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NUMERICAL SIMULATION OF FLUID ADDED MASS EFFECT ON A KAPLAN TURBINE RUNNER WITH EXPERIMENTAL VALIDATION

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Abstract. *The mechanical design of hydraulic turbines is conditioned by the dynamic response of the runner which is usually estimated by means of computational models. Nevertheless, the runner has complex boundary conditions that are difficult to include in the computational model. One of these boundary conditions is the water in which the runner is submerged. The effect of the added mass and damping of water can modify considerably the natural frequencies of the runner. An experimental investigation in a Kaplan turbine runner, using modal analysis, was carried out. Several tests with the runner freely suspended in air were perfomed. The response was measured with accelerometers located in different positions of a runner blade. From the modal analysis, the natural frequencies, damping ratios, and mode-shapes were determined. Therefore, in this paper, a numerical simulation to analyze the influence of the surrounding water in a turbine runner has been carried out using finite element method (FEM). First, the sensitivity of the FEM model on the element shape and mesh density has been analysed and an optimized FEM model for the modal behaviour with the runner vibrating in air and in water has been calculated. After, the added mass effect by comparison of the natural frequencies and mode shapes in both cases has been determined. The numerical results obtained have been compared with experimental results available. The comparison shows a good agreement in the natural frequency values and in the mode shapes. The added mass effect due to the fluid structure interaction has been discussed in detail.*

Keywords: *Added mass, natural frequencies, turbine runner, finite element method, modal analysis.*

1. INTRODUCTION

There is a constant trend to increase the power concentration in hydraulic turbines either for new power plants or during upgrade of existing ones. As a consequence, heads and fluid velocities are higher and the hydraulic excitation forces on the turbine runners increase. These conditions produce major stresses in runners and possible vibration problems that could cause fatigue and fracture of the blades. The fracture begins as small cracks, brought by critical operation conditions of the machine over long periods, until failure. Such forces acting alone or combined with a reduced ratio thickness/weight in runners produce high vibration levels that can cause fatigue damage. The mechanical design of hydraulic turbines is conditioned by the dynamic response of the runner that is usually estimated by a computational model (Rodriguez *et al*, 2006).

Different solution methods are proposed for modal analysis of turbines. Diniz*, et al.,* 2007 applied a methodology of transference matrices and Finite Element Method on the dynamic model of a Kaplan turbine and compared these two methods for the case of one dimensional structure, where the method of transference matrices is characterized by having an algorithm simpler than the Finite Element Method. However, for Escaler*, et al.,* 2006, further complex structures such as impellers or runners, demand the use finite element models. In this sense, a numerical simulation of a real Francis turbine was carried out with an experimental validation on the work of Rodriguez*, et al.,* 2006 and Liang *, et al.,* 2007 who measured and simulated respectively the added mass effect on a scale Francis turbine. Through modal analysis was possible to determine the natural frequencies, damping factors and modal shapes. With the use of one-dimensional values, reduction on the frequencies can be extrapolated to other Francis turbines which present similar geometrical characteristics.

On the other hand, Salajka*, et al.,* 2009 and Flores*, et al.,* 2012 conducted a dynamic analysis on a Francis turbine runner, obtaining the natural frequencies and vibration modes in order to evaluate safety and preview failures from fatigue stress. On the second stage of the research, a prediction was made of the life cycle based on the approach of the

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local deformation previewing the beginning of cracks, made in static form, applying the load from the fluid-structure interaction on the runner blades.

Therefore, in this work a numerical simulation to analyze the influence of the surrounding water in a axial-flow hydraulic turbine runner on modal behavior has been carried out using finite element method (FEM) and compared with experimental results for the case without water. The experimental results were utilized on a model adjustment algorithm to increase the accuracy of the data, and then generate a model using fluid-structure interaction to simulate the presence of water inducing an added mass effect on the turbine runner. The case study presented on this paper uses a small axial-flow hydraulic turbine runner shown on Fig. 1, existing at the Federal University of Pará,.

Figure 1. Views of the driving region of the axial-flow hydraulic turbine.

2. FUNDAMENTALS

2.1 Modal analysis with added mass

Generally, modal analysis made on structures neglects the adjacent effect of fluid for gases or fluids with low density. For the prediction of the dynamical characteristics of hydraulic turbines, it must be taken into account the effect of the fluid that surrounds it. The solution of the finite element modal analysis from the runner-water coupled model gives as a result the natural frequencies and the modal shapes of the structure. In recent work, Flores*, et al.,* 2012 discuss the system as of fluid-structure interaction a problem, where the equation of the dynamic structure has to be coupled with the fluids equations. It is well known that the equation of the dynamic structure could be formulated as follows:

$$[M_s]\{\ddot{u}\} + [C_s]\{\dot{u}\} + [K_s]\{u\} = \{F_s\} \tag{1}$$

where $[M_s]$ is the structural mass matrix; $[C_s]$, the structural damping matrix; $[K_s]$, the structural stiffness matrix, $\{F_s\}$, the applied load vector, and $\{u\}$, the nodal displacement vector.

According Liang*, et al.,* 2007, in the case of water-structure coupling, the behavior of the water pressure can be described with the acoustic wave equation, known as Helmholtz's equation:

$$\nabla^2 P = \frac{1}{c^2}\frac{\partial^2 P}{\partial t^2} \tag{2}$$

where P is the fluid pressure; c, the sonic speed in the fluid medium; t, time; ∇^2, the Laplacian operator . According to the fundamentals of acoustics, Eq. (2) is derived from equation of motion and the continuity equation by considering the following assumptions (Kinsler, *et al.,* 1982; Liang *et al*, 2007):

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

The density and pressure are uniform in the fluid. Since the viscous dissipation has been neglected, the Helmholtz's Eq. (2) is referred to as the lossless wave equation for propagation of pressure in fluids. In case of fluid structure interaction problems, Eqs. (1) and (2) have to be considered simultaneously.

In the interface between the solid runner and water, the relation between the normal pressure gradient of the fluid and the normal acceleration of the structure gives the equation (Flores, *et al.*, 2012):

$$\{n\} \cdot \{\vec{\nabla} P\} = -\rho_0 \{n\} \cdot \frac{\partial^2 \vec{U}}{\partial t^2} \tag{3}$$

where U is the displacement vector of the structure's interface, and ρ_0 the density of the fluid. Considering the pressure of the fluid that acts in the interface, Liang, *et al.,* 2007 determine that Eq. (1) can be described by the form:

$$[M_s]\{\ddot{u}\} + [C_s]\{\dot{u}\} + [K_s]\{u\} = \{F_s\} + \{F_{fs}\} \tag{4}$$

where $\{F_{fS}\}$ is the load vector because of the fluid's pressure acting in the interface. The complete set of finite element discretized equations for the fluid-structure interaction problem is:

$$\begin{bmatrix} [M_s] & [0] \\ [M_{fs}] & [M_f] \end{bmatrix} \begin{Bmatrix} \{\ddot{u}\} \\ \{\ddot{p}\} \end{Bmatrix} + \begin{bmatrix} [C_s] & [0] \\ [0] & [C_f] \end{bmatrix} \begin{Bmatrix} \{\dot{u}\} \\ \{\dot{p}\} \end{Bmatrix} + \begin{bmatrix} [K_s] & [K_{fs}] \\ [0] & [K_f] \end{bmatrix} \begin{Bmatrix} \{u\} \\ \{p\} \end{Bmatrix} = \begin{Bmatrix} \{F_s\} \\ [0] \end{Bmatrix} \tag{5}$$

where $[M_{fS}]$ is the mass matrix in the interface and $[K_{fS}]$ is the stiffness matrix in the interface. The added coefficients are a consequence of being submerged in fluid domain. Therefore, for a problem involving fluid structure interaction the fluid element generates all the matrices with subscript f in addition to the coupling matrices. The matrices with subscript s are generated by the compatible structural element used in the model. In case, the FEM model describes fluid element and solid element for solution of pressure field and nodal displacement (with 6 degree of freedom).

2.2 Numerical model adjustement

Generally, the divergence in numeric-experimental results is noticed by many authors. That because the FEM uses solution schemes by approximation and simplifying hypotheses to give a faster solution for the studied models. So, the model adjustment, according to Infantes (2000), is a technique which aims to reduce the relative error between the experimental and numerical data. One of the most used methods is the objective function, in which aims the maximization of the interconnection between the measured data and the analytical parameters of the model. To perform the model adjustment studied on this paper, first it was necessary the specification of the project variables, the state variables and the objective function. The project variables were defined as the material properties of the Kaplan turbine runner, such as elastic modulus (E) and density (ρ). The state variables were defined as the natural frequencies calculated by the FEM software. Topics that must be observed about this methodology are:

- It was adjusted only the natural frequencies of the model (it was not used modal shapes as state variables).
- The results of the adjustment for the project variables do not necessarily represent the real behavior of the runner blade material.
- Due to the model simplifications, it was searched only the adequacy of the inertia and stiffness properties of the system.
- It is known that is practically impossible that the convergence of the objective function is minimized in which it tends to zero, due to the model complexity.
- It was considered a maximum error of 10%, based on the experimental data (commonly accepted on engineering projects).

The adjustment algorithm was created on the FEM software language, obeying the methodology presented on the Fig. 2.

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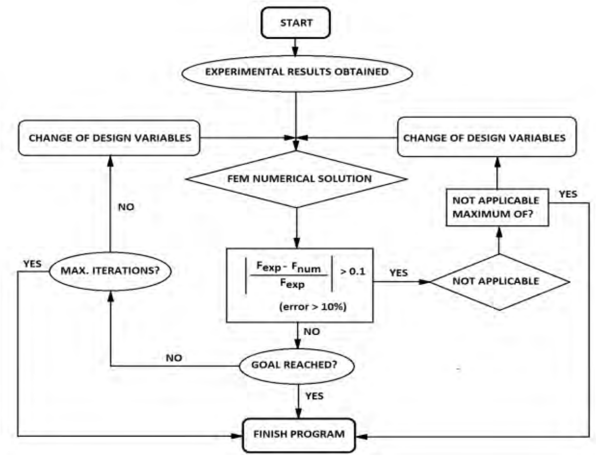


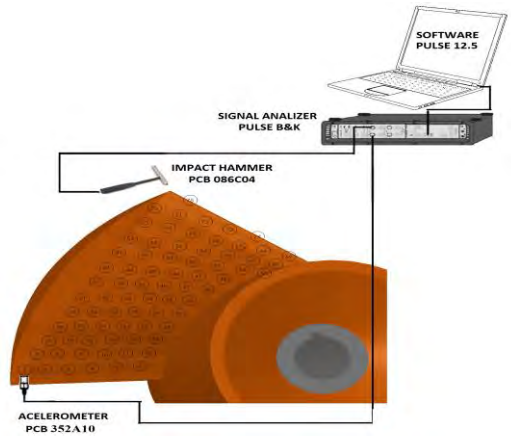


Figure 2. Flowchart adjustment Algorithm. Figure 3. Schematic of the test rig with the runner in air.

3. CASE STUDY

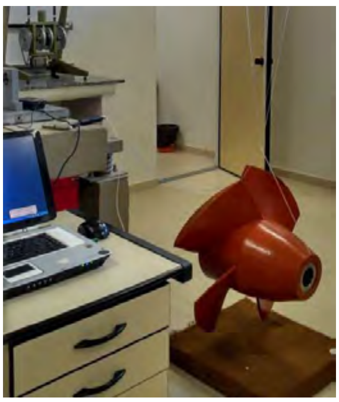
The tested model runner was constructed by Microturbines Inc. from Canada. It is a axial-flow hydraulic turbine runner with fixed blades deigned for a nominal power of 50 kW. The model runner has 4 blades and a diameter of 650 mm. The hub and blade are compost of composite material and the shaft-runner coupling is made stainless steel. Initially, the polymeric material is treated as an isotropic material, and subsequently, their behavior has proven to linear behavior through experimental tests performed. The mechanical proprieties of runner's polymeric material are: elastic module (E) = 1.1 GPa, Poisson ratio (μ) = 0.3 and density (ρ) = 1084.5 kg/m^3. The modal analysis was realized for air and water. The numerical analysis in air is compared with the vibration experimental results obtained for air. The runner's analysis in water considers the interaction of the structure and the fluid to obtain the natural frequencies. Also, the relation of the frequencies reduction was obtained due to the water that surrounds the runner.

3.1 Experimental setup

To determine the runner system parameters, a series of impact tests were carried out. An impact hammer especially designed for experimental modal analysis in laboratory (model PCB 086C04) and a small piezoelectric accelerometer (model PCB 352A10) used to excite and to measure the response of the runner, respectively. Figure 3 illustrates schematically the impact test. The proceeding the signals of impact and vibratory response are conditioned and analyzed with a Brüel & Kjaer module type 3109 with 4 input channels, which it is connected a computer (with analyzer software PULSE 12.5) for further processing.

In modal analysis the property of reciprocity can usually be assumed. It means that the transmission between a points "a" and "b" is the same as between the point "b" and the point "a".

This procedure has the advantage of measuring in a few points and changing the impact position without altering the test configuration. Although considerate a condition free to the runner (see Fig. 4), preliminary test shows that hub has much mass than blades and it makes just blade vibrate. It suggests that the runner vane behaves as if it were rigidly fixed. So the impact excitation is applied to 74 positions in the each blade. Points next to hub were considerate. Three impact tests are done in each impact position.



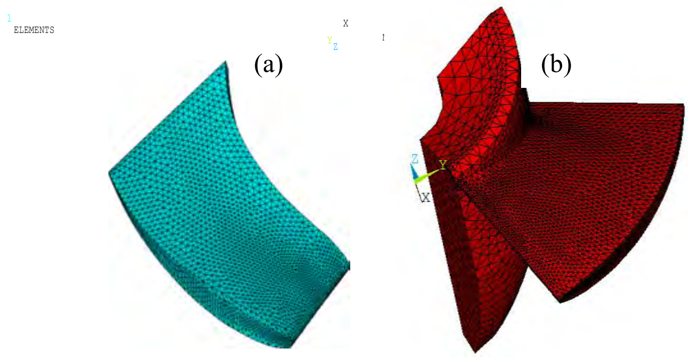


Figure 4. Turbine runner suspended by nylon. Figure 5. Turbine vane (a) and runner (b) meshed used on simulation.

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3.2 FEM model

Based on the cyclic symmetrical characteristic of the structure, a sector including one blade and covering an angle of 360/4 degrees was used to do the simulation. Tests were performed with complete model (hub+vanes) and considering just one runner vane. The results showed a small difference between the two cases and a considerable reduction in the processing time. Thus, the model considering only the blades was used. The mesh used structural elements (20 nodes by element) the sector runner, which contained 41312 nodes and 26960 elements for each sector. The final model mesh is shown on Fig. 5, from a mesh convergence test shown on Fig. 6.

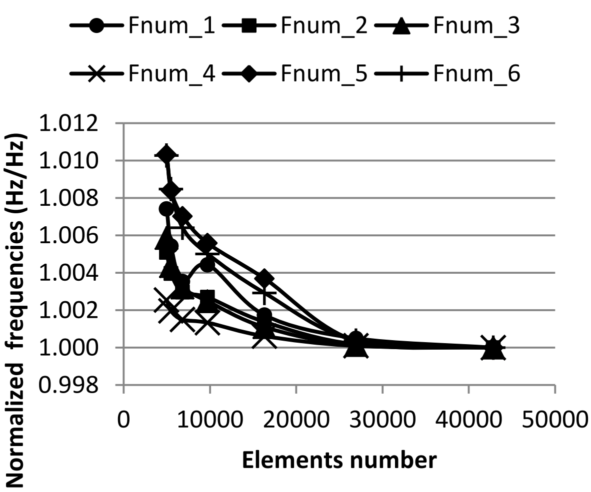

Figure 6. Results of the mesh sensitivity analysis.

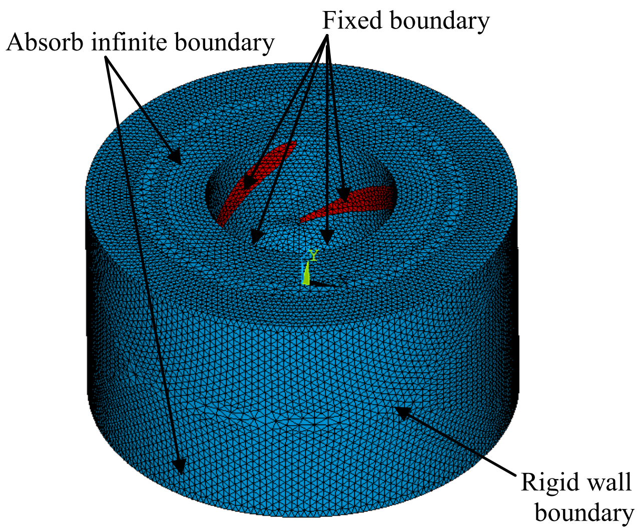

Figure 7. Turbine vanes meshed surrounded by water.

To make the complete model for the simulation in water, the mesh of the runner was surrounded by a cylindrical fluid domain considering just the fluid surrounding one runner vane. The fluid proprieties (environmental temperature and atmospheric pressure) necessary to fluid structure interaction analysis were: density ρ_0 = 1000 kg/m^3 and sound speed in water c_0 = 1483 m/s. The fluid mesh was extended from the structure mesh so that the same set of nodes was shared between both domains on the interface. The fluid element was used to build up the fluid mesh, which contained 209359 nodes and 172798 elements for each sector.

Based on a reservoir with as duct, the boundary conditions used in the simulation consider operating conditions, when the blades are fixed as rigid and the fluid is limited by duct wall. These conditions are described must be specified on the boundaries of the FEM model accordingly:

- Rigid wall boundary on fluid limit cylindrical surface:
- Fluid Structure Interface boundary on the fluid contact with runner surface
- Fixed boundary (the nodal displacement equals zero) where the runner is fixed on hub contact.
- Impedance zero (infinite absorb boundary) on the bottom surface of the cylinder:

The reflection of pressure waves has to be prevented at the cutting boundaries to stop excitation from infinity. Assuming locally reactive surfaces and using the impedance tube method, it is possible to obtain a relation between the normal absorption coefficient, α_n, and the specific normal acoustic impedance, Zn, given by (Melo, 2002)

$$Z_n = \rho_0 c_0 \cdot \frac{1+\sqrt{1-\alpha_n}}{1-\sqrt{1-\alpha_n}}. \tag{6}$$

This condition can be described by Eq. (6) considering α_n = 1. Figure 7 demonstrate all boundary conditions cited previously.

3.3 Modal analysis

The modal characteristics of runner vane were obtained using the modal analysis of ANSYS using the Block-Lanczos method for solution of the eigenvalue problem. The numerical-experimental results are showed on Fig. 8 (natural frequencies and modal shapes to 6 modes).

3.3.1 Simulation in air

Since the air effect on the experiment was negligible, the corresponding simulation was carried out in vacuum instead of in air. From FEM model with converged mesh, the modal form and natural frequencies in air were calculated. Details about the mode shapes will be discussed later.

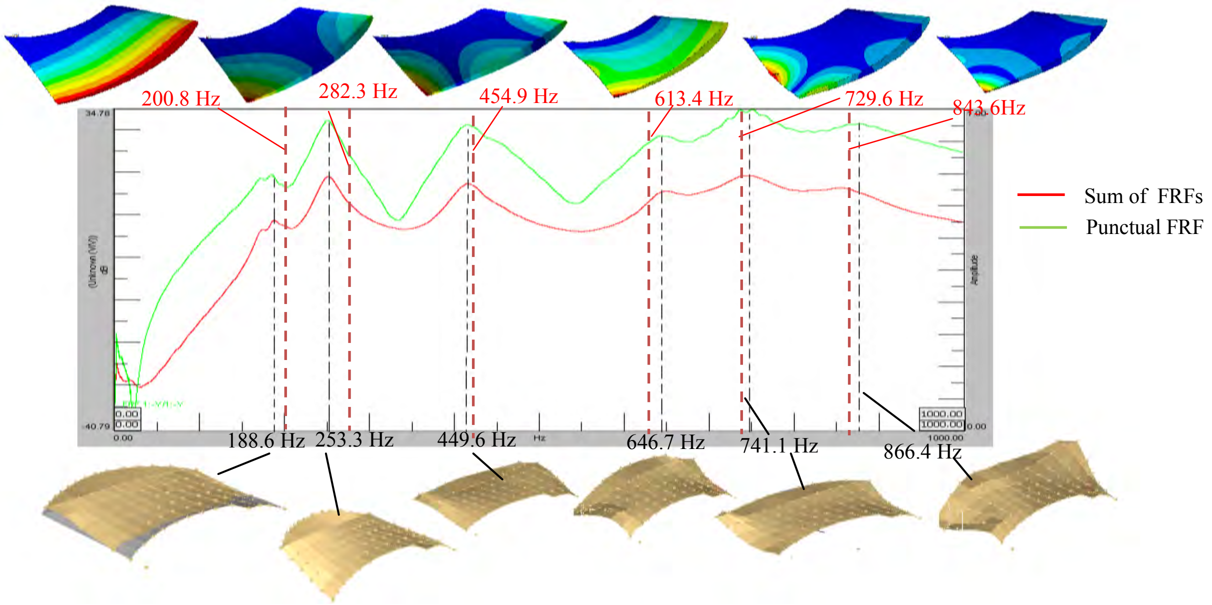

Figure 8. Vibrations modes of the runner vane in air by experimental analysis (bottom) and numerical analysis without adjustment (top).

In order to check the accuracy of the simulation with the experimental results, error between the simulation and experiment were calculated. The Equation (7) was used to list in Tab. 1 all the initial results,

$$E = \frac{|f_{num} - f_{exp}|}{f_{exp}} \cdot 100. \tag{7}$$

Where E (%) is the deviation in percent, f_{num} and f_{exp},indicate the numerical and experimental results. The frequencies obtained, not showed not good correlation between the simulation and the experimental results, with deviations above 10%, which is not acceptable to engineering conceptions. And then, using the adjustment algorithm, we can recalculate the design variables (it is means density and elastic module) from experimental results.

Table 1. Natural frequencies in air with deviation without adjustment model.

Analysis	Natural frequencies (Hz)					
	f_1	f_2	f_3	f_4	f_5	f_6
Numerical	188.63	253.37	449.64	646.73	741.10	866.39
Experimental	200.75	282.33	454.92	613.46	729.61	843.57
Error (%)	6.43	11.43	1.17	5.14	1.55	2.63

The new numerical results obtained with user adjustment model, show a better concordance between Simulation and modal experiments. It is observed a variation between $\pm 2\%$ and 8.2% depending on the frequency. The new

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mechanical properties calculate are 0.987 GPa e 1045.7 kg/m^3, Young module and density, respectively. Although, facts as: runner mechanical data are given by manufacturer and a composite material is being treated as isotropic material, it can be the mean reasons to appear this deviation.

Table 2 show the new results obtained from adjustment model.

Table 2. Natural frequencies in air with deviation with adjustment model

Analysis	Natural frequencies (Hz)					
	f_1	f_2	f_3	f_4	f_5	f_6
Experimental	188.63	253.37	449.64	646.73	741.10	866.39
Numerical	196.74	274.15	440.34	599.35	706.8	816.38
Error (%)	4.3	8.2	2	7.326	4.628	5.772

Although, the results show a good reduces the initial frequency deviations, other frequencies as f_3, f_4, f_5 and f_6 had a considerable increase. It is known that the frequency of operation of the machine happens next studied the first natural frequencies, which are the frequencies of interest in this work, justifying relevance and greater accuracy in them.

3.3.1 Simulation in water

According Flores, *et al.,* 2012, the theoretical and experimental study of the structures immersed in water indicates that the natural frequencies are reduced because of the interaction of this fluid with the structure. It is important to determine the natural frequencies of the runner in air and establish if there is a reduction of them when the runner is surrounded by water.

From this air simulation model calibrated, the simulation of the runner in water is performed. So, results obtained in the runner's simulations in air and submerged in water are compared.

According Liang, *et al.,* 2007, the fluid added mass effect can be estimated by calculating the frequency reduction ratio δ of each mode, defined

$$\delta = \frac{|f_{air} - f_{water}|}{f_{air}}. \tag{8}$$

Some author, in recent works, observed that a decrease in the natural frequencies in water do exist, when it is compared the results in air. Experimentally, Rodriguez, *et al.,* 2006 shows that ratio reduction decrease with natural frequency increment. It is just observed when the material has density too much bigger than water density. In this paper, due the lower polymer density, next to water density, the fluid added mass effect is reverse, as shown on Tab. 3. To prove this fact, we created a new numerical model with the property of steel (E_{steel} = 207 GPa and ρ_{steel} = 7800 kg / m³). The steel proprieties effect shows good concordance with results obtained on researches of Rodriguez, *et al.,* 2006 and Valero, *et al.*, 2010.

Table 3. Natural frequencies in air with deviation with adjustment model

Ratio (δ)	Natural frequencies (Hz)					
	f_1	f_2	f_3	f_4	f_5	f_6
Composite (real)	0.63	0.54	0.46	0.41	0.34	0.34
Steel (test)	0.20	0.21	0.33	0.40	0.45	0.45

The ratio of frequencies reduction varies from 0.34 to 0.63 depending on the frequencies. As might be expected, the modes of vibration observed were similar to the ones presented before, for the simulation of the runner in air. Figure 9 shows the natural frequencies of the runner in air and water.

The damping found in experimental results has high values in all the modes, around 4.3%. This means the typical behavior for a body vibrating with small amplitudes and high frequencies in stagnant fluid in the absence of wave radiation. It is known can be seen that the damping in water is always higher than in air, which it will result on smaller amplitudes that on air. Using nondimensional values (ratio added mass and damping surround water) it is possible to extrapolate the results of the particular runner tested to other runners (Rodriguez, *et al.,* 2006).

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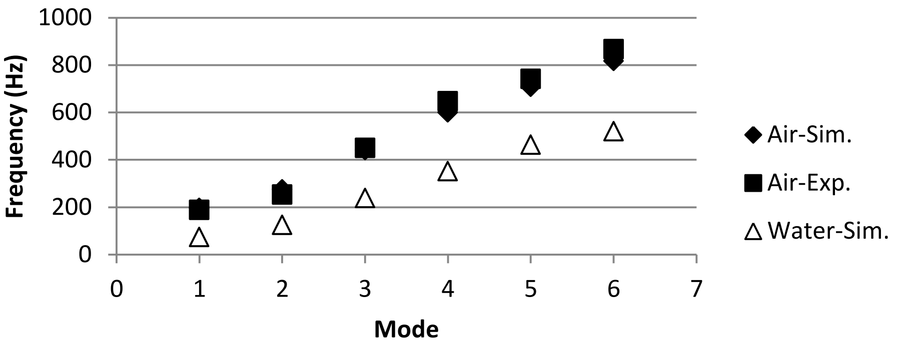

Figure 9. Natural frequencies in air (numerical e experimental) and water.

4. CONCLUSIONS

The added mass effects of the water on a small axial-flow hydraulic turbine runner using numerical modal analysis were determined with FEM method considering the structure and the surrounding fluid domain. From these analyses an optimized model with 26960 elements and 41312 nodes for runner vane was obtained.

The corresponding experimental results were used to check the accuracy of the simulation, which natural frequencies and mode shapes were calculated and compared. Simulation shows a good agreement with experiment, when used adjustment algorithm. It means analogous mode shapes between experimental and numerical ones having median deviation in the natural frequencies is around ± 5.3%. Mechanical proprieties are given by manufacturer and/or a composite material is being treated as isotropic material can be mean reasons to get this deviation.

The added mass effect of the surrounding water has been evaluated by comparing the optimized frequencies in air and in water. Natural frequencies are considerably reduced by the presence of water. The reduction ratio varies in a range of 0.63 to 0.34, depending on the mode shapes. A non-dimensional added mass factories derived, which can be practically extrapolated to estimate the natural frequencies of geometrically similar hydraulic turbine runners with different materials and dimensions.

As was descripted by Rodriguez, *et al.,* 2006, Salajka*, et al.,* 2009, Liang*, et al.,* 2007 and Valero, *et al.,*2010, it was observed a decrease in the natural frequencies of the runner in air and surrounded by water. The modal shapes in both cases were similar. The modal shape was, in order: flexural mode, torsion mode, second flexural mode, first flexural-torsion mode, second flexural-torsion mode and third flexural-torsion mode.

As a conclusions, it is stated that the computations of the analyzed type in which the ANSYS program system was used are viable for practical needs of water machinery development, design and acceptable as for both time and financial requirements.

Moreover, the method used in this investigation has been proved to be to be valid, which can be used to numerically study the dynamic behavior of any other axial-flow hydraulic turbine runner with other dimensions and materials.

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