The role of fluid-structure interaction for safety and life time prediction in hydraulic machinery

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Abstract. Reliable assessments of dynamic phenomena in hydraulic machinery require the consideration of fluid-structure interaction (FSI) if natural frequencies or damping properties of submerged structural parts clearly change due to the surrounding water or if the interaction may cause instabilities. Exemplarily, three different applications of strongly coupled FSI are presented which all have a high impact on operational safety or dynamic stresses and fatigue life. At first, flow induced damping effects at a Francis runner exhibit a strong dependency on operating condition and mode shape. Secondly, if a rotating disc is submerged, a splitting of natural frequencies is observed for mode shapes with different spinning direction. Finally, a hydroelastic instability is identified in a bypass valve at small opening.

1. Introduction
Recent trends of customer and market requirements lead to special challenges with respect to the integrity of components in hydraulic machinery. Wide operating ranges including rough operating conditions, thin profiles due to high performance guarantees, weight optimization, and frequent changes in operating conditions cause high demands on vibration behavior and fatigue life, see References [1-3]. However, reliable fatigue life assessments during the design phase require accurate predictions of dynamic stresses beside safe assumptions for the load universe. Hence, calibrated numerical simulation procedures for analyzing dynamic phenomena in hydraulic machinery have to apply, as shown in References [4-5]. Since the interaction of structural vibrations and hydrodynamics strongly affects natural frequencies and damping effects or even may cause hydroelastic instabilities, fluid-structure interaction (FSI) has to be an integral part of such simulation procedures.

Therefore, FSI plays an important role in design optimizations if operational stability and fatigue life are critical constraints. Since numerical FSI simulations are still quite costly, see Ref. [6], solution strategies have to be adapted precisely to a given application in order to minimize modeling effort and calculation time, see Ref. [7]. On the one hand, automated and fast procedures including simplified FSI approaches apply during the design phase. On the other hand, complex simulations are performed to derive design rules, to calibrate simplified approaches, or to investigate special phenomena.

The present contribution focuses on three different applications of complex FSI analyses. In Section 2, flow induced damping effects are investigated at a Francis runner for different operating conditions and vibration mode shapes. Section 3 deals with free vibrations of a submerged rotating disc and the influence of the rotation on natural frequencies. In the resonance vicinity, accurate frequency and damping predictions are a prerequisite for dynamic stress calculations and reliable fatigue life evaluations. Finally, Section 4 regards a hydroelastic instability in a bypass valve at small opening. A sufficient safety distance to the stability limit is crucial for operational safety since instabilities may cause strong vibrations, cracks and noise or even the immediate damage of the structure, see Ref. [8].
2. Hydrodynamic damping at a Francis runner

In the resonance vicinity, amplitudes of forced vibrations strongly depend on the damping behavior, and hence, accurate damping assumptions are a prerequisite for dynamic stress analyses and reliable fatigue life assessments. In Francis turbine runners, main sources of periodic excitation forces are rotor-stator interaction (RSI), part load vortex rope and v. Karman vortex shedding. While part load vortex excitation is located far below runner resonances and vortex shedding can be circumvented by an appropriate trailing edge design, RSI occurs in the range of runner resonance frequencies. And especially in modernization projects or trouble shooting tasks, a sufficient distance to resonance is not always achievable. Hence, reliable estimations for the hydrodynamic damping are necessary. Material or structural damping is much smaller and may be neglected. Up to now, damping assumptions are derived from strain gauge measurements at prototypes or special laboratory experiments, see Ref. [9]. However, the extrapolation of those measurement results to new runner designs leads to quite high inaccuracies if the influence of vibration mode shapes and flow conditions at different operating points is not fully understood. Therefore, in this section, strongly coupled FSI analyses apply to a 15 bladed Francis runner in prototype scale in order to investigate hydrodynamic damping effects depending on mode shape and operating condition.

2.1. Numerical modeling

In the FSI simulation, unsteady CFD is coupled to linear structural dynamics in generalized modal coordinates. Hence, an undamped modal analysis of the runner structure is performed to get the mode shapes which are spanning a reduced modal space. The corresponding finite element model of the 15 bladed Francis runner is given in Figure 1. Axial and circumferential displacements are fixed at the connection to the shaft.

For the CFD analysis in prototype scale, only a rotating fluid domain is regarded consisting of extended inlet area, runner channels, and draft tube cone. The mesh with about 13 million tetrahedron elements leads to an average $y$-plus value of approx. 300, which is acceptable since boundary layer effects have little influence on the hydrodynamic damping. For turbulence modeling, the Spalart-Allmaras model applies. At the inlet, radial and circumferential velocity components are prescribed, and at the outlet, zero pressure is imposed. The time domain solution of fluid and structure including the coupling is performed by the AcuSolve CFD solver, using a single iteration loop for flow equations, turbulence model, mesh movement, and structural dynamics in modal coordinates. A quite efficient solution procedure for the strongly coupled system follows in which natural frequencies of the runner are significantly reduced due to the added mass effect of the surrounding water.

![Figure 1: Finite element model of the runner](image.png)
2.2. Identified damping ratios depending on operating condition and mode shape

Hydrodynamic damping is compared for mode no. 4 with \( n = 2 \) and for mode no. 17 with \( n = 7 \), where \( n \) denotes the number of diametrical node lines. The mode shapes are shown in Figure 2, and corresponding natural frequencies in air \( f_{\text{air}} \) are given in Table 1. Both modes interact with flow conditions at full load and at part load operation, respectively. The different flow conditions are visualized by streamlines in Figure 3. For each FSI analysis, only the regarded mode shape is projected onto the CFD mesh assuming that flow effects only change frequency and damping but not the shape of the mode. The corresponding modal coordinate is perturbed in the beginning of each simulation. From the decaying time history of the modal coordinate, the damped natural frequency in water \( f_{\text{fsi}} \) and the modal damping ratio are identified using the logarithmic decrement method.

![Figure 2: Axial displacement of mode no. 4 with \( n = 2 \) (left) and mode no. 17 with \( n = 7 \) (right)](image)

![Figure 3: Streamlines at full load operation (left) and part load operation (right)](image)

Due to the added-mass effect, the natural frequencies of both modes are clearly reduced, from 77 to 57 Hz for \( n = 2 \) and from 153 to 60 Hz for \( n = 7 \). The out-of-phase vibration of adjacent blades for \( n = 7 \) causes a much stronger added-mass effect, while the neglected effect of the water in side chambers and labyrinths would lead to an additional frequency reduction, but mainly for \( n = 2 \) with clear deflection of band and crown. In contrast to the frequency, the damping is strongly affected not only by the mode shape but also by the flow condition. As known from generic investigations at a profile, see Ref. [7], hydrodynamic damping depends linearly on the flow velocity. Hence, part load operation with reduced flow velocities especially in the region of large blade deflections leads to much smaller damping ratios. However, the influence of the mode shape on the damping is even higher. The reason is that hydrodynamic damping is caused by inertia effects of the water which are phase-delayed due to the presence of convective flow. Therefore, mode no. 17 with \( n = 7 \) exhibits much higher damping since the added-mass effect is much higher, too. Table 1 summarizes the results in terms of frequencies and relative damping ratios, which are given as percentage of the case with highest damping. The results clearly indicate that a single design value is insufficient to estimate hydrodynamic damping at turbine runners. For reliable dynamic analyses in the resonance vicinity, damping effects may be identified by fluid-structure interaction depending on geometry, vibration shape, and flow condition.
3. Free vibrations of a submerged rotating disc

For smooth operation of rotating machines, natural frequencies should be sufficiently separated from excitation frequencies. Especially natural frequencies of modes with distinct diametrical node lines are of interest because such shapes may match typical excitation sources like rotor-stator interaction. In order to determine natural frequencies of submerged components which rotate, the surrounding water and the casing have to be considered. The added mass of the water causes a decrease of natural frequencies, but the flow which is induced by the rotation may change natural frequencies, too.

3.1. Numerical modeling

In order to investigate the basic principles which govern flow effects on rotating runners, a simplified model is considered, namely a disc. This simplification is adequate since crown and band of high head Francis and pump turbine runners are similar to discs. In detail, free vibrations of a rotating disc located in a water filled container are analyzed. The geometry of disc and container is shown in Figure 4 (left). The disc with a diameter of about 0.4m is connected to a shaft. The shaft is considered up to the top side of the container where displacements are fixed. At the inner boundaries of the container, no-slip wall conditions apply for the fluid, and at the surfaces of disc and shaft which are connected to the water, fluid-structure interfaces are considered. Both domains are described in a rotating frame of reference. The coupled system is solved by two-way FSI simulations in time domain using ANSYS Mechanical for structural dynamics and ANSYS CFX for the fluid. If a mode shape with \( n \) diametrical node lines shall be excited, \( 2n \) forces act on the disc for a short time and cause an initial deformation, as sketched in Figure 4 (right) exemplarily for a mode shape with \( n=2 \). Subsequently, free vibrations of the disc are observed.

![Figure 4](image.png)

The fluid domain is discretized with approx. 5 million cells, and the SST model is chosen for turbulence modeling. Depending on the excitation mode shape and the corresponding frequency to be resolved, a minimum time step of 2.5e-5 seconds and up to 12 coupling iterations per time step are required to reach a stable and sufficiently converged solution. A calculation time of about 4 days follows to analyze a single mode shape on a cluster with 64 cores.

Table 1: Summary of runner frequencies and hydrodynamic damping identified by FSI simulations

| mode | node lines | \( f_{\text{nat}} \) [Hz] | operation   | \( f_{\text{fsi}} \) [Hz] | rel. damping ratio [%] |
|------|------------|----------------|-------------|----------------|-----------------------|
| 4    | \( n = 2 \) | 77            | part load   | 57            | 17                    |
|      |            |               | full load   | 57            | 26                    |
| 17   | \( n = 7 \) | 153           | part load   | 60            | 58                    |
|      |            |               | full load   | 60            | 100                   |

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3.2. Effect of mode rotation on natural frequencies

If the disc rests in the water without rotation, the excitation of a single mode shape leads to free vibrations at the corresponding natural frequency in water. Vibration nodes and locations with maximum amplitudes stay at a fixed place during the entire simulation. However, if a rotational speed of 480 rpm applies, the vibration behavior of the disc clearly changes. In the rotating frame of reference, vibration nodes and locations of maximum amplitudes rotate slowly against the rotational direction of the disc. This motion can be described by two modes rotating in opposite directions with different frequencies of which one is slightly lower and the other slightly higher than the natural frequency at rest. As shown in Ref. [10], the rotating natural frequencies depend on the natural frequency at rest $f_{n,0}$ and on the rotational speed $f_{rot}$ of the disc which causes a frequency shift $\Delta f_n$. Together, the rotating natural frequencies can be expressed by

$$f_{\pm n} = f_{n,0} \pm \Delta f_n(f_{rot})$$

which is valid if the rotational speed is small compared to the natural frequency at rest. In Eq. (1), the frequency $f_{\pm n}$ is lower than the frequency at rest and the corresponding mode shape rotates in direction of the disc. In contrast, the mode shape with the higher frequency $f_n$ rotates against the direction of the disc. The FSI simulation results for mode shapes with $n=2$ and $n=3$ are summarized in Table 2. The calculated natural frequencies are in good agreement with experimental results, see Ref. [11].

| mode shape | $f_{ua}$ [Hz] | $f_{n,0}$ [Hz] | $f_{sa}$ [Hz] | $f_{ua}$ [Hz] |
|------------|----------------|----------------|----------------|----------------|
| $n = 2$    | 267.0          | 158.4          | 152.8          | 164.0          |
| $n = 3$    | 589.7          | 393.4          | 386.1          | 400.7          |

Table 2: Natural frequencies of the submerged disc at rest and at 480 rpm

4. Self-excited vibrations in a bypass valve

In high head turbines, spherical valves often feature a bypass to reduce the pressure difference prior to the opening if the hydraulic system behind the spherical valve is depressurized. The flow through the bypass is controlled by a bypass valve. In the present case, such a bypass valve did not work properly but remained at a small opening due to problems with the servomotor, causing a high speed flow through narrow gaps while the gaps may vary due to elastic deformations. This is a typical example of self-excited vibrations caused by the interaction of gap flow and structural motion. In these vibrations, the gap distances on opposite sides vary opposite in phase, leading to flow conditions which also vary opposite in phase but with some phase shift to the gap change. The resulting hydroelastic instability is characterized by high amplitude vibrations of the entire valve and even the pipe system.

In order to investigate this practical example of complex FSI by means of numerical simulations, strong coupling of unsteady CFD and linear structural dynamics in generalized modal coordinates apply. The results demonstrate that state-of-the-art numerical simulation techniques are able to predict reliably hydroelastic instabilities in real hydraulic systems or components, offering engineers the ability to optimize designs and to ensure stable operation even at critical operating conditions.

4.1. Numerical modeling

The structural vibration behavior of the valve is described by a linearized finite element model, shown in Figure 5. The model consists of plunger (red), bushing (yellow), casing (blue), and a simplified inlet pipe (green) which is shorter but more flexible than the real pipe system. The non-linear behavior of the bushing is linearized by means of an adapted (reduced) Young’s modulus. An undamped modal analysis of the structure leads to uncoupled mode shapes allowing the transformation of the dynamic system into a reduced modal space. For the FSI simulation, the first 20 natural mode shapes are projected onto the CFD mesh at the fluid-structure interface.
In order to simulate the water flow through the valve, unsteady RANS equations are solved on an unstructured tetrahedron mesh with 12 prism layers at wall boundaries, see Figure 6, resulting in about 6.3 million elements and a maximum y-plus value of approx. 200. Since this investigation is focussed on FSI effects, possible cavitation behind the gap is neglected, and for turbulence modeling, the comparable simple Spalart-Allmaras model applies. Again, the solution and coupling of fluid and structure is performed by the AcuSolve CFD solver, using a single iteration loop for flow equations, turbulence model, mesh movement, and structural dynamics in modal coordinates. This efficient solution procedure for strongly coupled systems is a prerequisite to perform hydroelastic stability analyses in an acceptable time frame.

4.2. **FSI simulation results**

Two mode shapes (mode no. 7 at 440Hz and no. 10 at 516Hz) are mainly involved in the self-excited vibration. In both modes, a relative motion between plunger and casing is clearly present and hence, strongly deforming gaps appear, see Figure 7, where contour plots of the displacement magnitude are visualized on top of the deformed geometry.
In order to identify the stability limit, the inflow velocity increases slowly during the entire time domain analysis, starting at a value of 3m/s. The pressure difference corresponding to this initial velocity is clearly below the turbine head of approx. 500m, see Figure 8. An initial perturbation is acting on mode no. 7 causing damped vibrations during the first part of the simulation. The stability limit is identified by the time instant and corresponding inflow velocity at which the amplitudes of the free vibration start to increase. This behavior is detected at about 6m/s inflow velocity, corresponding to a head of approx. 600m, see time histories of modal coordinates in Figure 9 (left). Both modes (no. 7 and no. 10) are involved in the coupled system response although only mode no. 7 was initially excited. Due to the added-mass effect, the natural frequencies of both modes are clearly reduced, from 440 to 410Hz and from 516 to 494Hz, see modal amplitude spectra in Figure 9 (right).

In Figure 10, the unsteady flow during a period of the unstable motion is visualized using streamlines and contour plots of the pressure at the gap. Increased and reduced flow rates are alternating between the lower and the upper gap while the gap clearance is alternating too, but with a phase shift being responsible for the energy transfer from flow to the structure.

The identified stability limit of around 6m/s corresponds to a head of approx. 600m which clearly exceeds the real turbine head of approx. 500m. However, considering all the assumptions and simplifications (e.g. unknown gap width, assumed support conditions at inlet and discharge pipe, linearized bushing contact, and neglected cavitation) and having in mind that some numerical damping is present, this is still a good and promising result.

Figure 8: Inlet pipe velocity (left) and pressure (right) during the FSI simulation

Figure 9: Modal displacements against inflow velocity (left) and corresponding amplitude spectra (right)
5. Conclusions

In order to guarantee operational safety and sufficient life time for systems and components under dynamic loading conditions, reliable assessments of dynamic phenomena are required including the accurate prediction of dynamic stresses even in the resonance vicinity. For submerged hydro turbine components like runners, the interaction with the water flow strongly affects the dynamic behavior in terms of frequencies, damping effects and stability characteristics. Hence, fluid-structure interaction has to be considered. This is feasible using advanced simulation technologies and state-of-the-art software tools either for specific applications or for deriving and calibrating simplified models. If simplified FSI models can be included within fast design tools, smooth vibration behavior, sufficient life time and operational safety can be secured during design optimizations of submerged components.

In the present contribution, advanced FSI simulations have been performed to investigate three different phenomena with clear impact on dynamic stresses or operational safety:

- Flow induced damping effects at a Francis runner exhibit a strong dependency on operating condition and mode shape.
- At a rotating disc which is submerged in a closed tank, a splitting of natural frequencies is observed if the corresponding mode shapes exhibit different spinning directions.
- The example of a bypass valve at small opening proves that FSI simulations may apply to identify hydroelastic stability limits.

Figure 10: Variation of streamlines and gap pressure during a period of the unstable response
References

[1] Seidel U, Mende C, Hübner B, Weber W, Otto A, 2014, Dynamic loads in Francis runners and their impact on fatigue life, *27th IAHR Symposium on Hydraulic Machinery and Systems*, Montreal.

[2] Weber W, Mende C, Koutnik J, 2013, Advanced fatigue analysis for transient operating conditions of Francis turbines, *5th IAHR International Meeting of the Workgroup on Cavitation and Dynamic Problems in Hydraulic Machinery and Systems*, Lausanne.

[3] Avellan F, Etter S, Gummer JH, Seidel U, 2000, Dynamic pressure measurements on a model turbine runner and their use in preventing runner fatigue failure, *20th IAHR Symposium on Hydraulic Machinery and Systems*, Charlotte.

[4] Seidel U, Hübner B, Löfflad J, Faigle P, 2012, Evaluation of RSI-induced stresses in Francis runners, *26th IAHR Symposium on Hydraulic Machinery and Systems*, Beijing.

[5] Flemming F, Fisher RK, 2009, Application of unsteady CFD to assess dynamic loads on a Francis runner, *Waterpower 16*, Spokane.

[6] Hübner B, Seidel U, 2007, Partitioned solution to strongly coupled hydroelastic systems arising in hydro turbine design, *2nd IAHR International Meeting of the Workgroup on Cavitation and Dynamic Problems in Hydraulic Machinery and Systems*, Timisoara.

[7] Hübner B, Seidel U, Koutnik J, 2012, Assessing the dynamics of turbine components using advanced fluid-structure interaction, *HYDRO 2012*, Bilbao.

[8] Hübner B, Seidel U, Koutnik J, 2008, Analysis of flow-induced vibrations of a slide gate chain using monolithic fluid-structure coupling, *9th International Conference on Flow-Induced Vibration*, Prague.

[9] Roth S, Calmon M, Farhat M, Münch C, Hübner B, Avellan F, 2009, Hydrodynamic damping identification from an impulse response of a vibrating blade, *3rd IAHR International Meeting of the Workgroup on Cavitation and Dynamic Problems in Hydraulic Machinery and Systems*, Brno.

[10] Weber W, Seidel U, 2015, Analysis of natural frequencies of disc-like structures in water environment by coupled fluid-structure-interaction simulation, *6th IAHR International Meeting of the Workgroup on Cavitation and Dynamic Problems in Hydraulic Machinery and Systems*, Ljubljana.

[11] Presas A, Egusquiza E, Valero C, Valentin D, Seidel U, 2014, Feasibility to use PZT actuators to study the dynamic behavior of a rotating disk due to rotor-stator interaction, *Sensors*, Vol. 14(7), pp. 11919-11942.