

Simulation with simple or complex models of the dynamical behaviour of mechanical systems, performed by ANSYS, is not only essential for design but also for processing measured data.

Two examples are shown related to blade vibrations and Tip-Timing Measurement results.

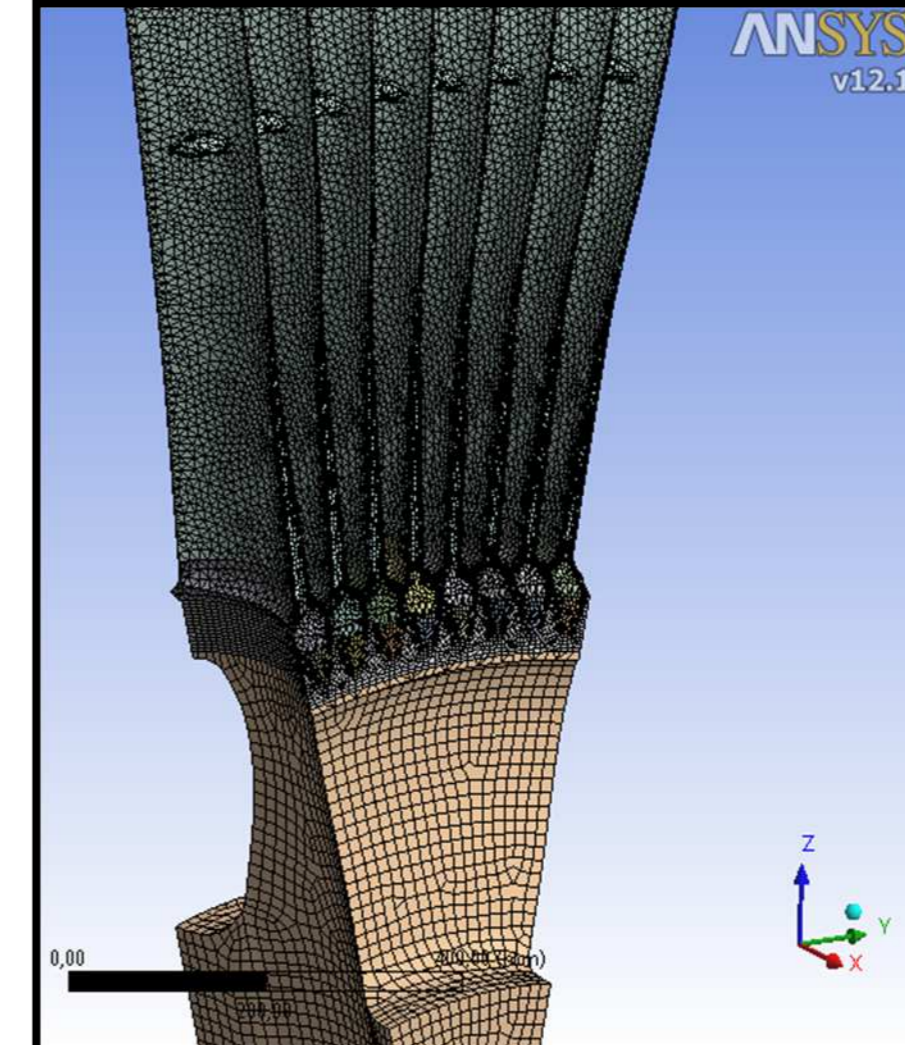
**CASE HISTORY:** In the L-0 stage of a condenser steam turbine, composed by 120 blades, grouped in packs of 8 blades, an inspection revealed that cracks had developed in the first and last blades of several packs of 8 blades in critical operating conditions. The modal analysis of the 3D model of the pack by means of ANSYS Workbench allowed to define natural frequencies and associated modal shapes. The first tangential and axial modes presented roughly the same vibration amplitudes for all blades of the same pack. The third mode (at 225 Hz) was the so called X-mode, which presents maximum amplitudes for first and last blade of the pack. Therefore it was assumed that the pack vibration mode which was excited and caused the cracks was the X-mode.



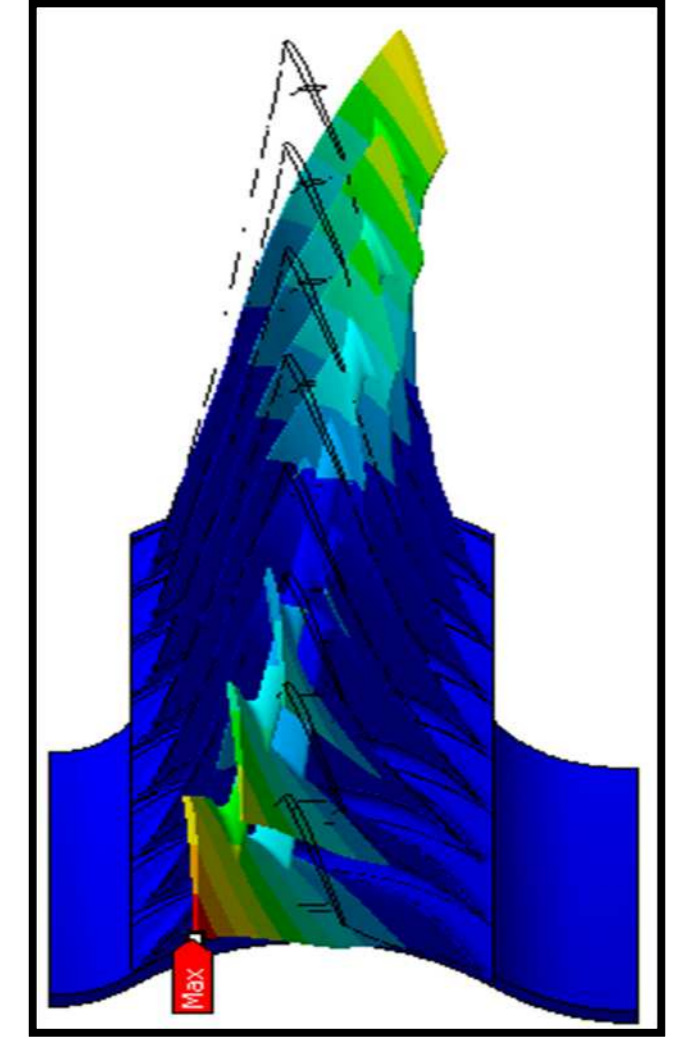
L-0 blade row: 120 blades with fir tree roots, grouped in packs of 8 blades by welded snubbers.



Crack developing in the fir tree root of the first blade of the pack of 8 blades.

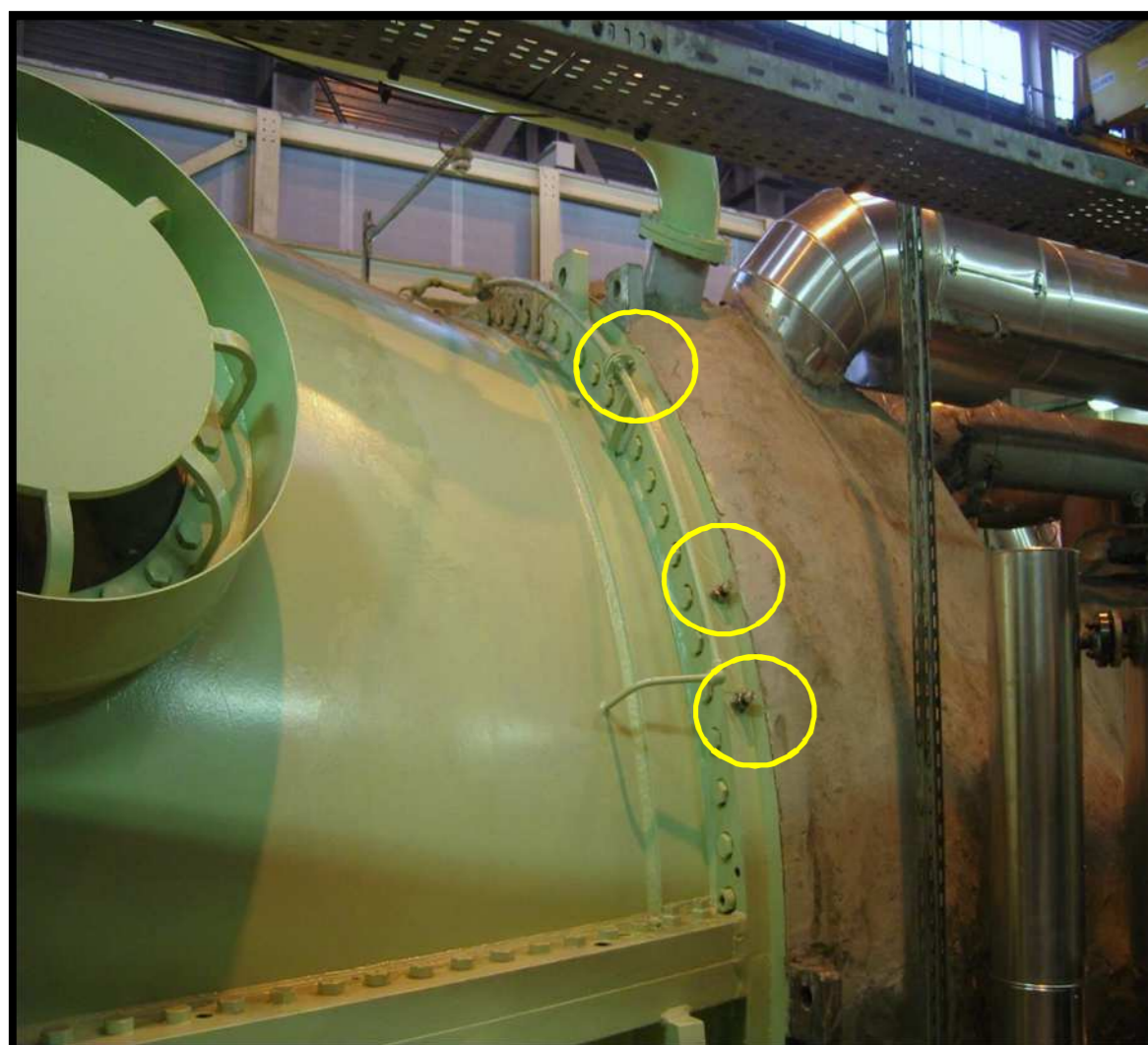


Mesh of the 3D model of the 8 blade pack (ANSYS Workbench).

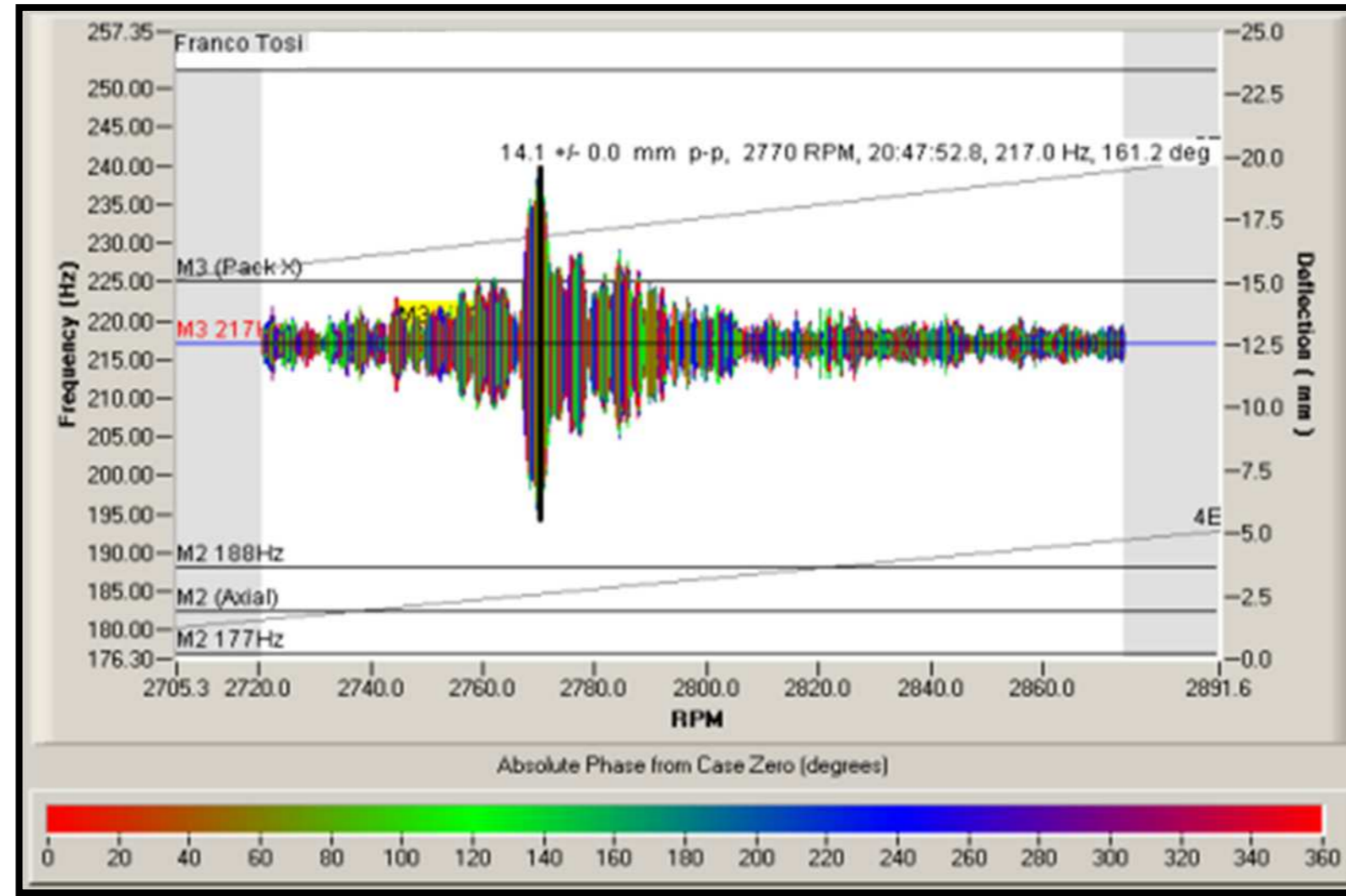


Modal analysis results 3rd mode (X-mode) at natural frequency 225 Hz.

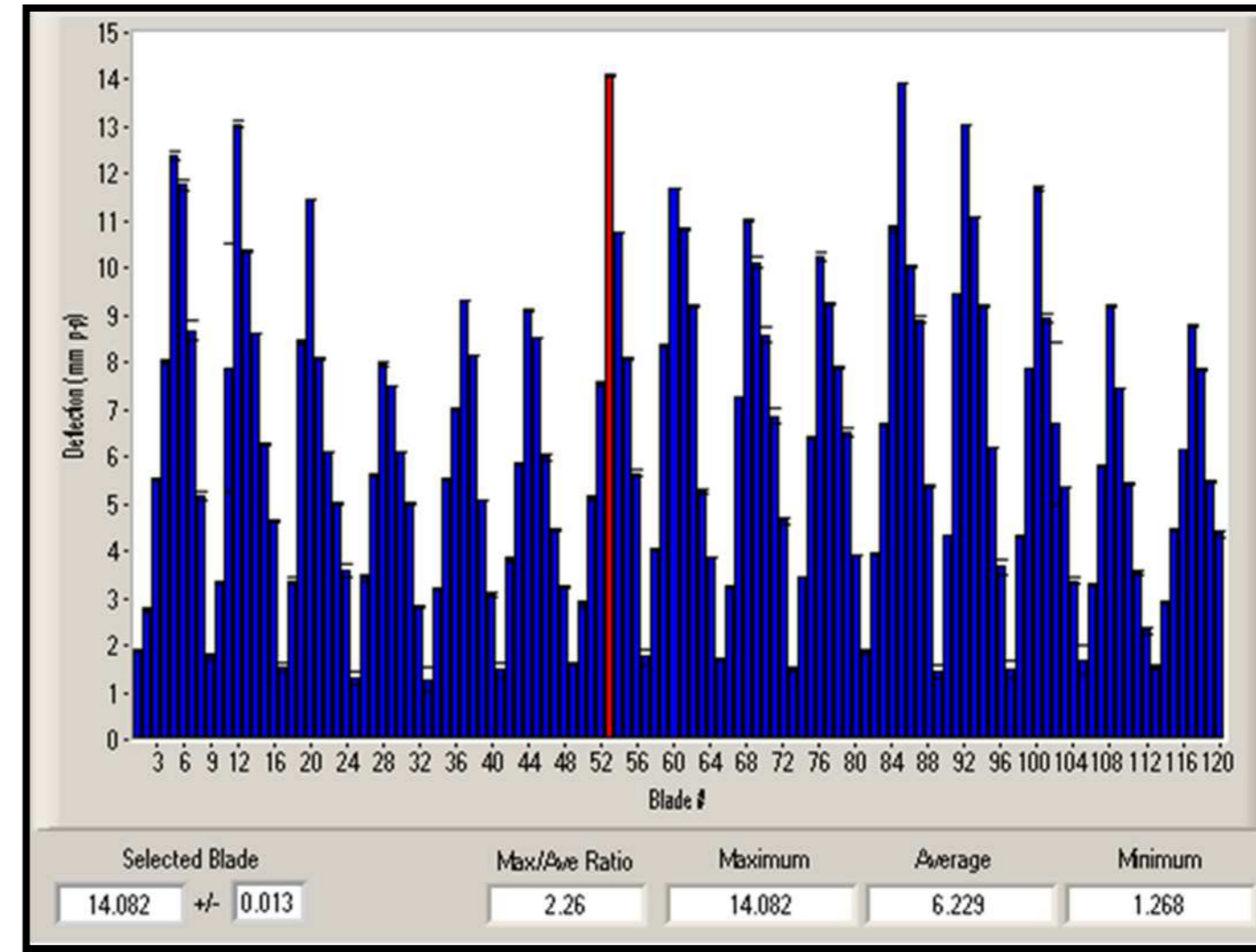
**CASE HISTORY:** Proximity probes have been mounted on the turbine exhaust casing in correspondence of the last stage. Tip Timing measurement system (provided by Agilis) has been installed. Turbine operation has been brought in critical situation for exciting vibrations. Apparent vibrations have been recorded during the run up. Time history of one blade where maximum vibration amplitudes occurred shows non synchronous vibrations at 217 Hz corresponding to the X-mode). Being the vibration non-synchronous, the phase is random and meaningless. The X-mode showed unstable behaviour with high vibration amplitudes. The amplitude distribution along the 120 blades, which shows maximum amplitude every 8 blades confirms the X-mode excitation.



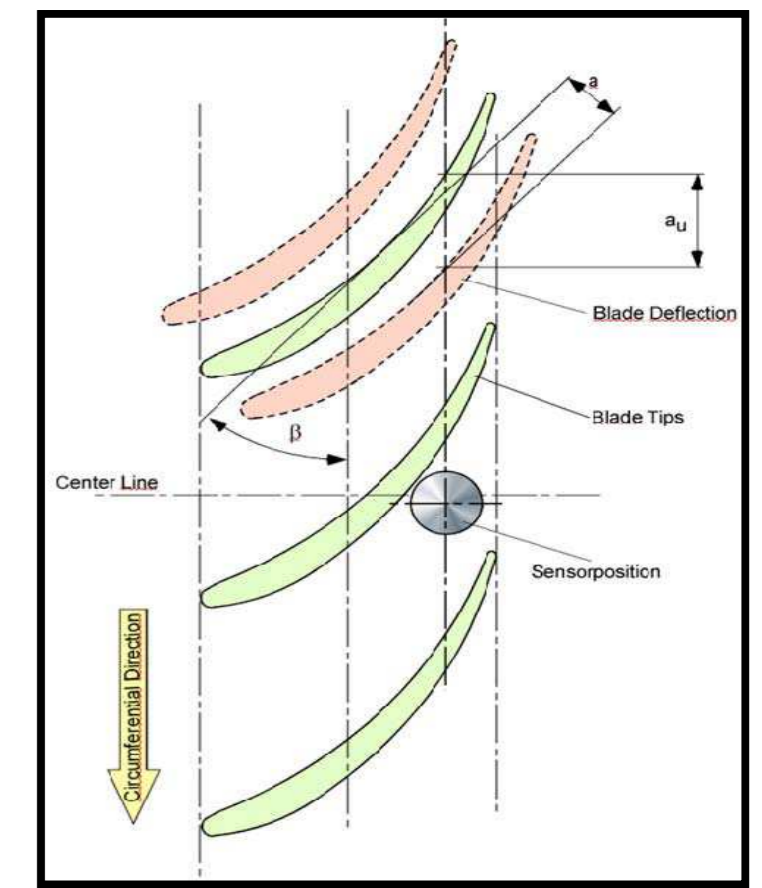
Proximity probes installed on the turbine exhaust casing.



Time history of blade 53 at critical turbine operating condition apparent maximum peak-to-peak amplitude is 14.1 mm.

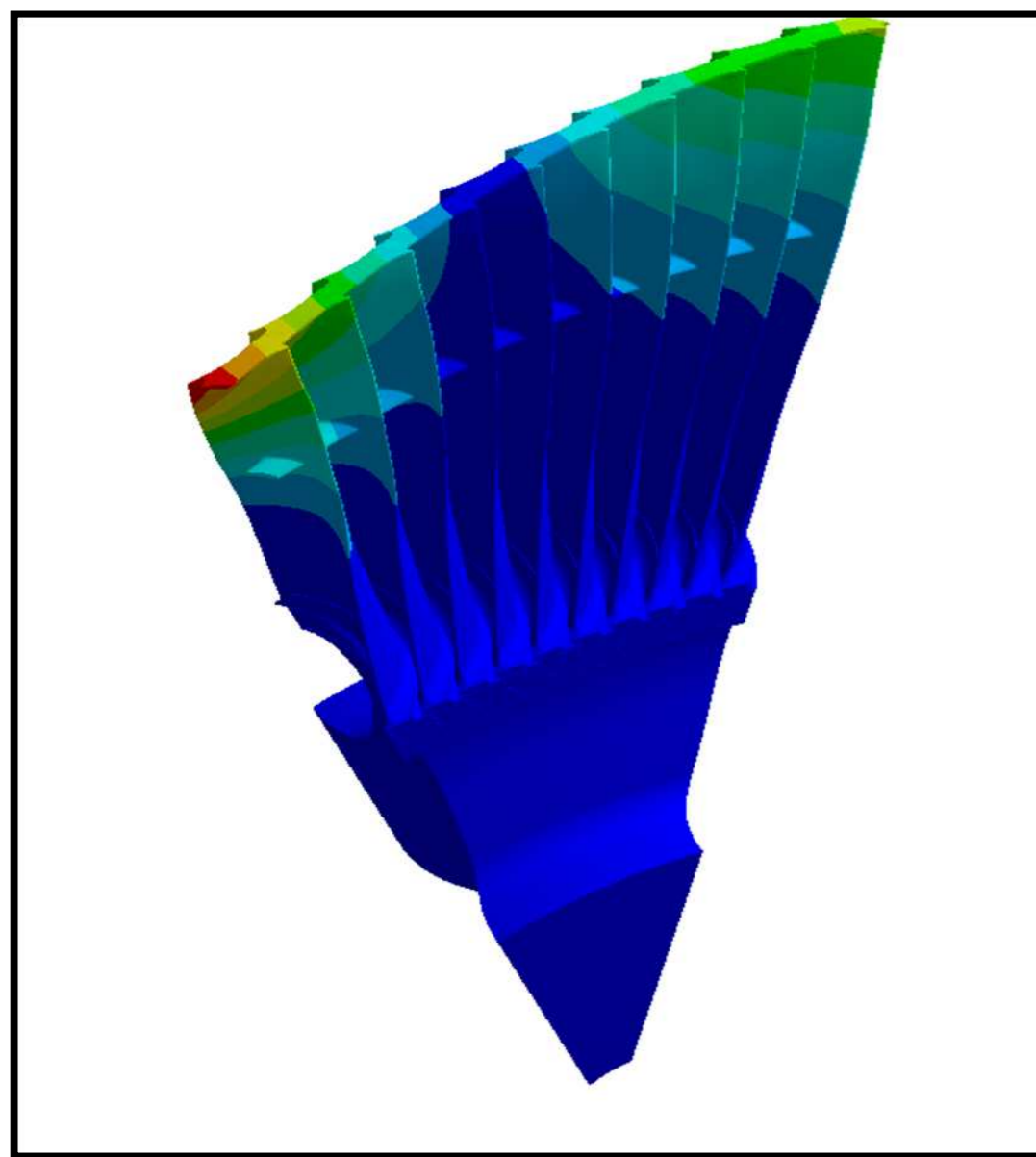


Amplitude distribution along the 120 blades: maximum amplitude every 8 blades confirms the X-mode excitation.

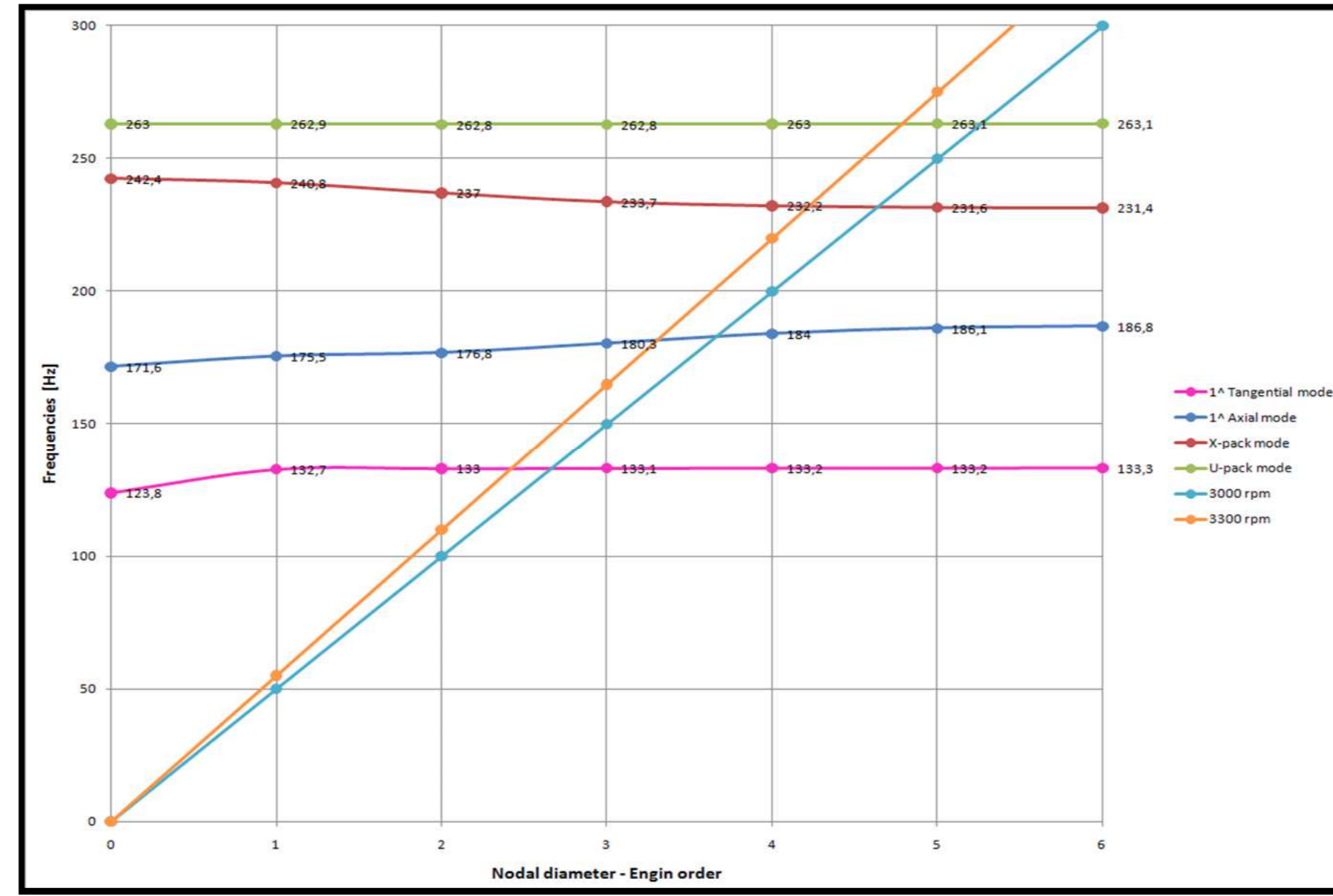


Conversion from apparent vibration in circumferential direction into real vibration according to the vibration direction requires knowledge of vibration mode shape, which must be calculated with the 3D model of the blade pack: real peak-to-peak vibration amplitude is 4.3 mm.

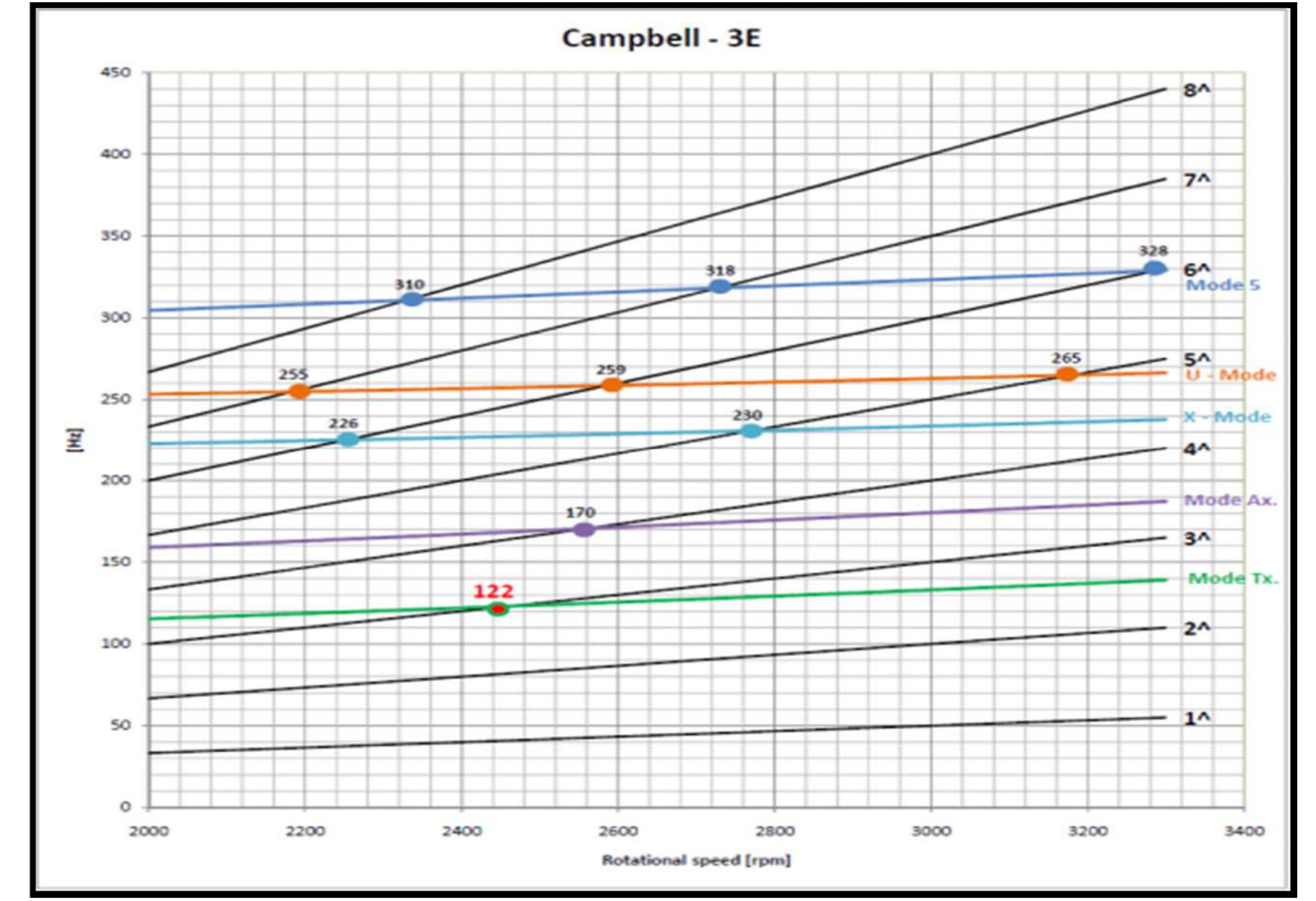
**CASE HISTORY:** the same turbine has been equipped with new blades grouped in packs of 10 blades by welded shrouds. The 3D model of the blade row, performed by ANSYS Workbench, allowed to calculate natural frequencies and check the avoidance of resonant conditions at rated speed, with the different possible engine order excitations. The X mode which was strongly excited by the fluidynamic instability had shifted to 230 Hz. The Campbell diagram related to the 3 nodal diameter modes shows that during the run up transient the first vibration mode in tangential direction can be excited in resonance with the 3rd engine order excitation. The instability cannot be predicted with the linear model which does not consider fluidynamic interactions.



3D model of shrouded 10 blades pack X mode at 230 Hz (ANSYS Workbench).

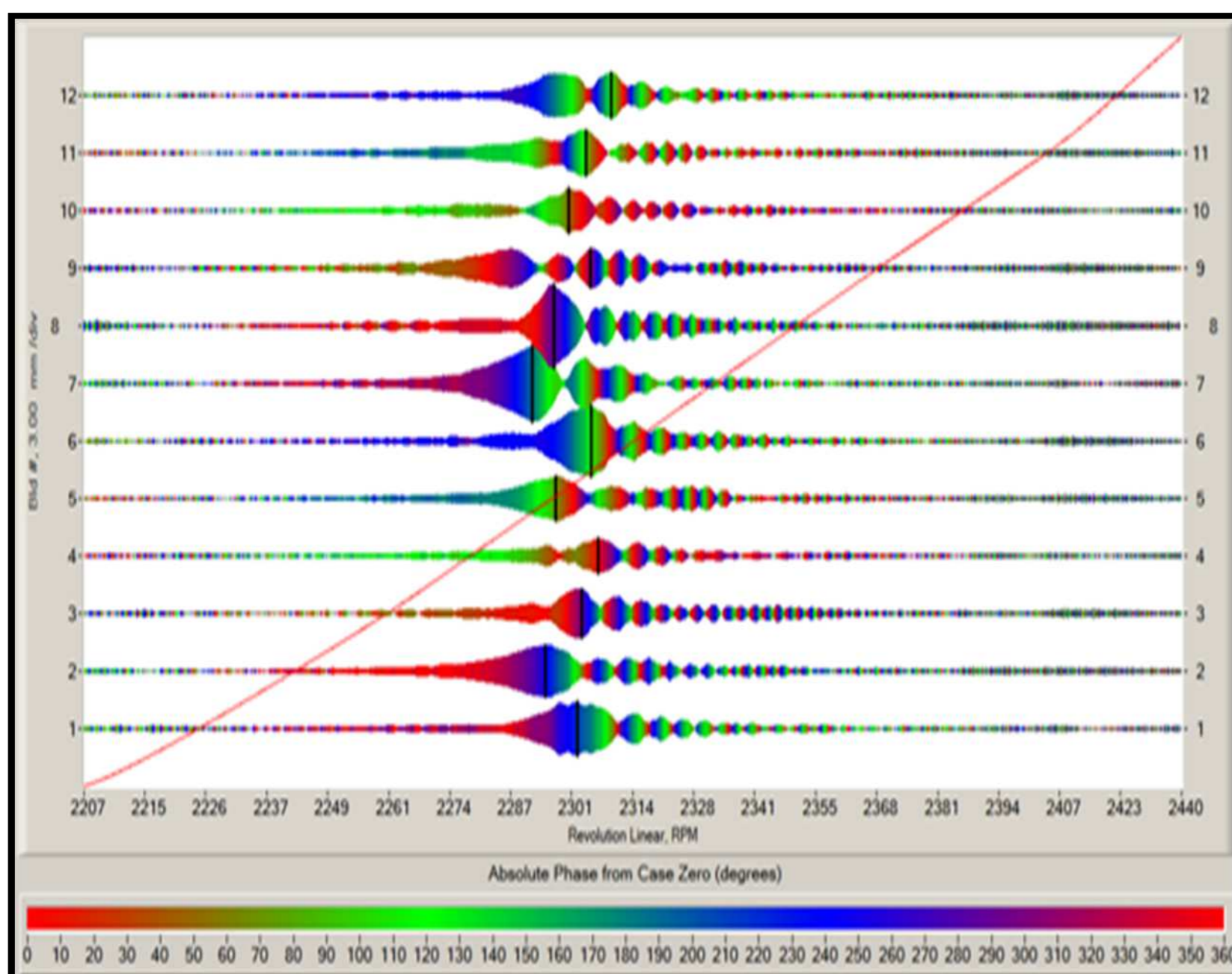


Natural frequencies at rated speed as function of number of nodal diameters (ND) ND of the different families never coincides with the engine order (EO) of excitation: → no resonances!



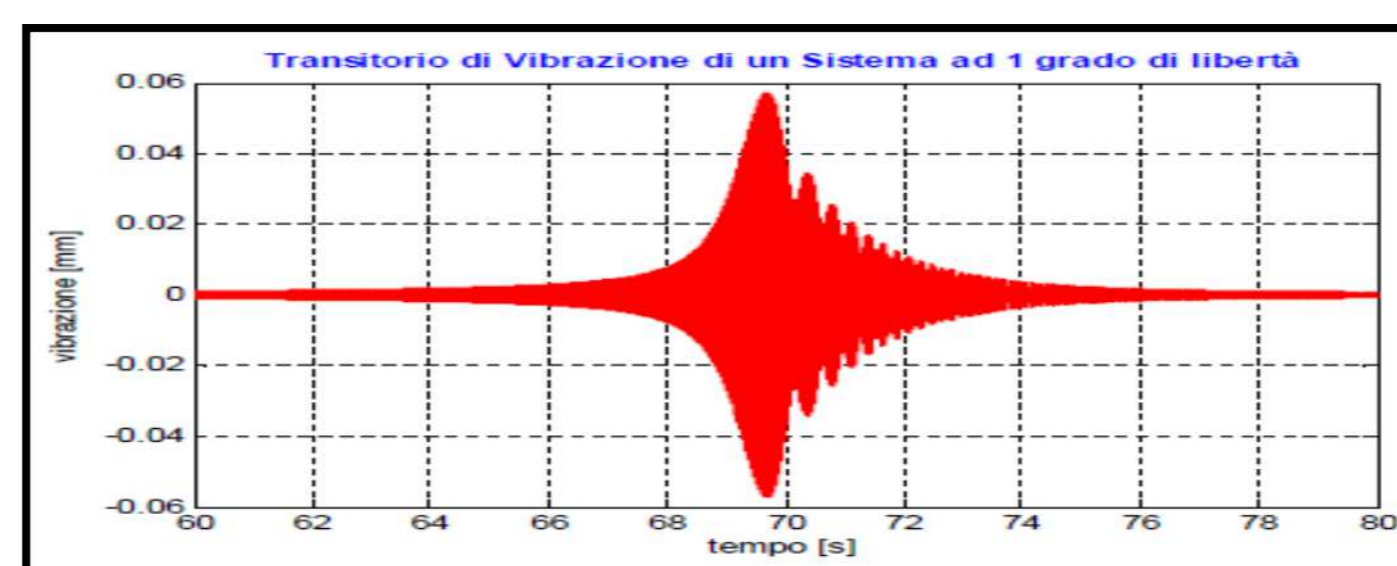
Campbell diagram: natural frequencies of 3 nodal diameters modes showing a possible resonance at 2440 rpm: 3 ND tangential mode has the same frequency (122 Hz) as the 3 EO excitation.

**CASE HISTORY:** The same tip timing measuring system allowed to measure the vibrations, of the first blade of each one of the 12 packs. In this case the apparent vibration coincides with the real vibration because the vibration direction is also the measuring direction. The predicted resonance at 2440 rpm (at 122Hz) occurred at 2300 rpm (at 115 Hz), during the speed rise between 2207 and 2440 rpm which occurred in 7 seconds (with an acceleration of 3.48rad/sec<sup>2</sup>). But the acceleration of the 3rd harmonic component is 3 times this acceleration. The time histories of the 12 packs show typical transient behaviour when resonances are passed with acceleration.

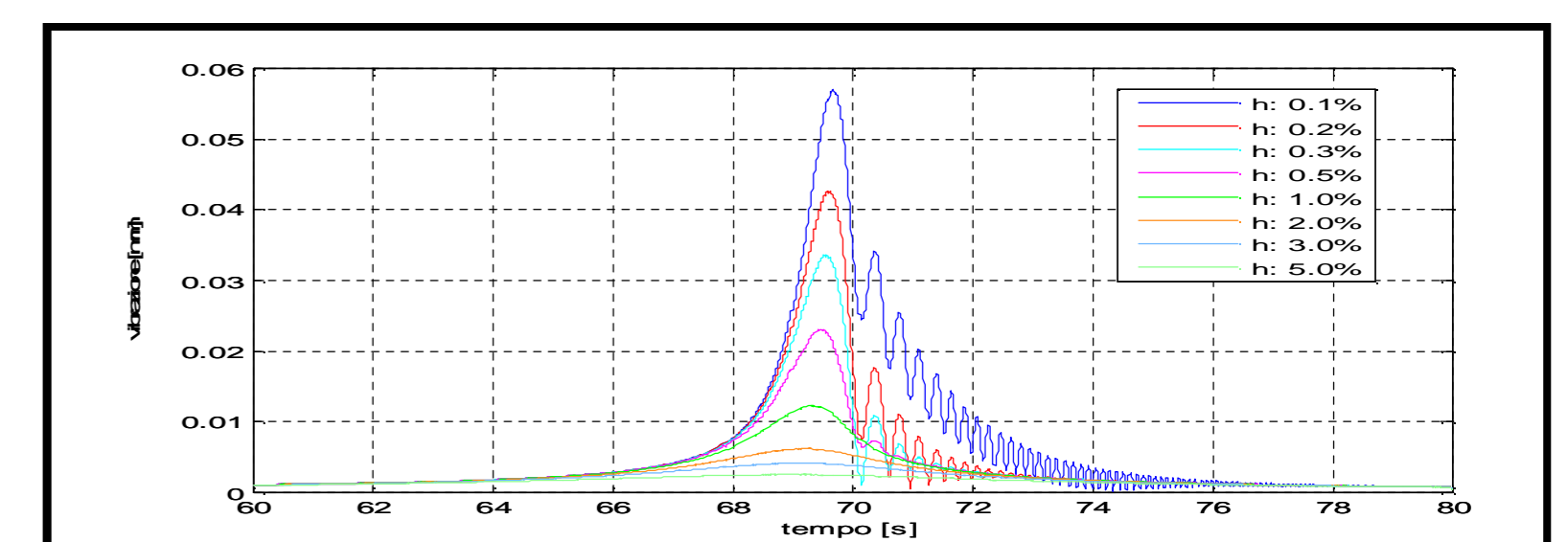


Synchronous vibration time histories showing resonance passing of 12 blade packs Packs have slightly different natural frequencies. Colours indicate gradually changing phases of vibrations. Excitation intensity and damping are slightly different for different packs.

Modal damping ratio identification by means of half power width (based on 1 dof model simulation) is not reliable in the transient condition. The transient situation must be simulated with the simple 1 dof system. This has been done, and the best fitting curve allowed to define the damping ratio of the different packs. Once the damping has been identified also the exciting force can be identified with the simple 1 dof system model.



Single degree of freedom system time history with same natural frequency of blade pack first tangential mode and same acceleration of blade packs during the run up. Assumed damping ratio is 0.1%.



Amplitude envelope curves for different damping ratios: best fitting with experimental envelope curves allows the identification of blade pack damping ratio in its tangential mode.