nodal diameter/ one nodal circle for a steam turbine bladed disk analyzed by the authors

The mode shapes are roughly in 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. In the terms “nodal diameter” and “nodal circle”, a node 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, and thereby 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, 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 to be changed to avoid certain running speeds, or a retrofit is required which modifies components in a manner that passes 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 including 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 in order to determine a potential problematic mode in the Interference Diagram.

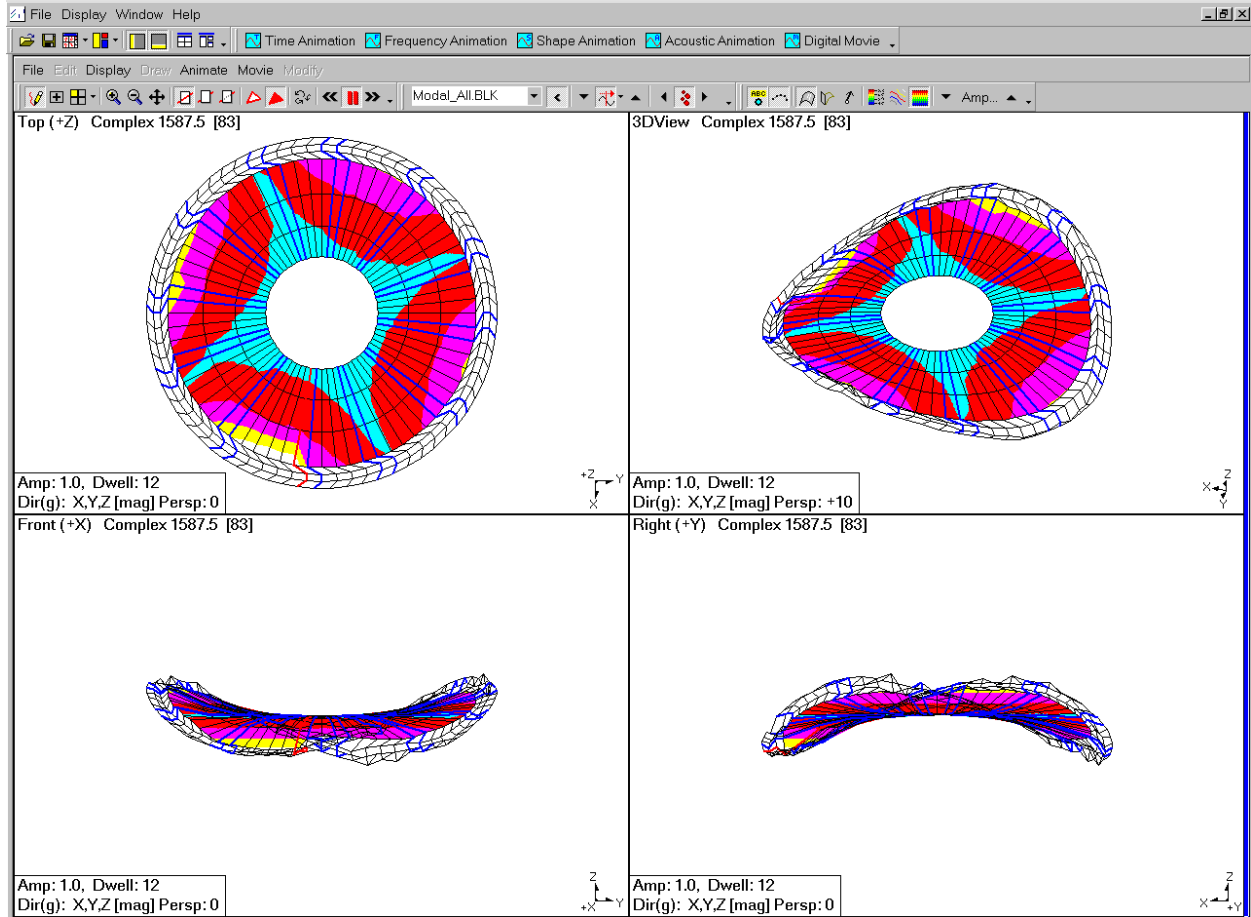


Figure 3: An example of a single turbine bladed disk mode shape obtained from data collected by the authors with on-site modal testing. A single disk will have hundreds of modes, most of them benign. In this case, the field testing and Interference Diagram were used to help resolve a problem. The modal 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 as the 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.

With reference to Figure 4, a resonance problem is indicated when a natural frequency falls near where any slanted line crosses a multiple of the nozzle pass frequency. These intersection points

will fall at nodal diameters that correspond to sum and differences between the number of nozzles (n) and number of blades (m). The pairs of diagonal lines on this plot represent the lower and upper speed ranges, respectively.

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 which 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, such as first reported in the literature around 1980 by Jay of Allison, who subsequently avoided future failures by applying the Interference Diagram method.

Referring to Figure 4, at a running speed of 5193 RPM, the nozzle pass frequency is 7790 Hz ($90 \text{ vanes} \times 5193/60$). So any natural frequency near 7790 Hz which exhibits an 18 nodal diameter pattern (90 nozzles minus 72 blades) would potentially be resonant. As shown, such a mode was predicted (Figure 5). At 5193 RPM, this mode shape would have near maximum available oscillating energy fed to it, depending upon the nozzle spray pattern details. As a matter of principle, the authors considered 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, facilitating match-up with the expected excitation force pattern. In this instance, the authors 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.

A simple rule of thumb for equipment users is to specify an Interference Diagram (sometimes called a SAFE Diagram) in situations where your company currently specifies Campbell Diagrams. Interference Diagrams should be provided for any critical piece of new equipment or if a vendor is supplying a newly designed compressor impeller or turbine wheel for an existing machine. Interference Diagrams should also be provided by independent third parties who are performing an audit of the OEMs machine under contract with the OEM, End-User, or architect/engineer.



Figure 4. Potential problem mode was identified by using an Interference Diagram. As a result, the number of nozzle of 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.

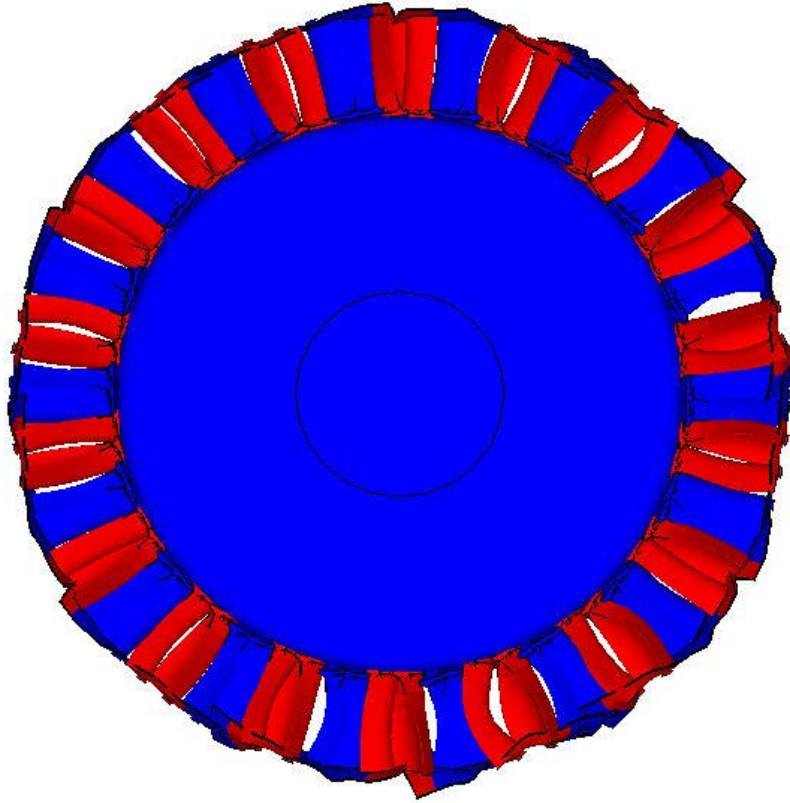


Figure 5. Potential problem mode identified as 18 nodal diameter /zero nodal circle at 7790 Hz.

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