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Application of fluid-structure coupling to predict the dynamic behavior of turbine components

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Abstract. In hydro turbine design, fluid-structure interaction (FSI) may play an important role. Examples are flow induced inertia and damping effects, vortex induced vibrations in the lock-in vicinity, or hydroelastic instabilities of flows in deforming gaps (e.g. labyrinth seals). In contrast to aeroelasticity, hydroelastic systems require strongly (iteratively) coupled or even monolithic solution procedures, since the fluid mass which is moving with the structure (added-mass effect) is much higher and changes the dynamic behavior of submerged structures considerably. Depending on the mode shape, natural frequencies of a turbine runner in water may be reduced to less than 50% of the corresponding frequencies in air, and flow induced damping effects may become one or two orders of magnitude higher than structural damping. In order to reduce modeling effort and calculation time, the solution strategy has to be adapted precisely to a given application. Hence, depending on the problem to solve, different approximations may apply. Examples are the calculation of natural frequencies and response spectra in water using an acoustic fluid formulation, the determination of flow induced damping effects by means of partitioned FSI including complex turbulent flows, and the identification of hydroelastic instabilities using monolithic coupling of non-linear structural dynamics and water flow.

1. Introduction

Hydro turbine runners and components have been designed and built by Voith Hydro for more than 100 years. Even though this product has a long tradition, recent trends of customer and market requirements lead to special challenges regarding the integrity of runners. Wide operating ranges including rough operating conditions, thin profiles due to high performance guarantees, weight optimization and frequent changes in operating conditions cause additional demands on vibration behavior and fatigue life (e.g. Avellan et al. [1]). In order to predict fatigue characteristics of hydro turbine runners during the design phase, the accurate determination of the dynamic response including dynamic stresses by means of numerical simulations is a prerequisite.

In hydroelastic systems (interactions of water flow and elastic structures), the surrounding water has a strong impact on the dynamic behavior of the structure and must not be neglected even at low flow rates. The fluid mass which is moving with the structure (added-mass effect), may reduce natural frequencies of submerged turbine components to less than 50% of the corresponding frequencies in air (e.g. Rodriguez et al. [2]), and flow induced damping effects may become one or two orders of magnitude higher than the structural damping behavior (e.g. Roth et al. [3]). Von Karman vortex shedding at elastic blades may lead to lock-in phenomena if shedding frequencies are close to natural frequencies resulting in large amplitude vibrations and fatigue problems (e.g. Ausoni [4]). Even more dangerous are hydroelastic instabilities due to self-excited vibrations of flows in deforming gaps (e.g. labyrinth seals, nearly closed guide vanes or Kaplan blades) and flows along or inside of slender structures which induce rapidly increasing amplitudes usually destroying the structure in short time (e.g. Païdoussis [5], Hübner et al. [6]).

In order to reduce modeling effort and calculation time of numerical simulations of hydroelastic systems, the solution strategy has to be adapted precisely to a given application. While the added-mass effect of surrounding water can be modeled with sufficient accuracy by an acoustic fluid approach, the consideration of hydrodynamic

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damping effects and the identification of hydroelastic instabilities requires more realistic descriptions of the fluid flow using at least potential flow theory or even full Navier-Stokes flow including turbulence modeling. In contrast to aeroelasticity (e.g. Farhat et al. [7]) where loose coupling schemes of different solvers are quite common, hydroelastic systems require strongly (iteratively) coupled solution procedures (e.g. Hübner and Seidel [8]) or even monolithic formulations (e.g. Hübner et al. [9, 10]) in which the coupled system is assembled and solved in a single set of equations. A short explanation of the different coupling procedures is given in Section 2. However, monolithic coupling formulations of complex turbulent flows and structural vibrations are not available because of ill-conditioned equation systems which may hardly be solved by massively parallel iterative solvers used for complex CFD simulations. Thus, depending on the problem to solve, different analysis procedures from monolithic formulations including simplified fluid models up to strongly coupled partitioned solutions of non-linear structures and complex turbulent flows may apply.

In Section 3, different application examples arising in hydro turbine design are presented. Acoustic fluid-structure interaction based on a monolithic formulation is used to determine natural frequencies and mode shapes of a pump turbine impeller in water. The same approach applies to compute the dynamic response and dynamic stresses of a Francis turbine runner due to rotor-stator interaction. In order to determine hydrodynamic damping effects on a thin hydrofoil depending on the flow velocity, two different solution procedures are used. Strong coupling in time domain of the CFD solver ANSYS-CFX and the structural dynamics solver ANSYS-Mechanical is compared to monolithic coupling of potential flow and natural mode shape vibrations in frequency domain. Finally, a university code offering monolithic FSI of viscous fluid dynamics and non-linear structural dynamics based on a unified space-time finite element formulation is used to identify hydroelastic instabilities of a slide gate chain in axial flow.

In general, industrial development processes tend more and more to a virtual product development including comprehensive multi-disciplinary optimization. This requires a good understanding of the underlying coupled physics in order to choose the appropriate modeling and solution technique for a complex multi-disciplinary problem situated somewhere in the space of coupled field approximations, shown in Figure 1, according to Löhner et al. [11]. In this diagram, the disciplines are restricted to computational structural dynamics (CSD), computational fluid dynamics (CFD), and computational thermal dynamics (CTD). By moving away from the origin, computational cost and model preparation time may increase by orders of magnitude. Goal of future developments in simulation based engineering is to have a well suited solution procedure for all essential combinations within the complete space of coupled field approximations enlarged by additional disciplines like electromagnetics, active and passive control or chemical reactions.

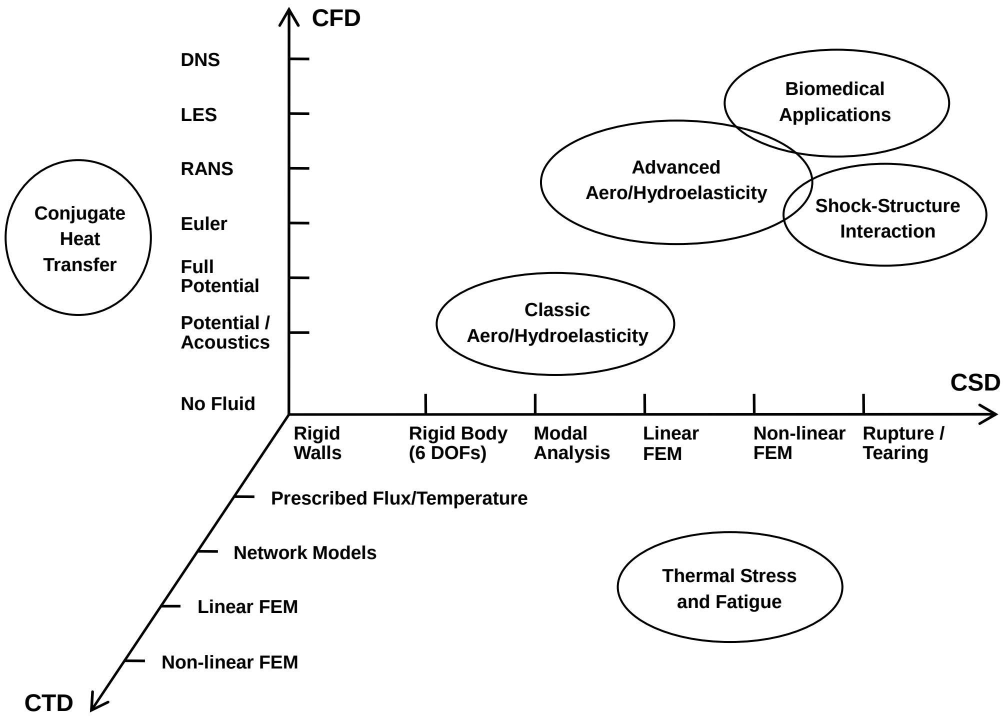

Figure 1: Space of coupled field approximations according to Löhner et al. [11]

2. Theoretical Background

Simplifying the coupled problem to fluid-structure interaction on fixed grids, the coupled set of equations may be written as

$$\begin{bmatrix} \mathbf{K}_{\mathbf{SS}} & \mathbf{K}_{\mathbf{SF}} \\ \mathbf{K}_{\mathbf{FS}} & \mathbf{K}_{\mathbf{FF}} \end{bmatrix} \cdot \begin{bmatrix} \mathbf{x}_{\mathbf{S}} \\ \mathbf{x}_{\mathbf{F}} \end{bmatrix} = \begin{bmatrix} \mathbf{f}_{\mathbf{S}} \\ \mathbf{f}_{\mathbf{F}} \end{bmatrix} \tag{1}$$

where the subscrips F, S denote fluid and structure, respectively. In a monolithic approach, used e.g. in acoustic fluid-structure interaction or by special solvers developed for a limited application area, this equation system is solved as a whole. However, due to different physics and different scales included therein, the equation system is usually ill-conditioned and needs to be solved by direct solvers demanding high computational effort for large models. In contrast, partitioned solution procedures use different (multi-purpose or customized) solvers for fluid and structural dynamics and may be described by the following equations

$$\mathbf{K}_{\mathbf{SS}} \cdot \mathbf{x}_{\mathbf{S}}^{i} = \mathbf{f}_{\mathbf{S}} - \mathbf{K}_{\mathbf{SF}} \cdot \mathbf{x}_{\mathbf{F}}^{i-1} \tag{2a}$$

$$\mathbf{K}_{\mathbf{FF}} \cdot \mathbf{x}_{\mathbf{F}}^{i} = \mathbf{f}_{\mathbf{F}} - \mathbf{K}_{\mathbf{FS}} \cdot \mathbf{x}_{\mathbf{S}}^{i} \tag{2b}$$

which are coupled only via right hand side terms. In a loose coupling scheme both equations are solved only once per time step leading to an explicit solution with a time lag between the sub-system solutions. Consequently, stability characteristics of loosely coupled solutions are quite poor, and especially in hydroelasticity, where the added fluid mass moving with the structure is quite high, converging solutions are hardly obtained. In order to improve stability characteristics of the coupled solution, strong (iterative) coupling algorithms may apply. In each time step, the coupled Equations (2a) and (2b) are solved by a Jacobi type iteration scheme which may be accelerated by appropriate under-relaxation. After this procedure is converged, time accurate coupled solutions may follow, but accuracy order and convergence behavior of the coupled solution depend on whether or not matching time integration schemes are used for solving Equations (2a) and (2b).

3. Application Examples

The following examples of fluid-structure interaction give an overview on applying problem adapted numerical approximations and solution procedures to predict the dynamic behavior of turbine components at Voith Hydro.

3.1 Natural frequencies of a pump turbine impeller in water using acoustic fluid-structure interaction

In order to compare natural frequencies and mode shapes of a pump turbine impeller in air and water, acoustic fluid-structure interaction based on a monolithic finite element formulation applies. The regarded impeller is situated in the middle of a cylindrical water tank with comparable large distances to all boundaries and without displacement constraints. Thus, effects from shaft mounting as well as small distances to stationary parts at side chambers and sealings are neglected in the present study, although these effects may strongly influence natural frequencies. The obtained frequencies and mode shapes are compared with modal analysis results of the unsupported impeller without acoustic fluid interaction, denoted as impeller in air. Natural frequency results for both air and water agree quite well with experimental data exhibiting deviations of less than 2% (e.g. Egusquiza et al. [12]).

Mode shapes and corresponding frequencies in air and water are shown in Figure 2 using the order of natural frequencies in air. The mode shapes are characterized by the number of diametrical node lines denoted by k and the number of nodal points of blade bending vibrations denoted by b. Obviously, due to the added mass effect, natural frequencies in water are smaller than the corresponding frequencies in air, for all mode shapes. However, depending on the mode shape, the frequency reduction factor f_{water}/f_{air} is varying between values of 0.5 and 0.8 leading to a different order of natural modes in water.

Aim of this investigation is to show that the acoustic FSI approach is suitable to capture added fluid mass effects. Thus, simulation results in air and water have been compared with experimental data. However, in order to study the vibration behavior of a runner during operation, not only the shaft mounting and correct seal and side chamber geometries have to be considered, but also realistic excitation pressure fields varying in space and time. This is possible only in harmonic response or time domain analyses.

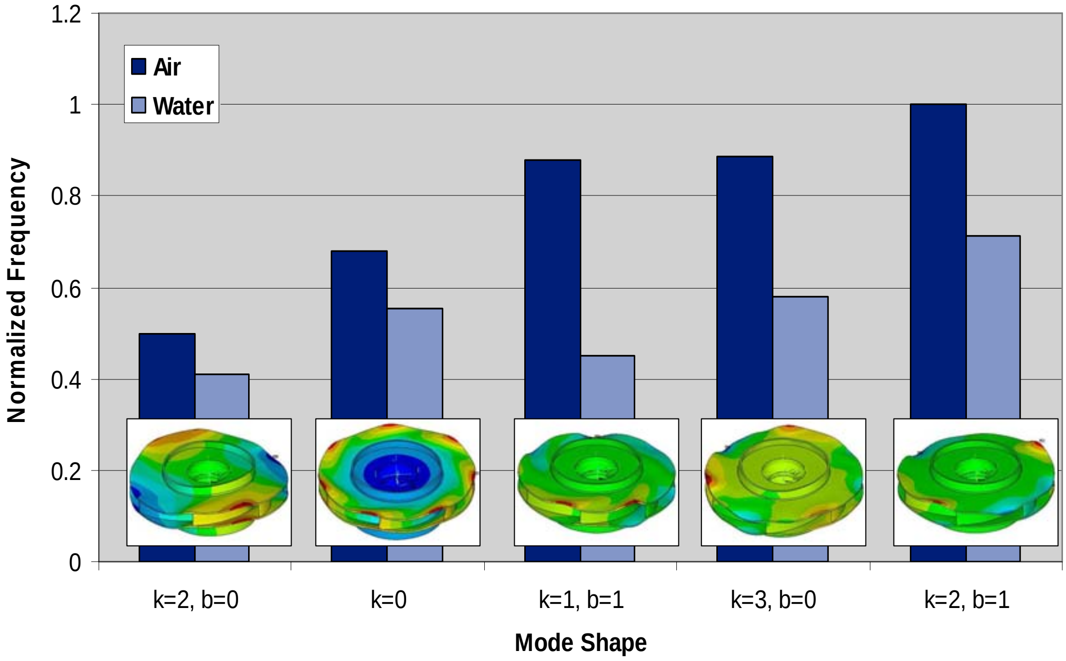

Figure 2: Pump turbine runner - Comparison of natural frequencies in air and water

3.2 Harmonic response analysis of a Francis turbine runner in water excited due to rotor-stator interaction

For higher head Francis turbine runners, dynamic excitation due to rotor-stator interaction (RSI) may be one of the main fatigue contributors. Hence, resulting dynamic stresses have to be evaluated during runner design and are a prerequisite for fatigue life analyses. As shown by Seidel and Grosse [13], harmonic response analyses of the entire runner in water using acoustic fluid-structure interaction are the suitable tool to predict the dynamic runner response and dynamic stresses due to RSI. The advantage of this method is that dynamic properties of the runner in water are well represented and the wave propagation is included.

The excitation pressure field induced by RSI phenomena in vaneless space is varying in space and time and may be composed by superimposing rotating pressure fields with characteristic numbers k of diametrical node lines (e.g. Fisher et al. [14]). The k numbers and corresponding frequencies of most important pressure mode shapes to be regarded for dynamic stress calculations are defined by the number of runner blades, the number of wicket gates and the rotational speed. Due to the linear analysis procedure, the runner response may be analyzed separately for each excitation mode shape by imposing the rotating pressure distribution in terms of Dirichlet constraints onto the acoustic fluid in vaneless space. Initially, excitation pressure amplitudes may be normalized, and when building the combination of results from all regarded mode shapes, amplitude magnification factors identified from unsteady CFD calculations apply in order to scale the normalized results.

In the present study, a higher head Francis turbine with 15 runner blades, 24 wicket gates, and a rotational speed of 150 rpm is regarded. Exemplarily, only the k=6 pressure mode which fluctuates with the gate passing frequency (GPF) of 60 Hz is used to excite the runner. The entire runner structure in water including side chambers and gaps as well as correct fillet radii essential for stress evaluations is modeled by finite elements, see Figure 3. The following boundary conditions apply: The runner structure is fixed at the connection to the shaft, rigid walls are assumed where the fluid domain is connected to stationary parts, and non-reflecting (impedance) boundary conditions are used at inlet and outlet of the fluid domain.

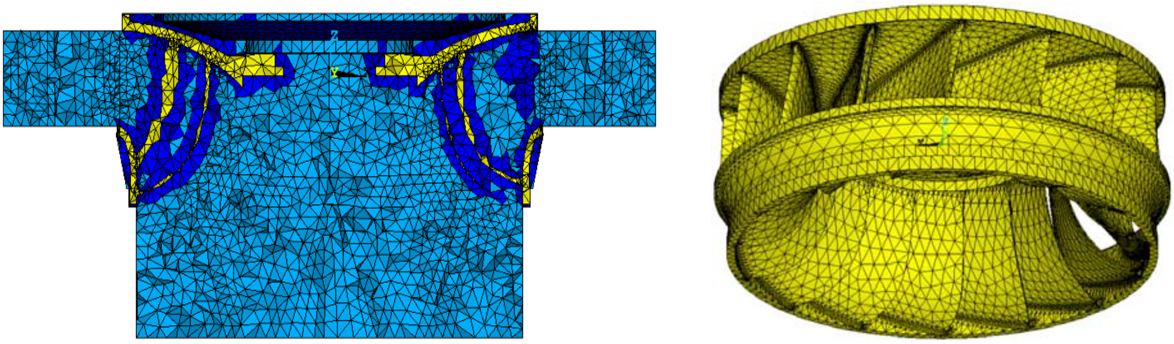

Figure 3: Vertical cut of the finite element model of the runner in water (left) and mesh of the runner structure (right)

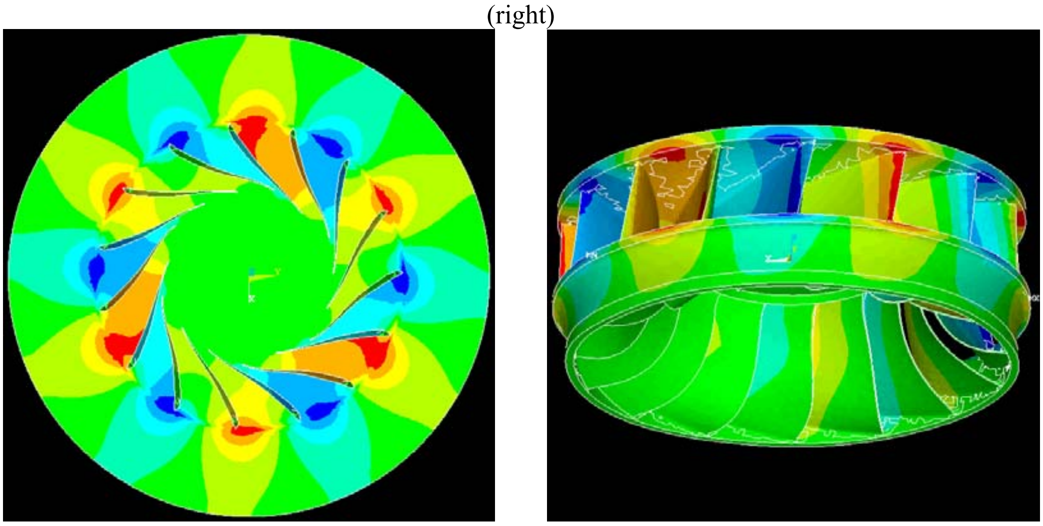

Figure 4: Horizontal cut of the pressure field with k=6 diametrical nodes (left) and pressure distribution on the runner (right)

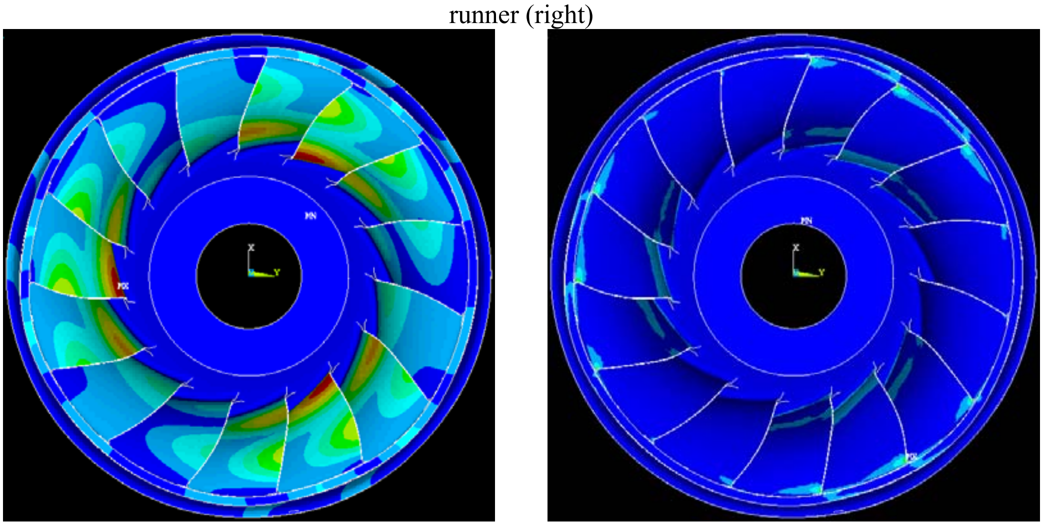

Figure 5: Displacement sum (left) and dynamic v. Mises stress (right) for a normalized pressure excitation with k=6

The dynamic response of the runner structure excited by the normalized k=6 pressure mode shape at GPF=60Hz is computed assuming realistic damping behavior derived by comparison of strain gauge measurements and harmonic response analyses for a comparable project. The pressure distribution on a horizontal cut at distributor center line and on the runner is shown in Figure 4. Figure 5 shows the displacement sum (left) and the dynamic von Mises stress distribution on runner blades. By evaluating dynamic stresses at all possible hot-spots, input data for fatigue life analyses may be obtained.

In order to determine the distance to resonance and to obtain a better understanding of the dynamic behavior of the runner, not only the dynamic response at the main excitation frequency (GPF) is of interest but also response spectra in the entire range of possible excitation frequencies (e.g. due to transient operations or stochastic loading caused by highly separated flow conditions). However, this requires a harmonic response calculation for every frequency leading to very high computational costs if the full finite element model with all degrees of freedom (usually above 1 million) is used.

Therefore, a Krylov subspace based model order reduction technique suitable for acoustic fluid-structure interaction applies, see Lippold and Hübner [15]. After building the reduced order model with some hundred generalized degrees of freedom (having a computational cost comparable to solve the full model for a single frequency), a nearly arbitrary number of frequencies can be solved quite shortly. In addition, the assumed damping ratio may be varied on the reduced system level. Thus, the effect of different damping behavior may be investigated, and dynamic stresses may be calculated for different damping assumptions offering the possibility to determine upper und lower limits.

The response spectrum of the displacement amplitude at trailing edge center is shown in Figure 6 for a normalized pressure excitation with a k=6 mode shape assuming three different damping ratios. The main resonance point can be easily detected at a value of $f_{reso} = 80$ Hz providing sufficient distance to the main RSI excitation frequency of GPF = 60 Hz. Comparing the spectra for different damping ratios at the excitation frequency of GPF = 60 Hz, it can be seen that the influence of damping is quite low at a sufficient distance to resonance (please note the logarithmic scale of the diagram). Hence, even with comparably high uncertainties in damping assumptions, the accuracy of response amplitudes may be quite high, if sufficient distance to resonance is achieved.

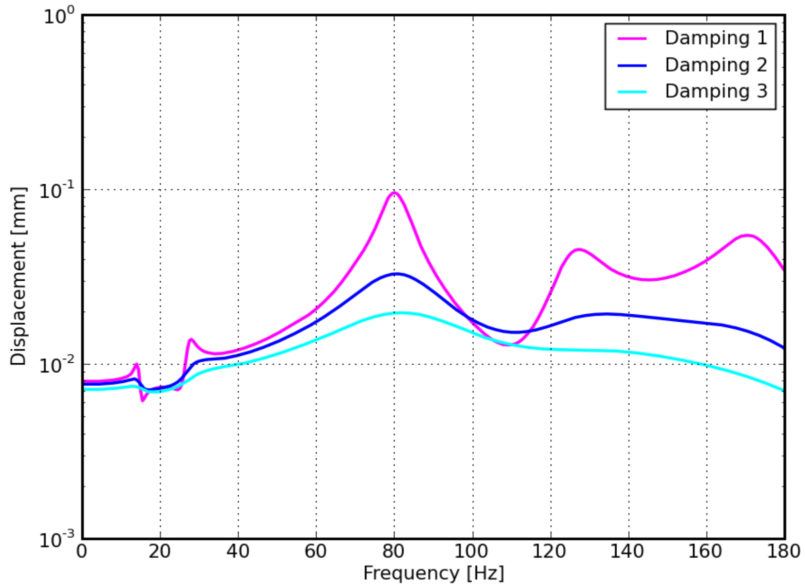

Figure 6: Amplitude spectra of the displacement response at trailing edge center for a normalized pressure excitation with k=

3.3 Hydrodynamic damping identification at a thin hydrofoil using iterative coupling of CFD and FEM

The knowledge regarding hydrodynamic damping effects in hydraulic machines is mainly derived from strain gauge measurements on prototype runners. Systematic theoretical or numerical investigations are hardly available. However, if excitation frequencies (e.g. rotor-stator interaction) are close to natural frequencies of turbine runners or stationary parts, dynamic response amplitudes strongly depend on damping effects. And compared to structural or material damping, hydrodynamic damping effects may be higher by one or two orders of magnitude. Hence, improved knowledge of hydrodynamic damping effects is a prerequisite for dynamic stress calculations and fatigue life analyses of structural parts excited close to resonance.

In the present study, a thin hydrofoil which may be regarded as a simplified model of a Francis turbine runner blade is situated in a water tunnel, see Figure 7 (left), in order to investigate flow induced damping effects depending on amplitudes, mode shapes, and flow velocity. Free bending vibrations of the hydrofoil interacting with the turbulent flow field are simulated using strong (iterative) coupling of the CFD solver ANSYS-CFX and the CSD solver ANSYS-Mechanical in time domain. The coupled solution procedure is visualized in Figure 7 (right).

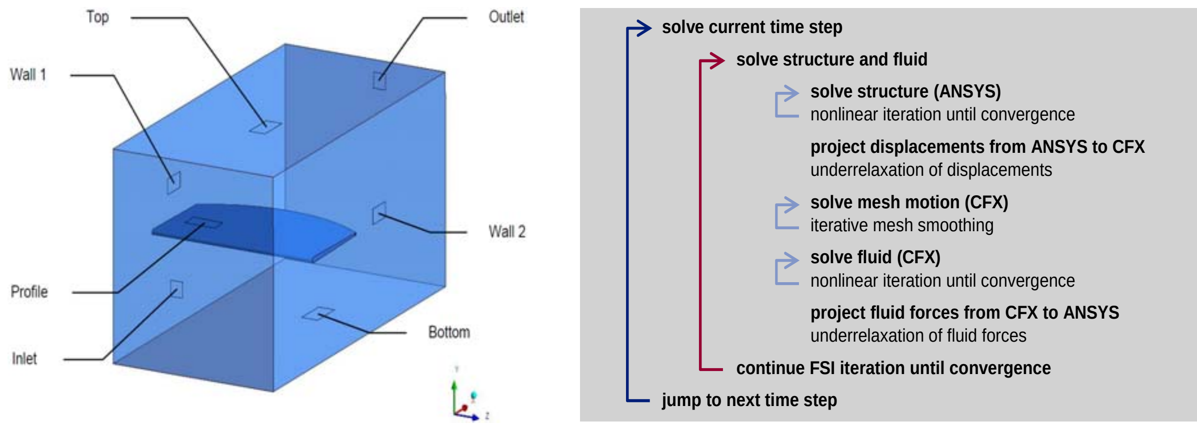

Figure 7: Computational model of thin a profile in a water tunnel (left) and strongly coupled solution procedure (right)

Time histories of hydrofoil deflections after initial pressure loading at the bottom side of the hydrofoil are shown in Figure 8 (left) for different inflow velocities. Damping ratios are identified by averaging the logarithmic decrement between the different peaks ① to ⑤. A linear relation between hydrodynamic damping and inflow velocity is found, see Figure 8 (right). For small vibration amplitudes regarded in the present study, hydrodynamic damping effects are nearly independent of the amplitude.

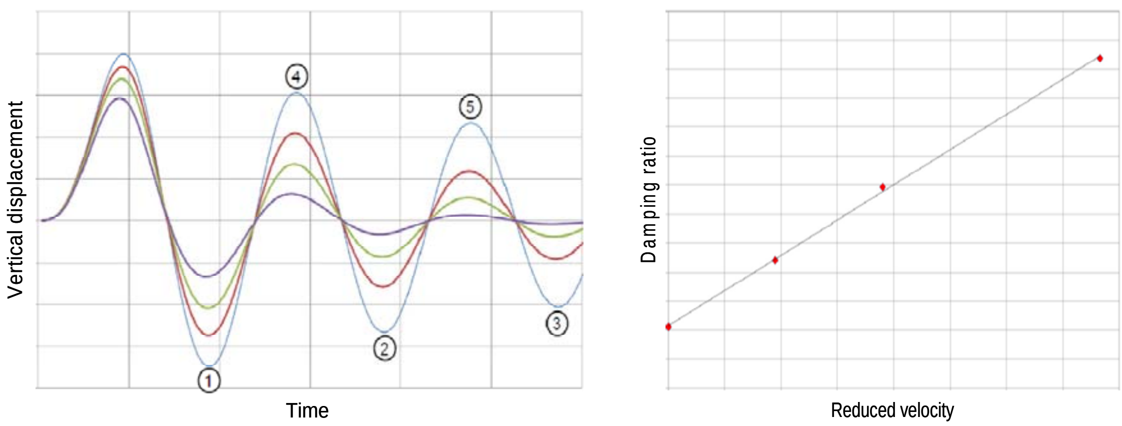

Figure 8: Time histories of free profile vibrations for different inflow velocities (left) and damping ratios (right)

The obtained results are compared with a simplified simulation which is based on the direct coupling of linear structural dynamics in generalized modal coordinates and a boundary element solver for full potential flow in frequency domain. True damping ratios of the coupled system corresponding to coupled mode shapes are obtained directly as real parts of eigenvalues of a non-linear eigenvalue problem which has to be solved iteratively, see Hassig [16].

The damping ratios obtained by this approach are in good agreement with the results from the CFX-ANSYS coupling, shown in Figure 8 (right), but the computational effort is much less. Hence, for analyzing hydrodynamic damping effects or instabilities (negative damping) of elastic structures interacting with non-separating water flow at low Mach numbers, simplified flow formulations may be sufficient not only for qualitative studies but also if quantitative predictions are of interest.

3.4 Analysis of hydroelastic instabilities of a slide gate chain using monolithic coupling of fluid and structure

During operation of a bottom outlet with a closed slide gate and an opend roller-mounted gate, a high velocity water flow (up to 80 m/s) occurs in the duct of the slide gate chain. The flow induces large amplidude chain vibrations perpendicular to the broadside of chain elements. Figure 9 shows a view into the duct including the chain.

Figure 9: View into the duct of the slide gate chain

In order to identify the excitation mechanism and to compare different cross-sectional shapes of the lifting device by means of numerical simulation, a simplified model suitable for efficient numerical analyses is developed. The chain is placed in the center of a square channel and modeled as a continuous plate structure with reduced stiffness to account for joints between the elements, see Figure 10. In addition to the existing chain model with rectangular cross-section, also a cable with circular cross-section, equal axial and low bending stiffness is regarded, to investigate if a more streamlined shape may prevent the hydroelastic instability.

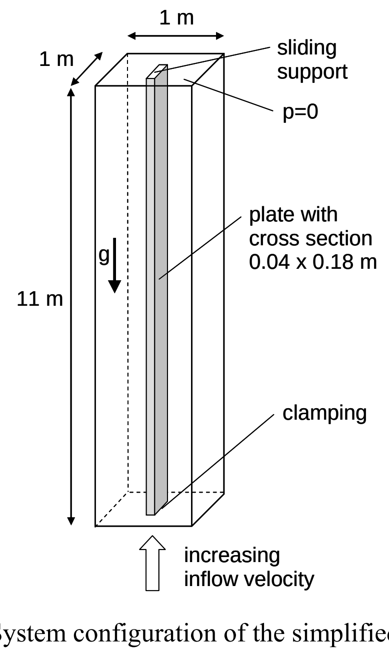

Figure 10: System configuration of the simplified chain model

For numerical simulation of the coupled system, a university code offering monolithic fluid-structure interaction of viscous fluid dynamics and non-linear structural dynamics applies, see Hübner et al. [9, 10]. The calculation model is based on a unified space-time finite element formulation. Stability limits of the simplified chain and cable structure are identified by means of transient simulations with slowly increasing inflow velocities. Initial pertubations are induced by a horizontal impulse load which is acting every 15 seconds on the upper end of the structure. The stability limit is given by the time instant at which the vibration amplitudes due to the impulse load start to increase.

By evaluating the time histories of horizontal displacements at center and upper end of chain and cable model, see Figure 11, stability limits of 43m/s and 60m/s flow velocity can be detected for chain and cable, respectively. Although the bending stiffness of the cable is clearly smaller compared to the chain model, instability occurs at much higher flow velocities. This behavior is caused by the different cross-sectional shapes leading to different flow situations. In case of the chain model, the flow is synchronized over the entire contact surface leading to approximately two-dimensional flow behavior, whereas, the contact surface of the cable is clearly smaller, and the circular shape causes a fully 3-dimensional and more complex flow situation.

Thus, by using a cable instead of a chain structure for slide gate operation, the situation may be improved, especially because a cable does not feature bolt connections which can be destroyed by strong vibrations. However, high amplitude flow induced vibrations cannot be prevented completely if the flow velocity in the slide gate duct reaches values up to 80m/s.

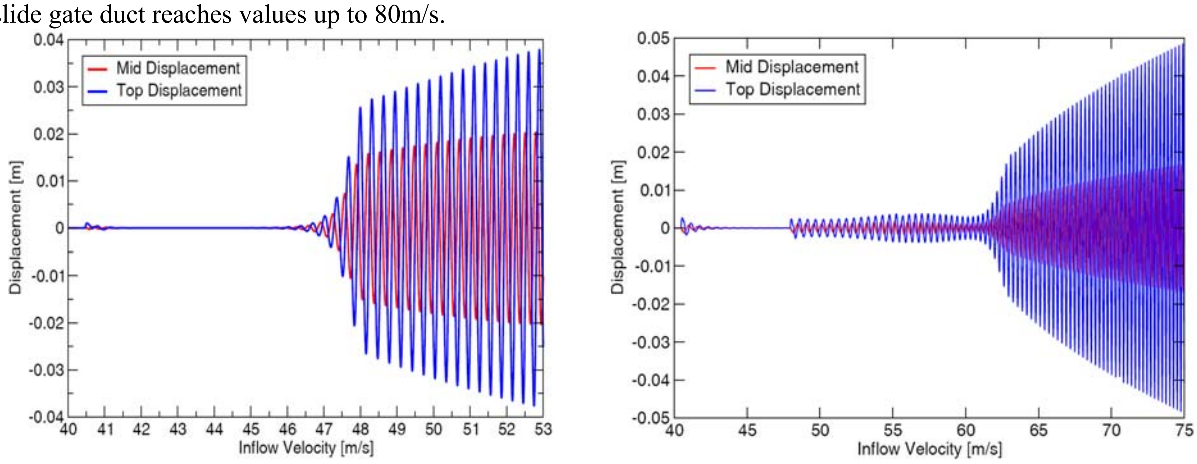

Figure 11: Time histories of horizontal displacements for chain model (left) and cable model (right)

4. Conclusion

In order to determine the reduction of natural frequencies of submerged structures due to the added-mass effect, the modeling of the water as an acoustic (stagnant) fluid is often sufficient, even if the water is flowing in reality. In contrast, the identification of hydrodynamic damping effects and hydroelastic instabilities requires the consideration of a flowing fluid. However, also in these cases, simplified fluid formulations (e.g. potential flow) may give reasonable results for some kind of industrial applications while other ones may require the iterative coupling of multi-purpose fluid dynamics (CFD) and structural dynamics (CSD) codes.

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