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Influence of the boundary conditions on the natural frequencies of a Francis turbine

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Abstract. Natural frequencies estimation of Francis turbines is of paramount importance in the stage of design in order to avoid vibration and resonance problems especially during transient events. Francis turbine runners are submerged in water and confined with small axial and radial gaps which considerably decrease their natural frequencies in comparison to the same structure in the air. Acoustic-structural FSI simulations have been used to evaluate the influence of these gaps. This model considers an entire prototype of a Francis turbine, including generator, shaft, runner and surrounding water. The radial gap between the runner and the static parts has been changed from the real configuration (about 0.04% the runner diameter) to 1% of the runner diameter to evaluate its influence on the machine natural frequencies. Mode-shapes and natural frequencies of the whole machine are discussed for all the boundary conditions tested.

1. Introduction

In order to satisfy the new market requirements for more dynamic and flexible energy generation, Francis turbines are demanded to increase their power concentration as well as their regulation capacity. This fact leads to increase heads and fluid velocities and to widen the operation range of units. Consequently, excitation forces are higher and severe vibration problems may appear [1-3].

The vibratory response of hydraulic turbines during operation depends greatly on the dynamic properties of the runner and the boundary conditions such as the confinement or hydraulic seals. Francis turbine runners are rotating structures submerged in water and confined with small axial and radial gaps to the stationary parts. The size of these gaps affects considerably the dynamic response of the runner, which is normally characterized by the runner's natural frequencies and damping ratios associated to each mode-shape. The smaller is the gap between a submerged structure and a nearby wall, the lower the value of its natural frequencies is. This phenomenon is well-known in the field and it is explained by the added mass theory [4-6]. However, the damping ratio associated to each natural frequency and mode-shape tend to be higher when the gap is smaller as it was demonstrated also by Valentín et al. [6].

Natural frequencies of Francis turbine's runners are commonly estimated through of numerical simulations [7]. Finite Element Method (FEM) models, together with acoustical formulation to consider the surrounding water, are capable to predict with precision the natural frequencies of submerged structures, even when they are near a rigid boundary [4, 5, 8]. However, this kind of simulation does not consider complex terms such as damping; therefore added damping due to the surrounding water is not taken into account. This fact could lead to overestimate amplitudes of vibration and to neglect hydrodynamic dissipative effects.

Moreover, in these simulation models is not possible to include all the components of the machine such as bearings or labyrinth seals because of their complexity. Therefore some simplifications are usually made, introducing some uncertainties in the results. The influence of water and nearby rigid surfaces not only is relevant in the runner natural frequencies, but also it may affect to the natural frequencies of other components of the hydraulic turbine such as shaft or generator. In this paper, the influence of nearby rigid surfaces is studied through numerical simulations for a real prototype of a large Francis turbine. The simulation model considers the main components of the machine (generator, shaft, runner and surrounding water) to see the influence of changing the gaps' dimension between the runner and the nearby walls on the natural frequencies of the whole machine.

2. Characteristics of the turbine

The turbine of the present study is a medium head Francis turbine located in MICA Power Plant in British Columbia, Canada. This turbine has been selected to study it under the European Project Hyperbole [9]. The specific speed of the Francis turbine is about ν=0.29 according to Eq. (1) [4], being the nominal speed of the machine Ω=13.47 rad/s.

$$\nu = \frac{\Omega (Q/\pi)^{1/2}}{(2E)^{3/4}} \qquad (1)$$

2.1. Runner

The main dimensions of the runner and boundaries can be seen in Fig. 1. The relationship between the inlet diameter (D_{in}) and the outer diameter (D_{out}) is D_{in}/D_{out}=1.1, a characteristic value of a medium-high head Francis turbine. The runner has 16 blades (20 wicket gates in the distributor) and it is made of stainless steel. The radial gap between the runner and the static parts (g) in the real configuration of the prototype is rather small in comparison with the inlet diameter of the turbine (D_{in}) ($g/D=4\cdot10^{-4}$). Regarding to the axial gap between the runner's crown and the upper cover, as it can be seen in Fig. 1, it is small in the part of the crown's tip, but larger for lower diameters of the runner. Between the crown and the upper cover, there is a structure whose purpose is to reduce the secondary flow due to the rotation reducing friction losses and, at the same time, reducing the added mass and increasing the damping. This structure is formed by two annular rings connected by 30 ribs (see Fig. 1). The distance between this structure and the runner's crown is about 0.075% the runner inlet diameter ($h/D_{in}=7.5\cdot10^{-4}$). Therefore, it is important to estimate the influence of the radial and axial gap distance on the natural frequencies of all components of the machine.

2.2. Shaft

The shaft of this machine is also made by stainless steel and its diameter is 0.25 times the runner inlet diameter (D_{in}), whereas its length is 1.5 times the runner's inlet diameter (D_{in}). It is a single piece structure with a hollow in the center (Fig. 2). The machine is an umbrella-type machine, supported by two radial bearings in the turbine and in the generator sides and one thrust bearing in the generator side.

2.3. Generator

As the rotational speed of the machine is not too high (13.47 rad/s), the generator diameter is quite large compared with the runner (2.25 times the runner inlet diameter (D_{in})). The generator is formed by 52 poles that are supported by a spider type structure (see Fig. 2).

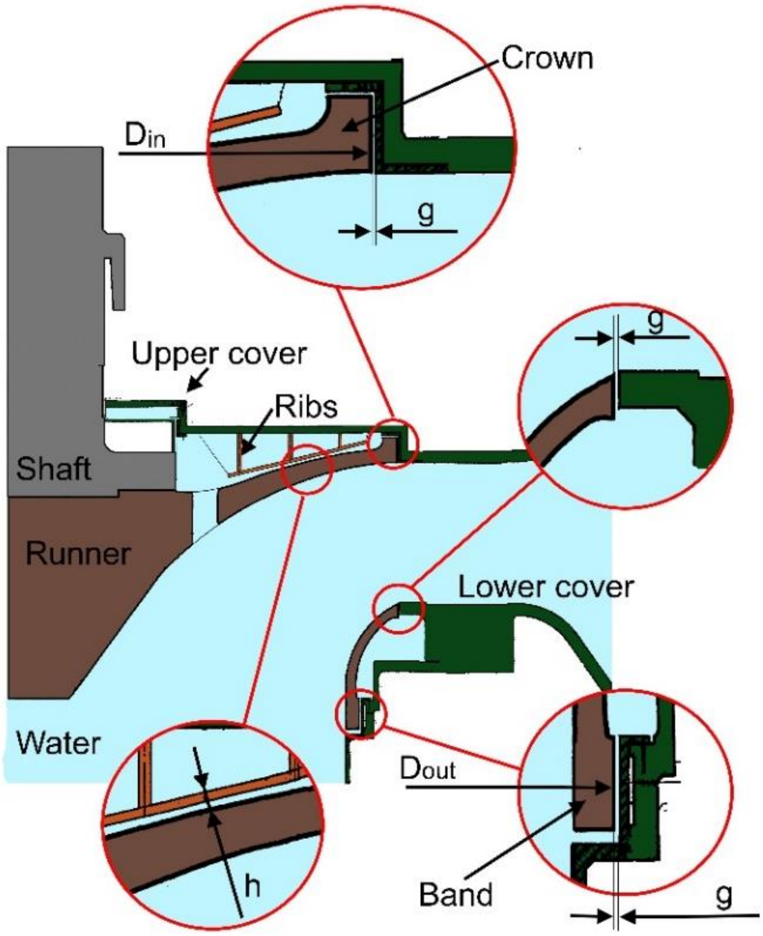

Fig. 1. Francis turbine of the study. Main dimensions and boundary conditions.

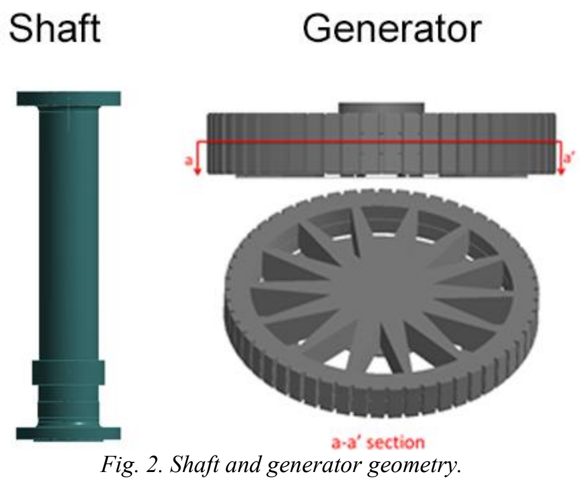

Fig. 2. Shaft and generator geometry.

3. Simulation model

To study the influence of radial and axial gaps between the runner and the static part on the natural frequencies of all the components of the machine, a structural-acoustical FSI simulation model has been built up. Ansys® [10] commercial software has been used to solve the simulation model. This simulation model considers the generator, the shaft, the runner and the surrounding water.

3.1. Mesh

The mesh of the generator and shaft is formed by 10-node tetrahedral elements (SOLID187 [10], see Fig. 3), whereas the mesh of the runner and surrounding water had to be constructed with 8-node hexahedral elements (SOLID185 for the runner and FLUID30 for the water) to consider the small gaps between the runner and static parts (g and h, see
Fig. 4). A mesh sensitivity analysis was carried out by increasing the number of elements of each part until the difference in natural frequency values between the case of study and the densest mesh tested was below 1%. The optimal mesh obtained for the whole machine was formed by approximately $9 \cdot 10^5$ elements (7,600 elements for the shaft, 77,800 elements for the generator, 230,000 for the runner and 584,600 elements for the water).

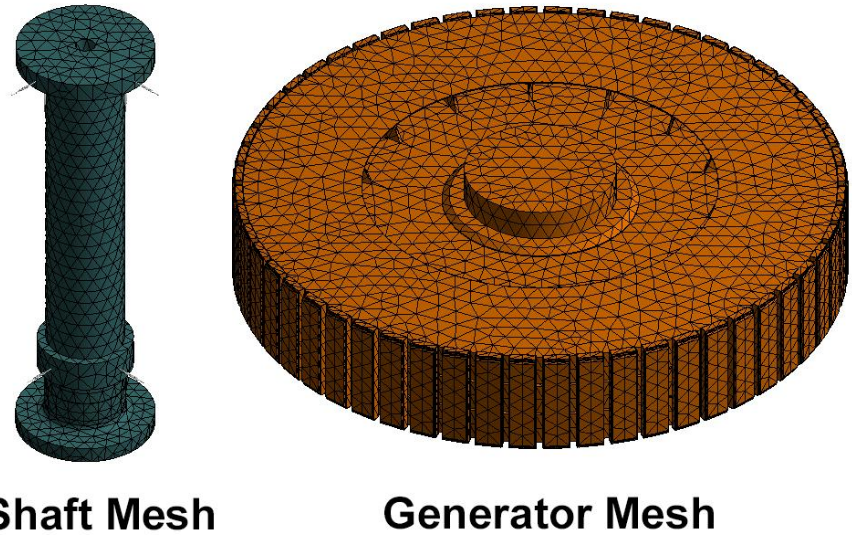

Fig. 3. Shaft and generator meshes.

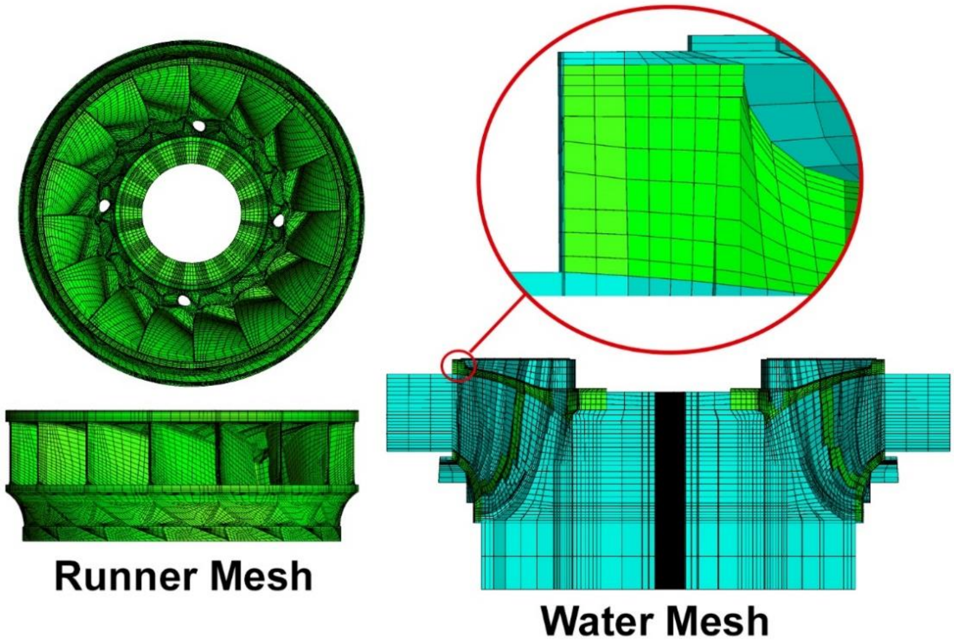

Fig. 4. Runner and water meshes.

3.2. Material properties

The shaft and the runner are made by stainless steel, therefore the properties for these two parts are the typical of this material (ρ=7800 kg/m^3, E=200 GPa, ν=0.3). However, the generator is formed by the poles and the spider structure, which are made from different materials. For the simulation, just one material was considered using the total mass of the generator according to the manufacturer data (ρ=3557 kg/m^3, E=200 GPa, ν=0.3). The water was considered as an acoustical fluid with ρ=1000 kg/m^3, and the speed of sound of 1430 m/s.

3.3. Boundary conditions

3.3.1. Bearings

The machine's bearings are modelled in the simulation using two radial springs (with a separation of 90 degrees between them) in the turbine bearing, other two radial springs at 90 degrees in the generator bearing and two axial springs to model the thrust bearing. According to experimental results obtained in previous studies [11], the value of these springs stiffness normally ranges between 10^9 N/m and 10^{10} N/m depending on the type of bearings and the machine. For the simulations of the present study, 10^{10} N/m was selected as the stiffness of all springs.

3.3.2. Water boundary conditions

The nodes of the water that are in contact with the static parts (upper and lower cover) are fixed without any displacement in all directions, whereas the nodes of the water in contact with the runner are defined as FSI (Fluid Structure Interaction) interface. The nodes at the inlet and outlet of the runner are configured with atmospheric pressure (Fig. 5).

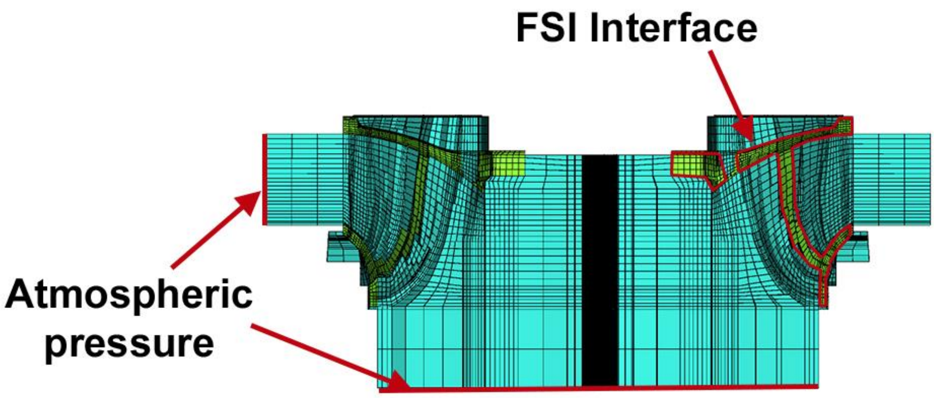

Fig. 5. Boundary conditions applied to the water.

4. Results

4.1. Mode-shapes

The mode-shapes of this machine can be separated in mode-shapes of each structural part (generator, shaft and runner mode-shapes) according to where they have the maximum displacement. Generator and runner mode-shapes are cylinder-like structures [12], this means that their mode-shapes are defined with nodal diameters (ND) (points which are stationary during the cycle of the mode-shape). However, the mode-shapes of the shaft are rather dependent on the bearing stiffness and they are normally classified by their apparition order in frequency. The mode-shapes within the frequency band 0-100Hz in water are shown in Fig. 6.

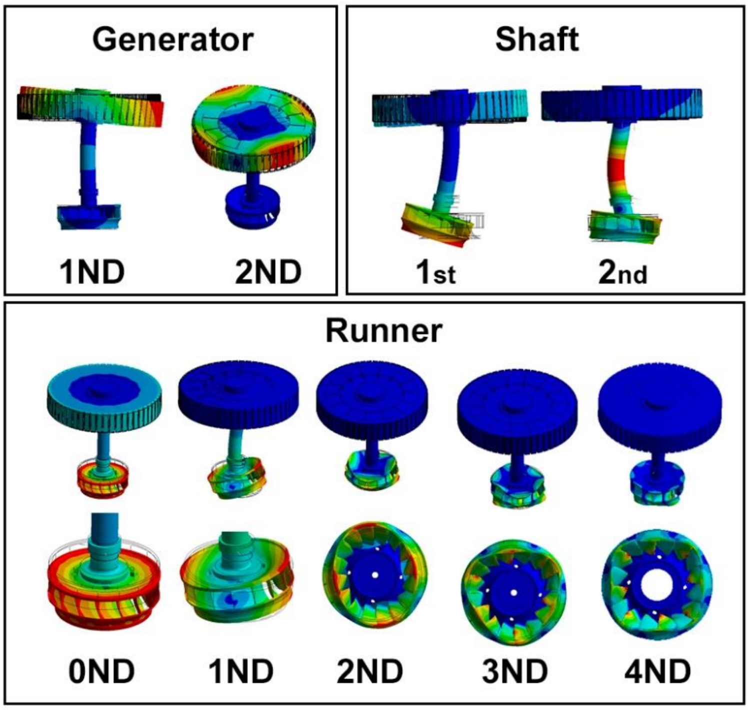

Fig. 6. Mode-shapes studied.

4.2. Radial gap influence

The radial gap (g) was increased from $4\cdot10^{-4}$ times the runner inlet diameter (D_{in}) to 10^{-2} times. In all these simulations the axial gap was maintained as in the real configuration and the ribs structure was not considered. The simulation was also run in air (without surrounding water) to see the influence of the added mass.

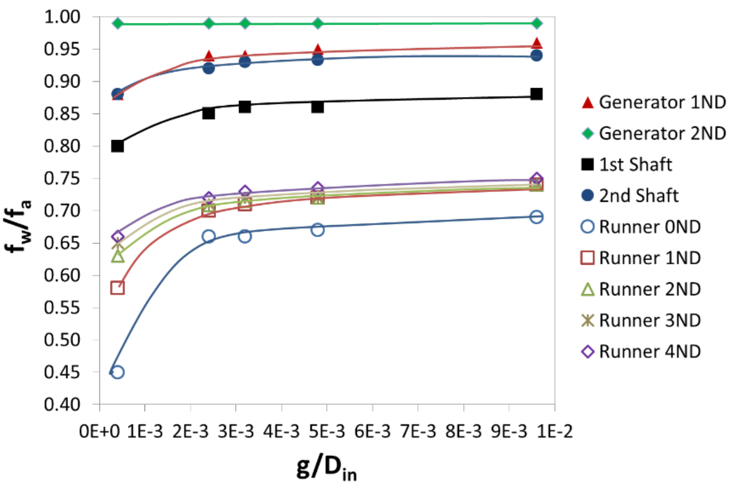

Fig. 7. Radial gap influence on the relationship between the natural frequency in water (f_w) and in the air (f_a)

Fig. 7 shows the influence of the radial gap in the natural frequencies of the machine. Natural frequencies in water (f_w) over natural frequencies in air (f_a) are plotted for all the mode-shapes. It is observed that the runner's mode-shapes are the most affected by the surrounding water. A reduction of between 30-35% in the natural frequency against the air configuration have been obtained. Moreover, it is seen that natural frequencies are smaller for smaller radial gaps. With the smallest radial gap tested, natural frequencies of the runner present between 35-55% of reduction against the air's case. The first mode-shape of the generator (Generator 1ND) and the mode-shapes of the shaft are also affected by the added mass of the water, being the reduction from 5% to 20% against the air's case. This is because, in these mode-shapes, the runner also has deformation (see Fig. 6) and therefore the radial gap is affecting it. However, the 2ND mode of the generator, which in fact does not have deformation in the runner, is practically unaffected by both water and radial gap.

4.3. Ribs structure influence

To see if this structure has an influence on the natural frequencies of the machine, another simulation was carried out but including the ribs structure. For this simulation, the radial gap was selected as $2.4 \cdot 10^3$ times the runner inlet diameter (D_{in}) and all other parameters were maintained as in the base case configuration. Table 1 shows the natural frequencies of the machine in both cases with and without ribs structure. It is seen that the difference for the runner natural frequencies are about 7%, whereas for the mode-shapes of the shaft and generator this difference is practically negligible. Therefore, this structure is affecting the added mass of the runner but the influence is not too high.

Table 1. Natural frequencies of the machine considering or not the ribs structure.

Mode-shapes	Ribs f_n (Hz)	No ribs f_n (Hz)	Difference in % $D(\%) = \frac{f_{n\,No\ Ribs} - f_{n\,Ribs}}{f_{n\,No\ Ribs}}$
Runner 0ND	33.8	36.0	6.1
Runner 1ND	21.1	22.7	7.0
Runner 2ND	28.2	30.3	6.9
Runner 3ND	62.5	65.2	4.1
Runner 4ND	93.3	94.5	1.3
1st Shaft	9.2	9.3	1.1
2nd Shaft	68.5	68.9	0.6
Generator 1ND	7.8	7.9	1.3
Generator 2ND	57.4	57.4	0

5. Conclusions

The natural frequencies and mode-shapes of a large Francis turbine have been determined by means of structural-acoustical FSI simulations. The simulation model considered the generator, the shaft, the runner and the surrounding water. Different radial gaps between the runner and the static parts have been tested. Moreover, the influence of considering the ribs structure between the runner's crown and the upper cover has been studied.

It has been demonstrated that the radial gap has a big influence in the runner natural frequencies (about 40-50% of reduction against the natural frequency in air depending on the mode-shape) and lower, but it is still important, in the shaft and generator natural frequencies. Regarding the ribs structure, which decreases the distance between the runner's crown and the upper cover, a small decrease in natural frequencies of the runner was appreciated (7% of difference between considering or not this structure).

Structural-acoustical FSI simulations have been demonstrated to be valid to study the added mass effects for different boundary conditions.

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