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Modal behavior of a reduced scale pump turbine impeller. Part II: Numerical simulation

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Abstract. A numerical simulation has been carried out to analyze the modal behavior of a reduced scale pump-turbine impeller. The simulation has been done using FEM method, in air and in water. The same boundary conditions than in the experiment were considered: free body in air and free body submerged in a reservoir of water. A sensitivity analysis to determine the influence of the number of elements was done. The influence of the input parameters was also taken into account. Finally, a mesh with 165000 elements for the impeller in air and of 508676 for the impeller in water was used. The results obtained with the simulation have been compared with the experimental ones (paper 1). Both the natural frequency values and the mode-shapes were compared. The numerical results showed small deviation from experiment in the first modes in modes with low modal density. In some coupled modes been found. With the updated model the mode-shapes have been analyzed. Some modes with high modal density have been found. As indicated in the experiment, the effect of the added mass reduces the natural frequencies and also changes the characteristics of the coupled modes.

1. Introduction

Pressure fluctuation induced by the rotor stator interaction phenomenon (RSI) can produce high vibrations and fatigue damage inthe impeller of pump turbines [1, 2]. For this, an accurate understanding of the dynamic behaviour of this type of impeller, especiallywhen it is submerged in water, is of paramount importance.

The fluid in contact to a body (likes a pump turbine runner) modifies the dynamic equations of motion beca use the fluid is set into motion. The natural frequencies of the structure may change considerably if it vibrates in a stagnant fluid with a large density. The system has to be treated as a fluid structure interaction problem. In this way, the structural dynamic equation has to be coupled with the fluid equations. By adopting following assumptions on the fluid:

-The fluid is slightly compressible (density changes due to pressure variations).
-The fluid is non-viscous (no viscous dissipation).
-The flow is irrotational.
-There is no mean flow of the fluid.
-Changes of mean density and pressure in different areas of the fluid domain remain small.

The equations for the fluid structure interaction problem can be written in form of [3,4]:

$$\begin{bmatrix} [M_s] & [0] \\ [M_{fs}] & [M_f] \end{bmatrix} \begin{Bmatrix} \{\ddot{u}\} \\ \{\ddot{p}\} \end{Bmatrix} + \begin{bmatrix} [C_s] & [0] \\ [0] & [C_f] \end{bmatrix} \begin{Bmatrix} \{\dot{u}\} \\ \{\dot{p}\} \end{Bmatrix} + \begin{bmatrix} [K_s] & [K_{fs}] \\ [0] & [K_f] \end{bmatrix} \begin{Bmatrix} \{u\} \\ \{p\} \end{Bmatrix} = \begin{Bmatrix} \{F_s\} \\ \{0\} \end{Bmatrix} \tag{1}$$

In the simulation using finite element method all the matrices with subscript f , as well as the couplin g matrices $[M_{fs}]$ and $[K_{fs}]$] will be generated by the fluid element. The compatible structural elemen used in the model will generate the matrices with subscript s .

Concepts of harmonic index (k) and nodal diameter (d) have also to be introduced. The nodal diamet er refers to the appearance of a simple geometry (e.g., a circular disk) vibrating in a certain mode. Most mode shapes contain lines of zero out-of-plane displacement which cross the entire structure. For a complicated structure exhibiting cyclic symme- try (e.g., a hydraulic turbine), the dynamic behavior b ecomes complex, and nodal diameter may not be clearly observa- ble in a mode shape. The nodal dia meter describes the actual number of observable waves around the structure. The harmonic ind ex is an integer that determines the variation in the value of a single degree of freedom (DOF) atpoint s spaced at a circumferential angle equal to the sector angle. The following equation represents the rela tionship be-tween the harmonic index k and nodal diameter d for a model consisting of N sectors:

$$k = \begin{cases} N/2 \;\; if \;\; N \;\, is \;\, even \;\, number \\ (N-1)/2 \;\; if \;\; N \;\, is \;\, odd \;\, number \end{cases} \tag{2}$$

$$d = mN \pm k$$

where m = 0,1,2,3…∞. From the equation [3], the nodal diameter equals to the harmonic index in only some cases, and the solution of a given harmonic index may contain more than one mode with differe nt nodal diameters.

Therefore, the vibration modes of a cyclic symmetric structure can be classified according to the numb ers of har- monic index (k) and nodal diameters (d). Defined by the condition k=0, the modes are sing let. These modes are inde- pendent of the angular coordinate θ and natural frequencies are distinct. The modes with $k \neq 0$ are doublet; they have a pair of mode shapes with the same natural frequency. Each member of such a pair has either sinusoidal or cosinusoidal θ-dependent mode shape. The only differen ce between them is a spatial phase shift of ϕ [18]:

$$\phi = \frac{\pi}{2d} \tag{3}$$

Some references are available in this topic [6,7]. [8] have published a detailed experimental investigation on a reduced scale model of a Francis turbine. The corresponding simulation using the same model runner and compatible conditions has been carried out by [9,10].

However, a pump-turbine impeller has a quite different design and a more complex geometry than the Francis turbine. In this paper, numerical modal analysis on a model pump-turbine impeller to determine the influence of surrounding water will be described. The natural frequencies, mode shapes and frequency reduction ratios were obtained. Numerical results will also be verified by the experimental modal tests (see Part 1 of this paper), which were carried out with the compatible condition with simulation.

2. Numerical simulation

The model runner has 9 blades. The shape of the simulated model runner is shown in Figure 1. The material used is a bronze alloy whose properties are given by Table 1.

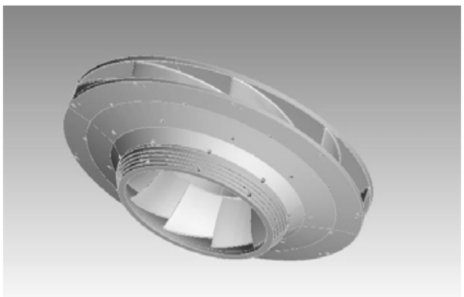

Fig. 1 Geometry of the whole model

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Table 1. Properties of material

Properties	Young's modulus	Poisson's ratio	Density
value	110GPa	0.34	8300 kg·m^{-3},

The model runner is a segmented body. The connections in the model runner are too complicated to be simulated properly. Some simplifications must be applied to approach the practical condition. Using the experimental modal analysis results, the numerical model can be optimized.

First of all, the runner will be treated as a one-piece body and mesh will be built based on the designed parameters. Then, a local sensitivity analysis on mesh density will be carried out to obtain a numerically converged model. By analyzing the correlation between the simulation and experiment results, different simplification will be tested and more sensitivity may be necessary to improve the correlation until acceptable accuracy is yielded. With the models verified, the simulation considering fluid effect can be carried out and the results will also be compared with experiment. Only the model giving good agreement both in air and in water can be defined as a proper approach to the reality, and be used in some further simulations.

3. Sensitivity Analysis and Simulation in Air

Before defining the final mesh configuration, the influence of mesh density was checked. Tetrahedral elements were considered. Three meshes with increasing number of elements were built up, as it can be observed in Figure 3.

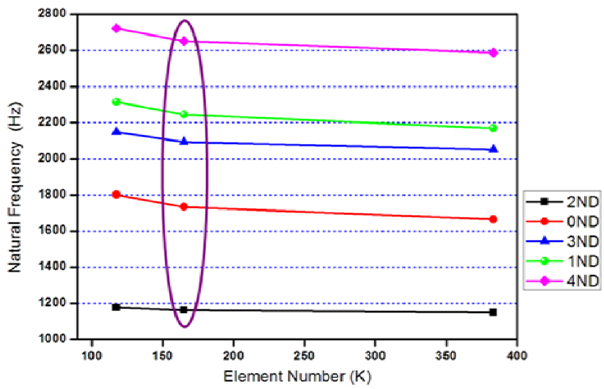

Fig. 2 Results of the mesh sensitivity analysis

Mesh2 (with a circle) was selected to do the simulation

Corresponding experiment is done with the runner suspended by a rope in air. The runner is treated as a free vibrating body without any constraint applied. Thus, the simulation results can be compared with experiment data to analyze the correlation.

Comparison between experimental and simulation results adopting values from table 1.1 are showed in the figure 3.

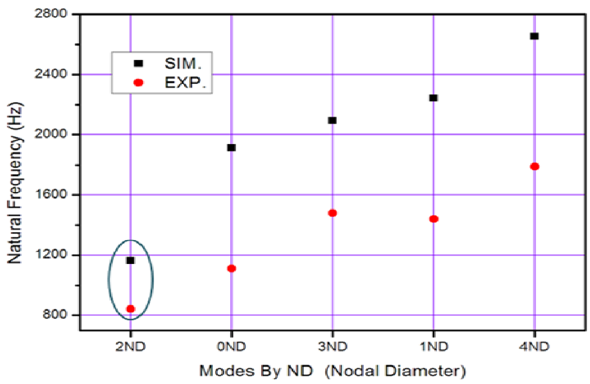

Fig. 3 Comparison of natural frequencies

It can be seen that are not enough good agreement between experimentation and simulation.
Compared with a one-piece runner, the segmented construction reduces the natural frequencies more or less proportionally, so the stiffness may be tuned globally while the density remained unchanged since they have the same mass [5].
A sensitivity analysis was carried out and the results are plotted in Figure 4 in which the frequency values are normalized by the corresponding experimental ones.

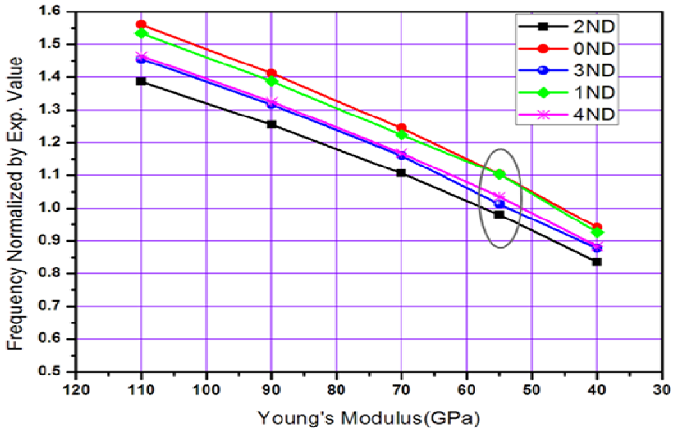

Fig.4 Sensitivity analysis about stiffness of material

By the sensitivity analysis, the 55 G Pa is selected as the value of global Young's Module, and the simulations with this adapted parameter were carried out both in air and in water. Natural frequencies are tabulated in Table 2. Mode shapes (2ND, 0ND, 3ND) (ND: nodal diameter) obtained from simulate and experimental results are showed in Figure 5.

Table 2. Comparison of natural frequencies in air

	SIM.AIR	**EXP.AIR**
2ND	825.26	838.00
0ND	1247.40	1111.00
3ND	1479.75	1463.00
1ND	1605.55	1439.00
4ND	1871.80	1811.00
5ND	2350.15	2282.00
6ND	2624.25	2769.00
7ND	2852.00	3002.50

Note: Unit of the frequencies is Hz.

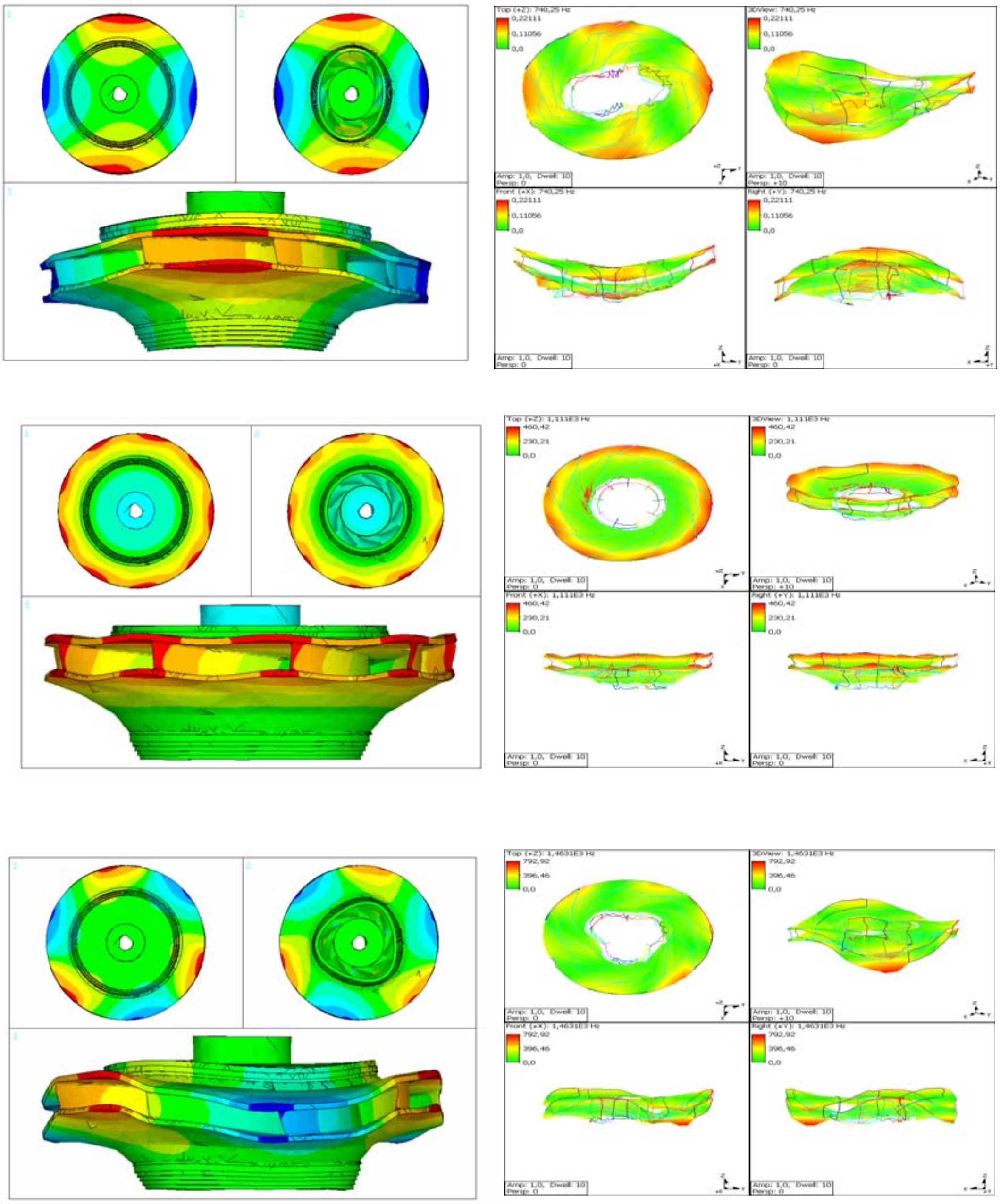

Fig.5 Mode shapes of the runner obtained by simulation
(From up to down: 2ND, 0ND, 3ND)

4. Simulation in Water and Comparison with Experiment

By the sensitivity analysis, the 55GPa is selected as the value of global Young's Module, and the simulation with this adapted parameter were carried out also in water. Natural frequencies and comparison with experimental results are tabulated in Table 3 and plotted in Figure 6.

	SIM.AIR	EXP.AIR	SIM.WATER	EXP.WATER	SIM.RATIO	EXP.RATIO
2ND	825.26	838.00	714.11	734.50	0.87	0.88
0ND	1247.40	1111.00	1103.80	1006.00	0.88	0.90
3ND	1479.75	1463.00	1332.90	1279.00	0.90	0.87
1ND	1605.55	1439.00	1482.05	1342.00	0.92	0.93
4ND	1871.80	1811.00	1668.90	1629.00	0.89	0.90
5ND	2350.15	2282.00	2062.05	2103.60	0.88	0.92
6ND	2624.25	2769.00	2458.80	2464.00	0.94	0.89
7ND	2852.00	3002.50	2661.15	2741.50	0.93	0.91

Table 3. Results of modal analysis of pump-turbine runner

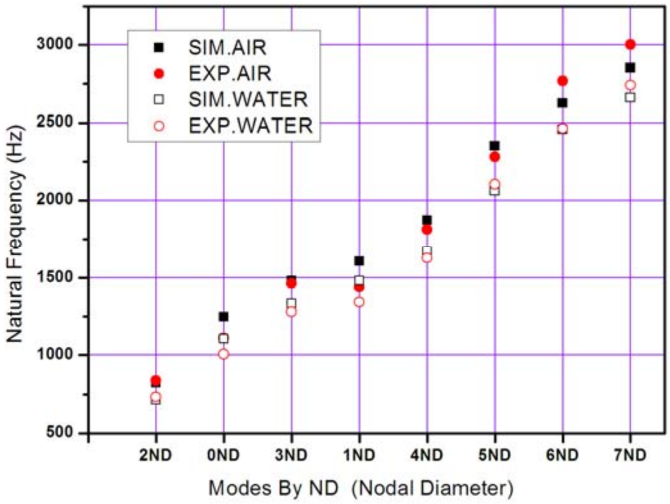

Fig. 6 Comparison simulation and experimental results in air and in water,

Unit of the frequencies is Hz (Herz);
Ratio = frequency in Water / frequency in Air.
With the frequency values obtained, the frequency reduction ratios are calculated and listed in Table 3 and the comparison is plotted in Figure 7.

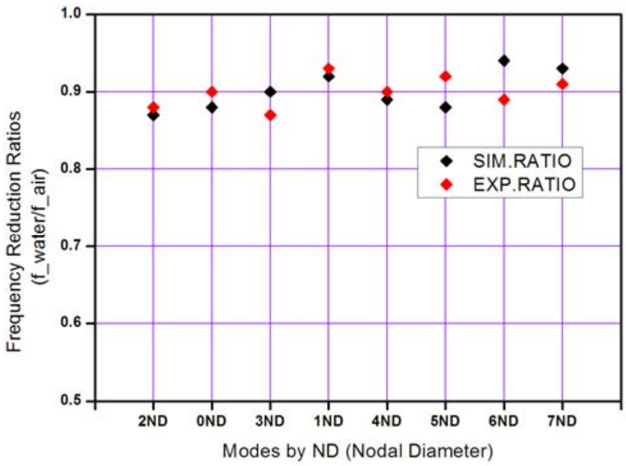

Fig.7 Comparison of frequency reduction radios

Ratio = frequency in Water / frequency in Air
In order to check the accuracy of the simulation with the experimental results, deviation between the simulation and experiment were calculated with Equation 4.

$$\Delta(\%) = \left[\left(U_{sim} - U_{\exp}\right) / U_{\exp}\right] \times 100 \qquad (4)$$

which is also listed in Table 4 and plotted in Figure 8.

Table 4. Deviations of simulation compared with experiment (%)

	2ND	0ND	3ND	1ND	4ND	5ND	6ND	7ND
Freq air	-1.52	12.28	1.14	11.57	3.36	2.99	-5.23	-5.01
Freq water	-2.78	9.72	4.21	10.44	2.45	-1.98	-0.21	-2.93

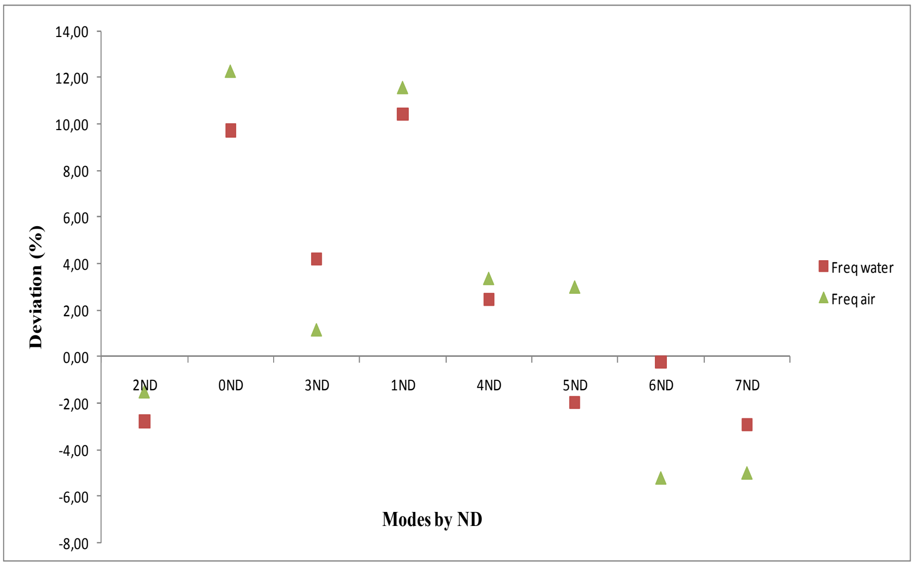

Fig.8 Deviations of simulation compared with experiment

5. Conclusions

The effect of the added mass effect of water on a pump-turbine model impeller has been analyzed numerically and experimentally. A theoretical analysis using FE method was also done. The numerical simulation considered the structure with and without the surrounding fluid domain. An optimized model was built by tetrahedral elements, containing 165000 elements for the impeller in air and of 508676 for the impeller in water was used.

Natural frequencies and mode-shapes were calculated. To check the accuracy of the simulation, natura l frequencies and mode shapes were compared with the corresponding experimental results. In general simulation gives rather good agreement with experiment both in air and in water. Mainly, the frequency modes represent the typical ND mode shapes for the cyclic geometry (2ND, 0ND, 3ND, 1ND, 4ND, 5ND, 6ND,7ND).

From the comparison of natural frequencies in air and in water, it can be clearly noticed that the nat ural frequencies are reduced by the presence of fluid. The frequency reduction ratio does not remain co nstant for all modes, but varies depending on the corresponding mode shape. The reduction ratio varies in a range of 0.87 to 0.94 depending on the mode shapes. The maximum reduction occurs in the 2N D mode. The added mass effect is explained from the analysis of the mode-shapes and energy theory. Modes with a relative motion between crown and band have the largest added mass effect.

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Nomenclature

[C_f]	Fluid equivalent "damping" matrix	[M_f]	Fluid equivalent "mass" matrix [kg]
[C_s]	Structural damping matrix	[M_{fs}]	Equivalent coupling "mass" matrix [kg]
ND	Nodal diameter	[M_s]	Structural mass matrix [kg]
{F_s}	Applied load vector [N]	{p}	Nodal pressure vector [N/m^2]
{F_{sf}}	Fluid "load" produced by structure displacement at the interface	Q	Flow rate [m^3/s]
g	Gravity [ms^{-2}]	{u}	Nodal displacement vector
H	Head [m]	x	Displacement [m]
k	Harmonic index	$\dot{x}$	Velocity [ms^{-1}]
[K_f]	Fluid equivalent "stiffness" matrix	$\ddot{x}$	Acceleration [ms^{-2}]
[K_{fs}]:	Equivalent coupling "stiffness" matrix	ρ_0	Mean fluid density [kgm^{-3}]
[K_s]	Structural stiffness matrix		

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