FLUID–MECHANICAL SYSTEMS SIMULATION

SHAKING ALL OVER

Multiphysics simulation solves a vibration issue in a Francis turbine.

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Courtesy Voith.

Strong vibration and pressure pulsation in hydraulic turbomachinery may be quite harmful to machine performance, longevity and safety. It can cause noise, cracks or even machine failure.

Voith Hydro — one of the world's leading suppliers of hydroelectric equipment, technology and services — observed strong vibrations that had the potential to cause fatigue cracking in the guide vanes of a Francis-type water turbine. In a vertical-shaft Francis turbine, water enters horizontally into a spiral-shaped pipe (spiral casing), which wraps around the circumference of a rotating runner. Stationary guide vanes regulate and direct the water to the periphery of the runner. Inside the runner channels, the

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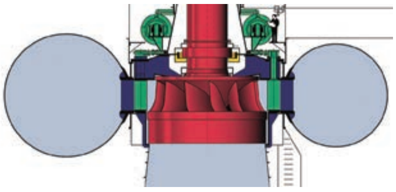
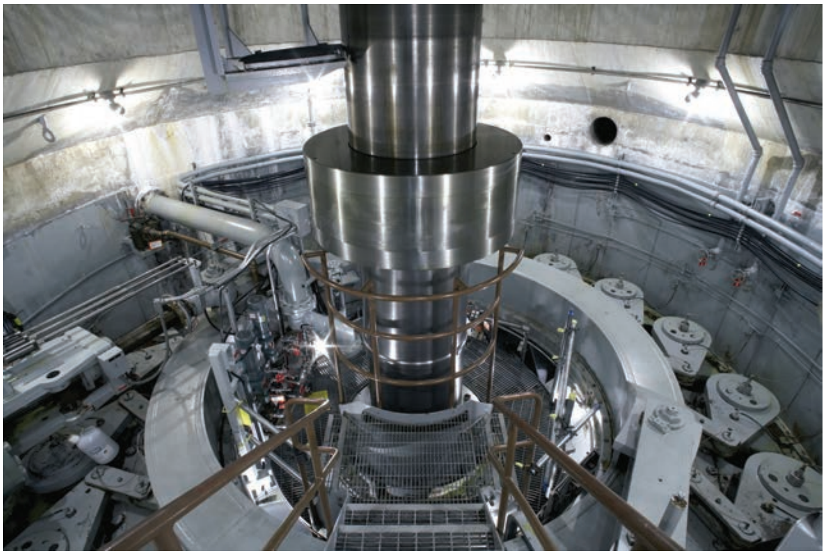
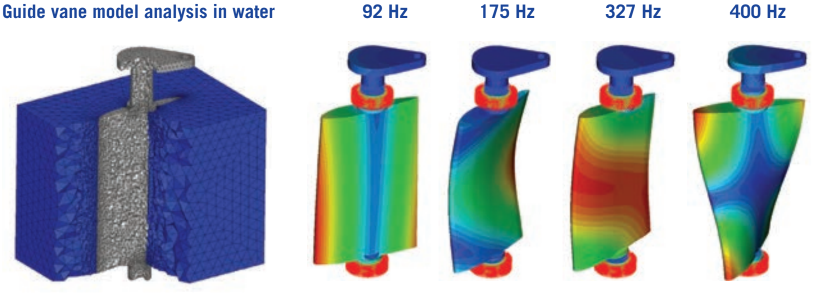


▲ Runner and shaft (red), guide vanes with servo motor (green). *Courtesy Voith*.



▲ Physical measurements of guide vane vibration. *Courtesy Voith*.

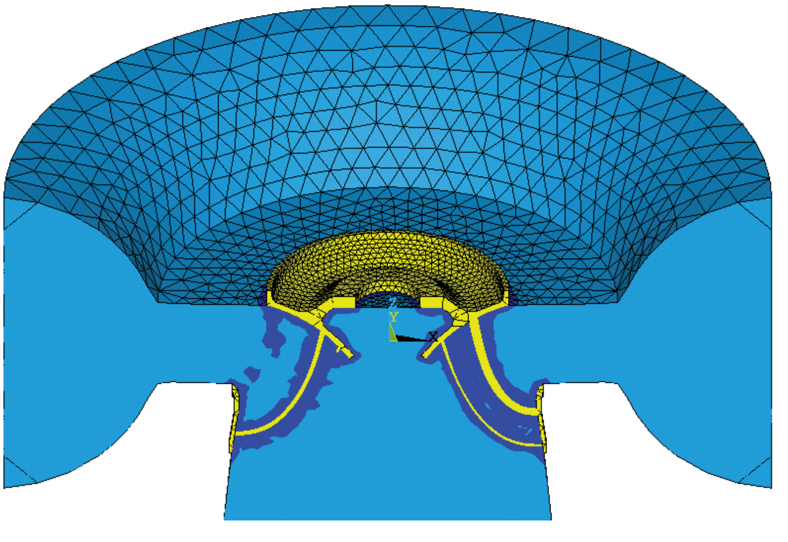


▲ Modal analysis showed that guide vane natural frequencies are far from measured frequencies of vibration. *Courtesy Voith*.

potential energy of the water pressure is transformed into torque, which causes the runner and attached shaft and generator to rotate. Water exits the runner vertically downward into the draft tube where remaining kinetic energy is transformed into additional pressure head.

Using structural simulation, the Voith engineering team ruled out self-excitation and resonance of the guide vanes as the cause of vibration. Employing computational fluid dynamics (CFD), they determined that there was vortex shedding on the runner blades, but not on the guide vanes, that could cause the vibration. This particular machine consists of 24 guide vanes and 13 runner blades; it has an operating speed of 75 rpm. Vibration measurements revealed that all guide vanes vibrated at exactly the same frequencies within the range of 290 Hz to 305 Hz, but it was not possible to perform vibration measurements on the runner blades during operation.

To establish how vortex shedding on the runner was affecting the guide vanes, the team used acoustic fluid–structure interaction with a finite element model of the runner in a water domain. The model used fluid finite elements to couple the dynamic behavior of the runner and water passage. The results proved that excitations at the runner blades' trailing edge were causing the vibration. The simulation matched the measured vibration frequency of approximately 300 Hz. After changing the trailing-edge shape of the prototype runner blades to minimize vortex shedding, observed vibrations were substantially reduced.



▲ Vibro-acoustic finite element model of runner in a simplified water domain. *Courtesy Voith*.

SELF-EXCITED VIBRATIONS AND RESONANCE

To determine the cause of vibration, Voith engineers began by examining the possibility of resonance effects or self-excited vibrations of a guide vane that would occur at a natural frequency. They used ANSYS Mechanical to create a finite element model of the guide vane in water and calculated the first four mode shapes

ANALYZING ACOUSTICS USING ANSYS MECHANICAL
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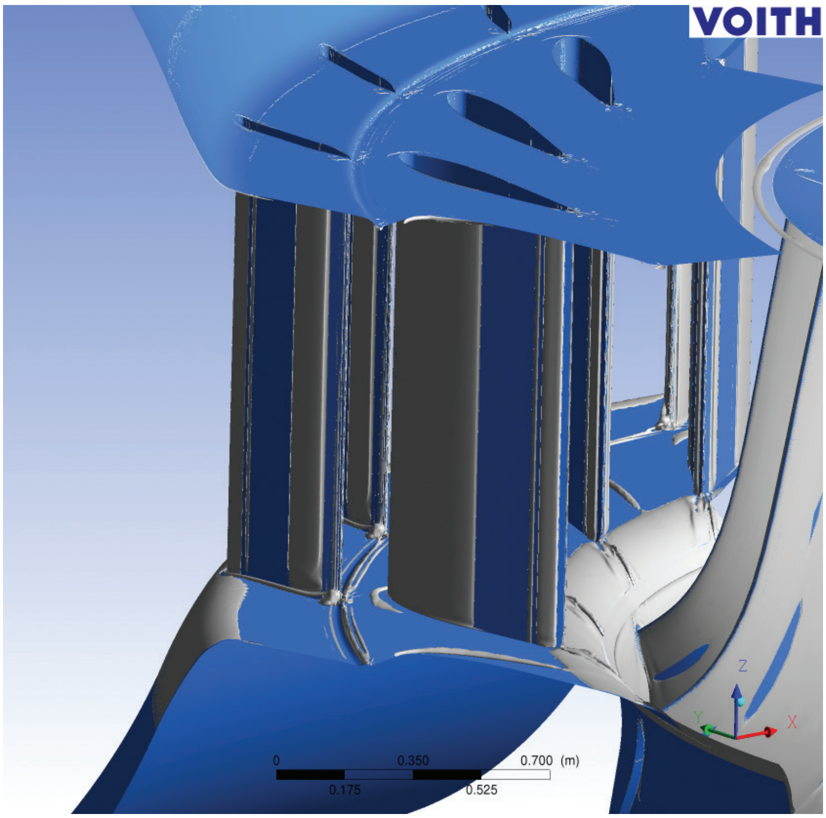
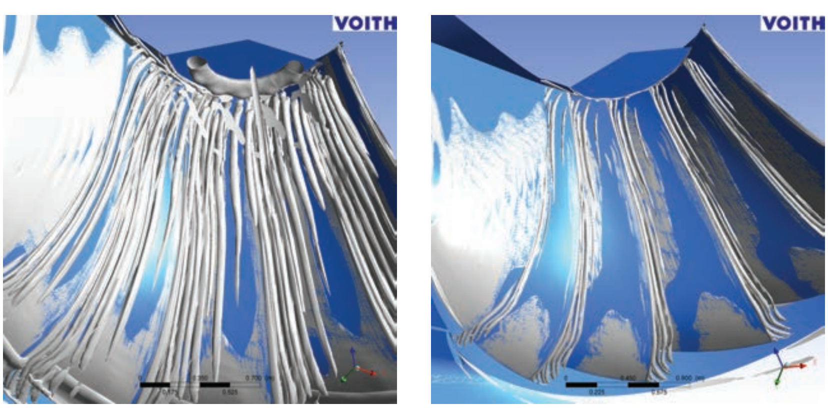
using undamped modal analysis. Engineers found that there were no natural frequencies close to the observed vibrational frequencies, indicating that guide vane resonance or self-excitation was not present. This finding was confirmed by physical measurements that showed all guide vanes vibrated within the same narrow frequency range, even though small differences in geometry and bearing conditions caused each of the guide vanes to have somewhat different natural frequencies.

To determine how vortex shedding on the runner affected the guide vanes, the team used acoustic fluid–structure interaction.



▲ Vortex shedding was not seen around guide vanes by looking at the Q-criterion that visualizes iso-surfaces of the second invariant of the strain-rate tensor, enclosing spatial regions with minimum pressure to identify vortices in a flow field. *Courtesy Voith.*



▲ CFD simulation of runner blades showed vortex shedding, visualized by two different Q-criteria. *Courtesy Voith.*

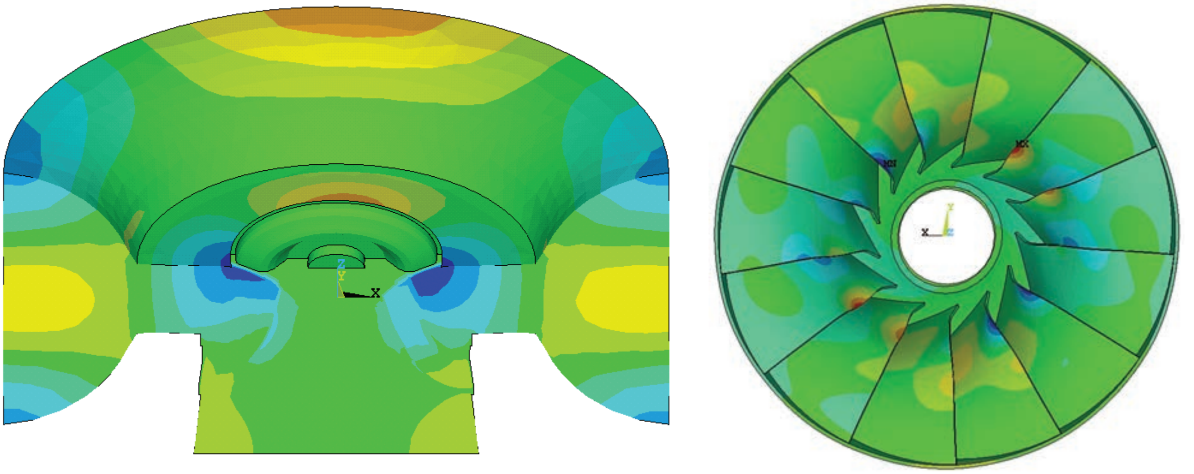
VORTEX SHEDDING

Voith performed unsteady CFD analyses with ANSYS CFX to investigate the possibility of vortex shedding at the guide vanes. The trailing edge used on the guide vanes was designed to prevent vortex shedding, and the analysis showed no sign of shedding. Therefore, the engineers concluded that the problem was not caused by vortex shedding at the guide vanes.

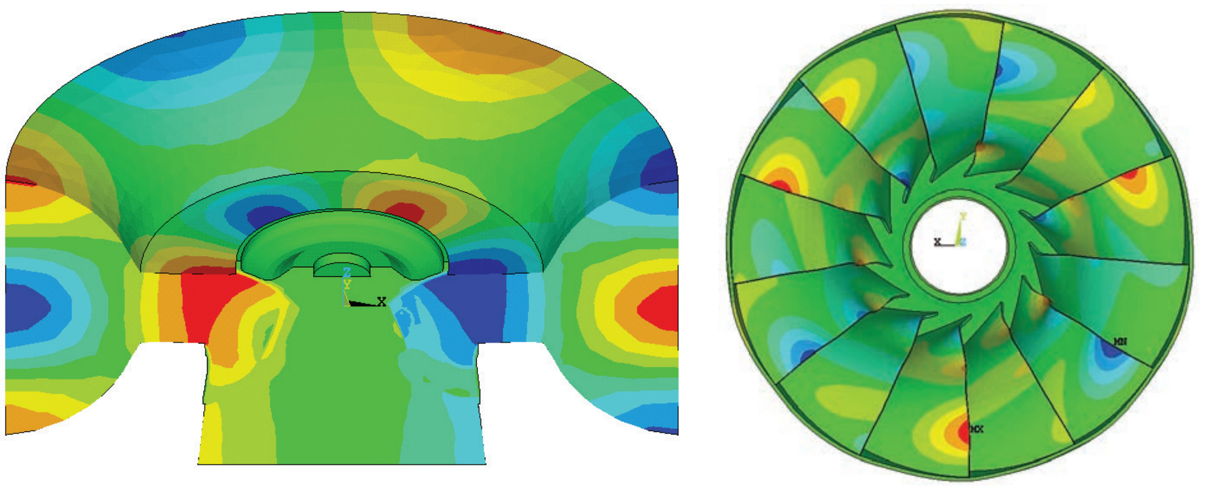
Next, the team performed unsteady CFD analyses at the runner blades. Because the manufactured trailing-edge shape may deviate slightly from the as-designed shape, engineers analyzed both the as-designed chamfered edge as well as a blunt trailing edge. Vortex shedding was clearly observed around 220 Hz for the blunt edge and 370 Hz for the chamfered trailing edge. For a rigid runner, vortex shedding frequencies at different blades and along the trailing edge of a single blade typically differ despite the fact that all of the guide vanes vibrate at the same frequency. The reason is that if some natural frequencies of the mounted runner in water are located in the frequency range of the vortex shedding, and if the corresponding mode shapes include trailing-edge bending, then the vortex shedding frequency may lock in and resonate at this natural frequency. The lock-in effect can cause large amplitude vibrations.

COUPLED DYNAMIC BEHAVIOR

However, vortices that separate from the runner blades move downstream into the draft tube and do not affect guide vanes directly. Thus, even with amplified vortex shedding due to lock-in effects, there must be an additional explanation for the propagation of the pressure pulse in the upstream direction to the guide vanes. Both modal and harmonic response analyses were performed with ANSYS Mechanical to investigate the coupled dynamic behavior of the entire runner and water passage using a vibro-acoustic model of the runner in a simplified water domain created using fluid elements. The finite element model included a rotating frame of reference of the runner and a simplified model of the stationary parts with full rotational symmetry. The runner structure was fixed in the axial and circumferential direction at the connection to the shaft,



▲ Pressure field (left) and axial runner displacement (right) of vibro-acoustic mode shape with two diametrical node lines at a natural frequency of 301 Hz. *Courtesy Voith.*



▲ Pressure field (left) and axial runner displacement (right) of vibro-acoustic mode shape with three diametrical node lines at a natural frequency of 325 Hz. *Courtesy Voith.*

ANALYZING VIBRATION WITH ACOUSTIC–STRUCTURAL COUPLING
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and a fluid–structure interface was coupled to the runner structure and acoustic fluid domain. This simplified modal analysis of the undamped vibro-acoustic model provided mode shapes and corresponding natural frequencies. Multiple natural frequencies were detected close to the measured frequency range of the guide vane vibrations. Most of the associated vibro-acoustic mode shapes exhibited large bending displacements at runner-blade trailing edges as well as strong pressure fluctuations in the guide vane area.

Harmonic response analysis was performed to get a clearer picture of the vibro-acoustic effects in the area of the runner and distributor. The runner was excited by rotating force patterns with distinct numbers of diametrical node lines. Each natural frequency has a particular mode shape defined by the number of diametrical node lines. At each runner blade, a single force acts on the trailing edge perpendicular to the blade surface. The results revealed vibro-acoustic resonances with large bending displacements and high pressure pulsations. Both pressure and displacement criteria exhibited clear resonance peaks at 295 MHz for a mode shape with three diametrical node lines and 306 Hz for a mode shape with seven diametrical node lines, which is close to the measured vibration.

The results of the harmonic response analysis together with modal analysis indicate that lock-in effects based on coupled vibro-acoustic resonance conditions synchronize and amplify vortex shedding. The corresponding vibro-acoustic mode shapes propagate and amplify pressure pulsations within the rotating and stationary components of the turbine. The pressure pulsations cause synchronized guide vane vibrations at the natural frequencies of vibro-acoustic mode shapes. The problem was solved by a modified trailing-edge shape that minimized and de-tuned vortex shedding at the runner blades, substantially reducing the guide vane vibrations.

Determining and solving this vibration issue may not have been possible using a single physics. It required understanding all physics involved and applying them appropriately to the problem at hand.

Reference

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