Accurate and efficient analysis of dynamic runner stresses considering hydrodynamic damping effects

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Abstract. Safe and reliable dynamic designs of Francis and pump turbine runners require an accurate and efficient prediction of the dynamic response during operation including dynamic stresses. If resonance phenomena cause significant dynamic amplifications, response amplitudes strongly depend on damping effects, and numerical simulations have to include realistic damping behavior. For submerged components like turbine runners, hydrodynamic (flow-induced) damping and hydro-acoustic radiation are much greater than structural or material damping, but cannot be assessed sufficiently accurate by simple assumptions. Therefore, a newly developed fluid-structure interaction (FSI) methodology is presented which offers accurate and efficient harmonic response analyses of submerged structures not only accounting for added-mass effects but also for flow-induced damping and acoustic radiation. The acoustic FSI approach, widely used to consider added-mass effects, is enhanced by convective terms to include fluid forces caused by the deflection of the mean flow due to structural motion. Only a stationary flow solution, obtained from standard computational fluid dynamics analyses, is required as additional input. The linearized second-order system with non-proportional damping matrix is solved in frequency domain using a Krylov subspace based model order reduction technique. The methodology is validated using experimental damping data from an elastic hydrofoil in a cavitation tunnel. A first application to a prototype Francis runner, being excited close to resonance by rotor-stator interaction, reveals a significant influence of the operating condition on hydrodynamic damping and dynamic stresses. The numerical results for both part load and full load agree very well with measured prototype strains. Meanwhile, the automated methodology is applied as a standard tool within the hydro-mechanical runner design process ensuring dynamically safe and reliable runner designs.

1. Introduction

Recently, the hydro power market demands not only high efficiencies and increased power densities but also the smooth and reliable operation of hydraulic machines within an extended operating range. This can result in turbines being susceptible to dynamic excitation phenomena, while at the same time, excitation forces may increase, see Ref. [1]. Hence, a safe and reliable dynamic design of turbine components and especially of Francis type runners becomes more and more important, as emphasized in Ref. [2]. On the one hand, all essential excitation phenomena have to be well understood and quantified, see References [3], [4], [5], and on the other hand, the dynamic response including dynamic stresses has to be predicted accurately and efficiently. For higher head Francis and pump turbine runners, the main fatigue contributor is usually the rotor-stator interaction (RSI) which has been studied widely using theoretical, experimental, and numerical approaches, e.g. References [6], [7], [8], [9], [3], [4].
If dynamic amplifications in the resonance vicinity cannot be avoided, response amplitudes strongly depend on damping effects. Examples are broadband excitation phenomena like turbulences covering a wide frequency range, but also periodic excitation forces like RSI which may act close to resonance if design spaces are restricted, e.g. in modernization projects. In such cases, numerical simulations of dynamic systems have to include realistic damping formulations. For submerged components like turbine runners, hydrodynamic (flow-induced) damping and hydro-acoustic radiation are usually much greater than structural or material damping, but highly depend on several conditions like operating point, vibration mode shape, and geometry, see Ref. [10]. Hence, in the last decade, considerable R&D activities in academia and industry were devoted to hydrodynamic damping in hydraulic machinery, see References [11], [12], [13], [14], [15], [16], [17]. In order to determine the damping ratio for given project conditions, unsteady computational fluid dynamics (CFD) is usually applied, either with 2-way fluid-structure interaction (FSI) or with prescribed harmonic motion of boundaries (flutter analysis). The computational effort for this additional analysis step is quite high, even if advanced numerical schemes may accelerate the simulation considerably, e.g. Transient Blade Row methods to reduce the number of included runner channels and Harmonic Balance approaches to switch to frequency domain.

In order to analyze the dynamic behavior and dynamic stresses of submerged structures like hydro turbine runners, finite element analyses (FEA) including acoustic FSI are the state-of-the-art technology to account for added-mass effects of the surrounding water. The accuracy in predicting natural frequencies in water is proven by comparison with experimental modal analyses, e.g. Ref. [18], providing that geometry, boundary conditions, and material properties of acoustic fluid and solid domain are properly defined. If harmonic response analyses apply to predict dynamic stresses, correct loading is crucial and is usually determined by unsteady CFD analyses, e.g. to calculate RSI-induced pressure fluctuations. In addition, for each vibration shape and flow condition, appropriate damping ratios are required which may be calculated using additional unsteady CFD analyses, as described above. However, a much more efficient approach which may be even more accurate, is to enhance the acoustic FSI formulation in such a way that the hydrodynamic damping is part of the solution, as described in Section 2. This approach is based on a convective wave equation including appropriate interface conditions to consider fluid forces caused by the deflection of the mean flow due to structural motion.

Only stationary flow solutions, obtained from standard CFD analyses, are required as additional input.

In Section 3, an efficient solution procedure for harmonic response analyses is presented using a Krylov subspace based model order reduction (MOR) technique which is suitable to solve the linearized second-order system with non-proportional damping matrix. In Section 4, the methodology is validated using experimental damping data from an elastic hydrofoil in a high-speed cavitation tunnel. In order to predict accurately dynamic runner stresses, Section 5 describes the requirements and assumptions for building a reliable FEA model of runner structure and surrounding fluid during operation. A first application to a prototype Francis runner being excited close to resonance by RSI is given in Section 6. A significant influence of the operating condition on hydrodynamic damping and dynamic stresses is observed. The numerical results for part load and full load agree very well with measured prototype strains, as shown in Section 7. Finally, Section 8 draws some conclusions using the experience from different applications of the automated procedure as a standard tool within the runner design process.

2. Theoretical background of linearized fluid-structure interaction

The compressible Navier-Stokes equations including continuity equation serve as the starting point to derive linearized flow equations which are suitable to develop a monolithic FSI formulation for analyzing turbine runner dynamics with flow-induced damping effects. At first, the Navier-Stokes equations are split into a steady mean flow and a linear perturbation for the fluctuations. In the next step, the linear perturbation equations are simplified by assuming small Mach numbers and incompressible mean flow, by neglecting viscosity effects in the fluctuating motion, and by building substantial derivatives to combine the set of equations to a single equation. Finally, the equation can be formulated
in pressure variables and extended by a right-hand side (RHS) excitation term representing a fluctuating mass source. This leads to the convective wave equation

$$\frac{1}{c^2} \cdot \frac{\partial^2 p'}{\partial t^2} + \frac{2}{c^2} \mathbf{v} \cdot \nabla \left( \frac{\partial p'}{\partial t} \right) - \Delta p' = \frac{\partial Q'}{\partial t}$$  \hspace{1cm} (1)

where $c$ denotes the speed of sound in water, $p'$ the fluctuating pressure, $t$ the time, $\mathbf{v}$ the mean velocity vector, and $Q'$ the fluctuating mass source. Equation (1) describes an acoustic fluid which is transported by a stationary and incompressible flow field.

For the structural part, the equation of motion for linear elastodynamics is used, and to account for the interaction of fluid and structure, dynamic and kinematic interface conditions apply. Special attention has to be paid to the kinematic interface conditions in order to describe flow-induced damping effects correctly. After finite element discretization of fluid and structure in space and switching to frequency domain, the equations can be coupled monolithically in a single matrix equation

$$
\begin{pmatrix}
-\omega^2 \mathbf{M}_S & \mathbf{B}_S & \mathbf{K}_S & \mathbf{K}_{SF} \\
\mathbf{M}_{FS} & \mathbf{B}_{FS} & \mathbf{B}_F & \mathbf{K}_F \\
\mathbf{K}_{FS} & \mathbf{K}_S & \mathbf{K}_{FS} & \mathbf{K}_{FS} \\
\mathbf{B}_S & \mathbf{B}_{FS} & \mathbf{B}_F & \mathbf{K}_F
\end{pmatrix}
\begin{pmatrix}
\mathbf{u}_S \\
\mathbf{u}_{FS} \\
\mathbf{u}_F \\
\mathbf{p}_F
\end{pmatrix}
= 
\begin{pmatrix}
\mathbf{f}_S \\
\mathbf{f}_{FS} \\
\mathbf{f}_F \\
\mathbf{p}_F
\end{pmatrix}
$$  \hspace{1cm} (2)

where $\omega$ is the angular frequency and $i$ the imaginary unit. In this enhanced acoustic FSI formulation, $\mathbf{M}$, $\mathbf{B}$ and $\mathbf{K}$ are sub-matrices of the coupled mass, damping and stiffness matrix, respectively. The indices $S$ and $F$ stand for structure and acoustic fluid, respectively. Nodal variables are the complex amplitudes of structural displacement $\mathbf{u}_S$ and acoustic pressure $\mathbf{p}_F$. On the RHS, complex amplitudes of structural load $\mathbf{f}_S$ or acoustic mass source $\mathbf{q}_F$ may be prescribed to excite the system. The damping matrix of the fluid $\mathbf{B}_F$ accounts for convective effects and non-reflective or partly-reflective impedance boundary conditions. Structural damping may be included in the matrix $\mathbf{B}_S$. The coupling matrix $\mathbf{K}_{SF}$ in the upper (structural) equation represents dynamic interface conditions, and the coupling matrices $\mathbf{M}_{FS}$, $\mathbf{B}_{FS}$ and $\mathbf{K}_{FS}$ in the lower (fluid) equation account for kinematic interface conditions including the deflection of the mean flow due to structural motion, being responsible for flow-induced damping.

3. Efficient harmonic analyses of submerged structures including flow-induced damping

For standard Francis runner applications, the fully coupled matrix equation system (2) consists of several million equations and can be solved for selected angular frequencies $\omega$ using an appropriate direct solver. However, if a highly resolved response over a wide frequency range is desired to accurately detect resonances frequencies, the solution of the full equation system becomes very costly. Therefore, a Krylov subspace based MOR technique, see Ref. [19], is employed which is suitable for second-order vibro-acoustic systems as given in Equation (2). To apply the MOR technique to equation systems either assembled by ANSYS Mechanical or by in-house FEA tools, the software toolkit MOR for ANSYS, see Ref. [20], is used. From the original FEA equation system

$$
(-\omega^2 \mathbf{M} + i\omega \mathbf{B} + \mathbf{K}) \cdot \mathbf{x} = \mathbf{f}
$$  \hspace{1cm} (3)

comprising several million equations, a second-order Arnoldi method extracts the reduction vectors $\mathbf{V}$ which are spanning the Krylov subspace. With the reduction vectors, the original FEA solution vector

$$\mathbf{x} = \mathbf{V} \cdot \mathbf{z}
$$  \hspace{1cm} (4)

is approximated, and the original equation system (3) is reduced to a low-dimensional system

$$
(-\omega^2 \mathbf{M}_r + i\omega \mathbf{B}_r + \mathbf{K}_r) \cdot \mathbf{z} = \mathbf{f}_r
$$  \hspace{1cm} (5)

denoted by the index $\mathbf{r}$ with the reduced solution vector $\mathbf{z}$ usually featuring some hundred unknowns. Solving the reduced system (5) for some thousand frequencies is not a big effort on a modern computer, e.g. by using a highly optimized Python package for numerical mathematics within a fully automated, Python-based analysis procedure.
However, the complete back projection of the reduced solution according to Equation (4) may be a much higher numerical effort, but usually, it is not necessary for all frequencies and for the entire model. In order to assess RSI-induced Francis runner dynamics, the standard approach is as follows. On the one hand, highly resolved response spectra are computed for selected variables and for the quadratic norm of the reduced solution vector $\mathbf{z}$, enabling an accurate and reliable check of the safety margin against resonance. On the other hand, the FEA solution either of the complete model or of a single runner sector is recovered only for selected frequencies, especially for the gate passing frequency (GPF) which is the only significant RSI excitation frequency for the runner, see Ref. [4]. For the selected frequencies, all kind of post processing can be performed, particularly dynamic stress evaluations and the creation of result plots and animations, permitting the accurate prediction of maximum RSI-induced stresses.

4. Validating the FSI formulation by comparison with an elastic hydrofoil experiment

The dynamic response of higher head Francis runners excited by RSI is mainly characterized by bending vibrations of the runner blades. Therefore, a thin elastic hydrofoil placed in a high-speed cavitation tunnel applies as a test case to check, if the formulation accurately accounts for the hydrodynamic damping of runner blades. The hydrofoil is clamped on both sides, and in the experiment, mostly the first bending mode, shown in Figure 1, is excited by the shock wave of a growing cavitation bubble generated by a spark plug. More details on a similar experiment can be found in Ref. [11].

![Figure 1](image1.png)

**Figure 1.** First bending mode of a thin elastic hydrofoil, clamped on both sides and placed in a water filled tunnel.

The free vibration response close to the trailing edge center is measured with a Laser Doppler Vibrometer for several inflow velocities, see the experimental set-up in Figure 2. From the measured vibration velocity, the damping ratio is identified as described in Ref. [11]. At first, a Butterworth filter limits the frequency range, then, a Hilbert transform provides the envelope, and finally, the envelope is fit with an exponential curve. Since structural and material damping are smaller by orders of magnitude, the identified damping ratio characterizes the hydrodynamic damping.

![Figure 2](image2.png)

**Figure 2.** Experimental set-up to measure hydrodynamic damping at a thin elastic hydrofoil in a high-speed cavitation tunnel.

The numerical investigation is based on two steps. At first, a steady-state CFD analysis of the test section with hydrofoil is performed, providing the mean velocity field. Subsequently, a finite element model of hydrofoil and water in the test section is build to solve the enhanced acoustic FSI formulation given in Equation (2). In the FSI formulation, the terms being responsible for hydrodynamic damping
depend linearly on the mean flow velocity. Hence, for the hydrofoil example, the damping effect linearly increases with the inflow velocity, and a single analysis for a selected inflow velocity is sufficient. To directly determine the damping ratio, Equation (2) is not solved by a harmonic response analysis, as described in Section 3, but the RHS is set to zero, and a damped modal analysis is performed. In Figure 3, the calculated damping ratio (relative to critical damping) of the first bending mode is compared to the experimental result, showing very good agreement. The inflow velocity is divided by the free-vibration frequency in water and a length scale of the hydrofoil structure leading to the reduced velocity commonly used in aeroelasticity.

Figure 3. Comparison of hydrodynamic damping at a thin elastic hydrofoil from measurement and calculation.

5. Finite element modeling of Francis runners to accurately predict the dynamic response

In order to accurately analyze resonance distances and dynamic stresses in Francis runners during operation by means of the (enhanced) acoustic FSI approach, a careful finite element modeling of fluid and structural domain is required. Since linear modal and harmonic response analyses are not capable to include different frames of reference, the rotating frame of the runner applies for the entire model. Hence, stationary components like guide vanes or head cover cannot be included, what is acceptable, since the influence on runner dynamics is neglectable, if stationary components are not excited in the vicinity of resonance. A typical FEA model for acoustic FSI analyses is shown exemplarily in Figure 4 including a cut view of the entire domain (left) and a bottom view of the runner (right).

In the structural model, geometric details are fully resolved including correct fillet radii and labyrinth seals. The overall mesh quality is sufficient high to obtain mesh independent natural frequencies. In addition, a single blade is locally refined at stress hot spots to provide converged values of maximum dynamic stresses. If the dynamic response (e.g. due to RSI) is characterized by spinning vibration mode shapes with two or more diametrical node lines, realistic boundary conditions for the runner structure are achieved by constraining the nodal displacements at the connection to the shaft.

Figure 4. FEA model for (enhanced) acoustic FSI analyses: Cut view of the entire domain (left) and bottom view of the runner (right).
The fluid domain contains side chambers with exact geometry of gaps and seals to accurately account for added-mass effects under operating conditions. At inlet and outlet of the fluid domain, non-reflecting boundary conditions usually apply to prevent spurious acoustic modes and to include damping effects due to acoustic radiation. Realistic assumptions for the speed of sound are crucial if acoustic or coupled structural-acoustic resonance phenomena play an important role and have been derived from dynamic pressure measurements at prototype turbines in operation. The mean flow velocity, which is an input parameter for the enhanced acoustic FSI formulation, is taken from steady-state CFD analyses of single runner channels performed during the hydraulic design optimization. The model-scale CFD velocities in rotating frame of reference are scaled to prototype conditions, copied to a full runner geometry, and projected onto the finite element mesh. If calculated or estimated flow conditions in runner side chambers are considered, too, the formulation accounts for the effect of natural frequency splitting depending on the spinning direction of the vibration mode, as revealed in References [21] and [10] for a submerged rotating disc. By including this phenomenon, operational resonance frequencies of spinning vibration modes with significant band or crown motion are predicted with higher accuracy.

Dynamic loading of Francis type runners mainly results from flow effects, and excitation forces may be predicted by unsteady CFD analyses, see Ref. [3]. For higher head runners, RSI is the main excitation source during normal operation, see Ref. [4]. Therefore, in the following Sections, the process to analyze RSI-induced runner dynamics is described and results are compared to measured prototype strains.

6. Harmonic response analysis of Francis runners excited by RSI

Using state-of-the-art technology to perform unsteady and incompressible CFD analyses of runner and distributor in model scale, RSI-induced pressure fluctuations in vaneless space can be predicted with high accuracy including upscaling to prototype conditions, see References [3], [4], [9]. Incompressible CFD results do not account for potential pressure amplifications from acoustic resonances in prototype runner channels. But in vaneless space, at the location of emergence, pressure fluctuations are well captured, see Ref. [9], and potential acoustic amplifications are fully resolved in the (enhanced) acoustic FSI formulation. From RSI theory, prototype measurement and CFD results, see References [6], [7], [3], it is well known, that RSI pressure fluctuations in vaneless space can be fully described by a linear combination of rotating pressure modes. Each mode has a signed number \( k \) of diametrical node lines and an associated frequency which is different in stationary and rotating frame. For a turbine with \( Z_r \) runner blades and \( Z_g \) guide vanes, potential RSI pressure modes in the rotating frame are given by

\[
k = \pm m \cdot Z_r - n \cdot Z_g \quad \text{with} \quad m, n = 0, 1, 2, \cdots
\]

where the sign of \( k \) specifies, if a pressure mode is spinning with (positive) or against (negative) the runner rotation. To determine amplitude and phase angle of each pressure mode and to identify relevant modes, CFD pressure fluctuations are evaluated using space-time Fourier analyses. For the medium high head Francis turbine with 15 runner blades and 22 guide vanes, considered hereinafter, the pressure modes with \( k = -22, -7, +8 \) are identified to be relevant for runner excitation. In the rotating frame, all three modes excite the runner with the gate passing frequency \( GPF = Z_g \cdot f_n \) where \( f_n \) is the frequency of runner rotation. Higher harmonics are neglectable, what is also known from strain gauge measurements at several prototype Francis runners, see Ref. [4].

Since the harmonic analysis is linear, the runner response is analyzed for each of the relevant mode shapes, using a corresponding RHS excitation field with normalized amplitude to excite the fluid in the vaneless-space region. Subsequently, the results are scaled and combined by comparing the pressure response due to normalized RHS excitations with pressure amplitudes identified from unsteady CFD, obtaining a nearly exact matching between combined harmonic response result and CFD pressure.

For the regarded runner, which is excited by RSI close to resonance, damping and hence, operating conditions are expected to have a significant influence on response amplitudes. Therefore, the runner response is analyzed for full load and part load. The respective mean flow conditions from steady-state CFD are visualized by streamlines in Figure 5 and serve as input for the enhanced acoustic FSI solver.
The FEA model of the runner is built according to Section 5. Then, the enhanced acoustic FSI procedure from Sections 2 and 3 applies to both operating points using the respective unsteady CFD pressure to scale and combine harmonic response results of relevant excitation modes. For the resulting mode shape combinations, highly resolved spectra of selected variables are calculated. Figure 6 shows the spectra for the quadratic norm of the reduced solution vector $\mathbf{z}$ of Equation (5) which includes all degrees of freedom, enabling a reliable resonance check, since all resonance phenomena are captured. However, the magnitude does not have a physical meaning. A quite small resonance distance for RSI excitation of about 3% and a significantly reduced damping ratio at part load can be observed.

To evaluate runner dynamics including dynamic stresses during operation, the complete solution is recovered for the GPF by back projection according to Equation (4). Axial displacement and dynamic von Mises stress at a selected point during the harmonic vibration period are shown in Figure 7 for full-load operation. Spatial distributions at part load are nearly identical, only the amplitudes differ. The runner is obviously responding with a pure $k=-7$ vibration mode, since the limited number of 15 blades causes the runner to react to the $k=-22, +8$ pressure excitations also with a $k=-7$ vibration mode.
7. Comparison of RSI-induced strains from simulation and prototype measurement

For the medium high head Francis runner under investigation, strain gauge (SG) measurements were performed at the prototype during operation, as described e.g. in Ref. [4]. To extract the RSI-induced fraction of measured strains, a band-pass filter with a narrow band around the GPF applies followed by the determination of a characteristic amplitude. Alternatively, a discrete Fourier transform may apply. At 16 SG sensors, placed on pressure and suction side of a single runner blade close to the trailing edge, strains are measured and can be compared to numerical results. For this purpose, the enhanced acoustic FSI solution for full load and part load is evaluated at the GPF to determine strain amplitudes at the position and in direction of each strain gauge.

Normalized RSI-induced strain amplitudes at the SG locations obtained by measurement and FSI simulation are compared in Figure 8 for full-load operation and in Figure 9 for part-load operation. As expected from resonance curves in the previous section, due to much less hydrodynamic damping at part load, the strain response in the resonance vicinity is significantly higher although the RSI pressure excitation is lower. Considering the different sources of potential inaccuracies like prototype tolerances, numerical simplifications, deviations in sensor positions, and measuring tolerances, the numerical results for both part load and full load agree very well with measured prototype strains. Hence, as proven in Section 4, flow-induced damping effects, inherently included in the FSI formulation, are well captured. If corresponding damping ratios have to be determined, either the results may be compared to acoustic FSI analyses with prescribed damping ratio, or damped modal analyses can be performed considering the enhanced acoustic FSI approach.

Figure 8. Normalized RSI-induced strain amplitudes at 16 SG locations from measurement and simulation at full load.

Figure 9. Normalized RSI-induced strain amplitudes at 16 SG locations from measurement and simulation at part load.
8. Conclusions

The development, verification, validation and application of an accurate and efficient procedure for predicting dynamic and especially RSI-induced stresses in Francis type runners is presented. The procedure accounts inherently for hydrodynamic damping effects with very little additional effort, using a newly developed linearized FSI formulation based on a convective wave equation. To efficiently solve the FSI formulation in frequency domain, a second-order Krylov subspace based MOR technique applies. The ability to account for hydrodynamic damping effects is validated using experimental damping data from an elastic hydrofoil in a high-speed cavitation tunnel.

Fortunately, for a prototype Francis runner being excited by RSI close to resonance, reliable strain gauge data is available. The measurement clearly shows that part-load operation with lower flow rate and hence lower damping is causing significantly higher strain amplitudes, although the excitation pressure is smaller compared to full load. Numerical results for both part load and full load agree very well with measured prototype strains, confirming a considerably reduced damping ratio at part load.

Nevertheless, according to internal guidelines and customer specifications, a sufficient distance to resonance is assured during design optimizations of Francis runners. However, the present results evidence, that based on accurate and reliable stress predictions, a safe operation without fatigue problems may be guaranteed even within the vicinity of resonance, at least for a medium head range.

By applying the procedure at first to different test cases and later on as a standard tool within the design process or during trouble-shooting activities, the following findings can be summarized:

- The procedure is proven to be an accurate and efficient analysis tool for developing reliable Francis type turbine runners featuring dynamic designs which safely reach guaranteed properties in terms of resonance distance and dynamic stress level.
- Thanks to the efficiency and the automated process, the procedure is included within the runner design optimization loop accounting for hydraulic, static, and dynamic requirements. During successful developments especially of high head runners, the optimization loop may be passed several times, leading to holistically optimized runner designs.
- Enhanced knowledge of using design parameters to control turbine runner dynamics is gained by collecting the experience from several applications of the presented technology.
- Since the MOR technique allows for arbitrary damping matrices, impedance boundary conditions may be used to model hydro-acoustic radiation at non-reflecting or partially-reflecting boundaries. If acoustic amplification effects with standing wave phenomena are present, as being quite likely in prototype turbines, the hydro-acoustic radiation may have a significant impact on coupled vibro-acoustic mode shapes, resonance frequencies, and amplitudes.
- If the circumferential flow in runner side chambers is included, the FSI formulation considers the effect of natural frequency splitting depending on the spinning direction of vibration modes, as shown for a submerged rotating disc in References [21] and [10]. This may clearly improve the accuracy of predicted resonance frequencies for vibration modes with significant band or crown motion.
- Beside the application to RSI, the procedure is suitable for all kind of periodic excitation acting on submerged structures. For example, it is applied successfully to investigate stationary and rotating turbine components excited by von Kármán vortex shedding phenomena.
- In addition to the efficient calculation of highly resolved harmonic response spectra, the formulation can be used to perform modal analyses including hydrodynamic damping and hydro-acoustic radiation, revealing explicitly the damping ratio for each vibration mode shape.

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