Experimental and numerical study on the detection of fatigue failures in hydraulic turbines

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Abstract Detecting fatigue cracks in hydraulic turbine runners is costly, as it requires to stop the unit, empty it of water and access the runner for inspection. Thus, an alternative way based on monitoring the changes of the structural modal response induced by the formation and growth of a crack has been investigated. To do so, the crack propagation induced by a resonance has been numerically predicted and experimentally machined on a disk-like structure that resembles a Kaplan turbine runner. The analysis of the results shows how the different stages of the fatigue crack growth can be monitored based on the change of the natural frequencies and mode shapes of several specific modes. Based on the obtained results, a Structural Health Monitoring system is going to be designed to monitor the turbine runner modes of vibration without the need to stop and inspect the unit.

1. Introduction

As a part of the H2020 EU-funded AFC4Hydro project, a Structural Health Monitoring (SHM) system for hydraulic turbines working at off-design operating conditions and during transients is under development [1]. Under such circumstances, deleterious hydrodynamic phenomena can take place, such as the rotating vortex rope (RVR) or the rotor stator interaction (RSI), which might damage severely the structure and deteriorate the turbine performance. In particular, the unstable loads being applied to the runner and to the shaft are responsible for periodic alternating cycles of stress that may cause fatigue cracks if sustained in time. As demonstrated by Bouboulas et al. [2], the initiation and development of a crack will modify the dynamic response of the structure during its growth, i.e. the natural frequencies and the mode shapes will change, and in the worst-case produce a breakdown. Detecting fatigue cracks is sometimes not an easy task, and involves stopping the unit and having to access the runner in order to apply non-destructive tests based on ultrasonic, thermographic, radiographic or eddy current methods [3].

In the case of a Kaplan turbine runner, the blades are ~~one of~~amongst the most susceptible parts to be damaged by fatigue due to the high and variable loads induced by unsteady flows. Concretely, the area closer to the connection between the blades and the runner, where the blades are supported, is the most critical zone owing to ~~as it presents~~ the higher stresses produced by the ~~a~~ discontinuity in the geometry ~~that suffers the highest stresses~~ [4, 5]. The evolution and propagation of a crack in a blade will produce a reduction of ~~the~~ its cross~~-~~section ~~and~~ plus a loss of its internal resistance. As the length of the crack increases, the residual resistance of the blade will decrease until ~~it will reach a point where the residual resistance of the blade will be lower than~~ the external stresses surpass this resistance, moment in which the remaining cross section might collapse due to plastic deformation or suddenly produce the final fracture due to the instability of the crack growth. At that moment, the crack will reach its critical length. However, an alternative and conventional definition of the critical length has been considered in this work based on the fact that the blade ceases to be functional in some sense (interference with the envelope, loss of performance, etc.). This alternative definition can be related to the blade modal stiffness k. For the purposes of this study it has been considered that once k is reduced to its half, which corresponds to the critical stiffness $k' = k/2$, it can be said that the mechanical properties of the component have radically changed ~~radically~~ and the structure might be about to a collapse or breakdown. Substituting k' into Eq. (1) corresponding to the natural frequency of a spring-mass damped system, where m is the modal mass of the mode shape, the critical natural frequency, f_{crit}, can be computed as a function the initial natural frequency of the structure without any crack, f, obtaining that $f_{crit} = f/\sqrt{2} = 0.7071f$. At that moment, the propagation of the crack will reach the crack critical length, L_{crit}, decreasing the stiffness of the structure to k' and its natural frequency to f_{crit}.

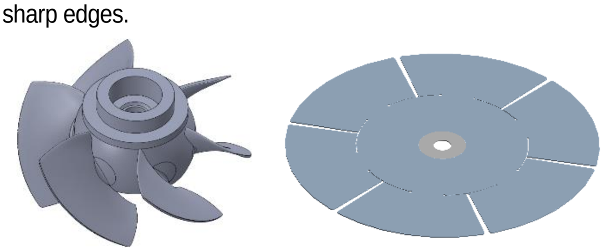
$$f = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \quad (1)$$

The objective of the current research ~~has been~~was to improve the understanding on the change of the modal response of turbine runners suffering the effects of a fatigue crack, going from its formation to its full development when the crack reaches L_{crit}. For that, a methodology to detect and monitor cracks has been tested and validated in a disk-like structure similar to a Kaplan turbine runner with 6 blades, installed in a bespoke ~~devoted laboratory~~ test rig available at the laboratory of the Barcelona Fluids & Energy Lab (IFLUIDS-UPC)~~, which is similar to a Kaplan turbine runner with 6 blades~~. The methodology ~~has~~ is ~~been~~ based on tracking the effects that different crack lengths, that would grow induced by a certain mode of vibration, produce to the modal response of the simplified structure.

2. Methodology

2.1 Geometry Simplification

At this stage of the research it was not possible to test a real turbine so it was decided to start with a simplified geometry that could be assimilated to it from the mechanical point of view, sacrificing similarity in terms of flow. The first approach done was to simplify the geometry of a typical Kaplan turbine runner, shown in Fig. 1a, into a disk-like geometry with six segments to simulate the blades as shown in Fig. 1b. The disk external diameter was 420 mm and the internal diameter of the segments representing the blades was 230 mm. The total thickness of the disk was 1.5 mm. Additionally, a series of small rounded bounds with a diameter of 4 mm were machined in the intersections between the blades and the circular central part of the disk to avoid a concentration of stresses induced by the discontinuities due to sharp edges.



(a) (b)

Fig. 1. (a) Typical Kaplan turbine runner; (b) Simplified disk-like body structure to carry out the study.

The fact that the geometry was similar to a disk permitted to take advantage from the well-known mode shapes of such type of geometries in order to design the experiments and to analyze the results. Preliminary, the crack propagation in the simplified structure was numerically simulated with a 3D model.

2.2 Numerical Simulation

The numerical structural model was built using the commercial software ANSYS®. A modal analysis was solved to calculate the natural frequencies and the strain distribution of the different vibrational mode shapes. In the numerical simulation it was considered that the material of the disk was standard stainless-steel, obtained directly from the Engineering Database of ANSYS®. A mesh sensitivity analysis was performed to obtain an optimum discretization of the geometry based on a trade-off between the computational time required to solve the model and the accuracy of the obtained results. Finally, the geometry was discretized using tetrahedral elements with a maximum size of 3 mm and further refining the mesh around the intersections between the blades and the hub, which theoretically are the locations where the crack should be more prone to appear. Thus, for the initial simulation of the disk without any crack a total of 327,474 mesh elements were used.

Even though the disk in the laboratory was attached to a shaft supported by two bearings, the shaft was not considered in the model as its influence on the different natural frequencies of the disk was found to be negligible. Hence, the boundary conditions applied to the numerical model were simplified and only the surface of the disk in contact with the shaft was fixed. Fig. 2 shows the mesh of the numerical model with the refined zones, in which the gray color zone in the center of the disk represents the contact with the shaft considered as a fixed boundary.

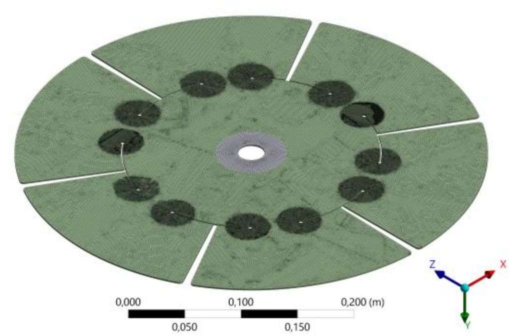


Fig. 2. Mesh of the numerical model, where the ~~with the~~ gray colored zone corresponds~~ing~~ to the fixed boundary condition of the shaft attachment~~ed to the shaft considered as a fixed boundary~~.

2.3 Crack Propagation

Experimentally recreating a fatigue crack and tracking its development by applying continuous cyclical stresses to the structure would have been very time-consuming and complex. Therefore, in order to accelerate the study, it was decided to estimate the crack propagation path using a numerical simulation and then artificially machine the simulated crack development at different stages of its growth using a laser ~~machine~~cutter. For each crack stage, the variations on the modal response of the structure were measured experimentally.

The approach followed to simulate the fatigue crack consisted of assuming that it is caused by the vibration induced by a resonance of the first natural frequency of the disk, which is more prone to be excited since the hydraulic excitations occur usually at low frequencies. In disk-like structures, the first natural frequency produces a certain mode shape corresponding to the one typically called 1ND. The adopted nomenclature follows the number of nodal diameters, ND, and/or nodal circles, NC, corresponding to the number of straight lines crossing the disk center or the axisymmetric circumferential lines, respectively, where the displacement of the disk-like structure is zero during a vibration cycle of a mode shape. Additionally, a simultaneous bending movement of all the blades of the disk was identified and called BLADES mode, where each pair of adjacent blades bend out of phase. Fig. 3 shows the identified mode shapes of the disk-like structure obtained in the numerical simulation without any crack, where the color scale indicates the relative displacement of the corresponding mode shape.

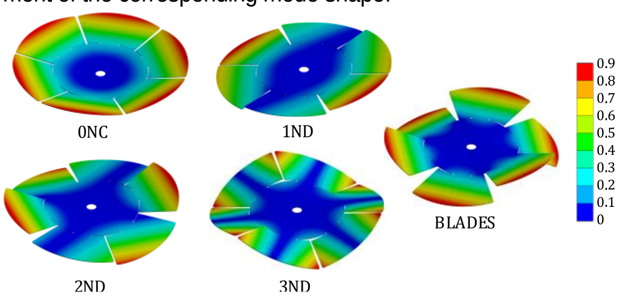


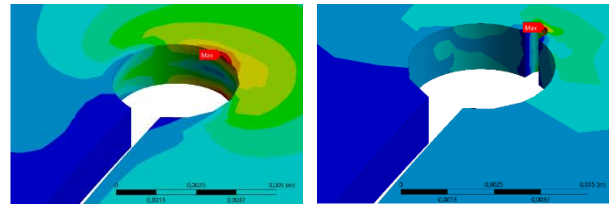
Fig. 3. Identified mode shapes for the disk in the numerical simulation.

Since for the object of the study it was not necessary to determine the crack path precisely, the evolution and shape of the crack was estimated based only on the stress field. For a more precise study, fracture mechanics approaches should be used to numerically estimate the crack path with better accuracy [6].

The evolution and shape of the crack was numerically obtained with an iterative procedure between a Computer Aided Design (CAD) program, such as SolidWorks®, and a Finite Element Model (FEM) of the structure, built in ANSYS®. To begin, a modal analysis of the FEM was solved to locate the point and direction with maximum Von-Mises stress induced by the movement of the first vibrational mode, i.e. the 1ND mode shape. With that information, the most favorable direction of crack growth was identified and the geometry was modified by creating a small micro-crack using the CAD program. Then, the new FEM was solved to recompute the stress distribution of that new geometry and the process was repeated again enlarging the crack in the correct direction. At each successive iteration, an extra refinement of the mesh was applied around the tip of each new micro-crack.

Fig. 4a shows a zoom of the initial equivalent Von-Mises stress distribution around the connection of the blade and the hub without any crack, which was considered as the starting point of the iterative procedure. The location with highest stress has been pointed out, where the first micro-crack of 0.5 mm was created perpendicularly to the maximum principal strain direction.

As an example, Fig. 4b shows the new stress field distribution obtained from the new modal analysis once the geometry has been updated, in which the propagation of the first micro-crack has changed the stress distribution of the 1ND mode shape. Hence, ~~and~~ a new point of maximum stress and its direction can be identified and used as a reference to continue with the next iteration.



(a) (b)

Fig. 4. (a) Zoom of the initial Von-Mises stress distribution around the area that will suffer the initiation of the crack; (b) Zoom of the same area but with the first micro crack, after the first iteration.

The iterative procedure to grow the crack was repeated until the 1ND natural frequency decreased from its initial value, f = 30.1 Hz, to its critical natural frequency, $f_{crit} = f/\sqrt{2} \approx 21.2$ Hz, moment at which the rigidity of the structure vibrating at that frequency is halved. Fig. 5 shows the evolution of the 1ND natural frequency as a function of the crack length until it reaches its critical length L_{crit} = 72.5 mm, which presents a sigmoid shape that matches with the one corresponding to the resistance degradation of a structure at the different phases of fatigue cracks presented by Ayneto [7].

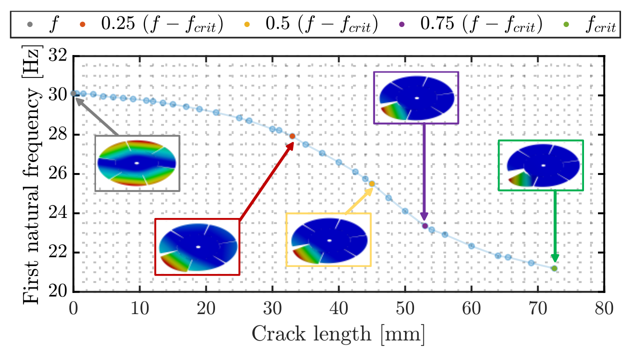


Fig. 5. Evolution of the numerical first natural frequency as a function of the crack length, showing the ~~and plots of the~~ mode shapes for different crack lengths.

The trend shown in Fig. 5 basically shows that at the beginning of the crack formation, where only a micro-crack is present, the loss of strength is almost negligible and the frequency evolution follows a flat behavior. Once the crack starts growing, the degradation and loose of internal resistance of the body starts, and a linear decrease occurs. Finally, when the residual resistance of the structure gets closer to the extreme stress values, this tendency slows down and, whenever the residual resistance is smaller than the external stresses, a total fracture can occur.

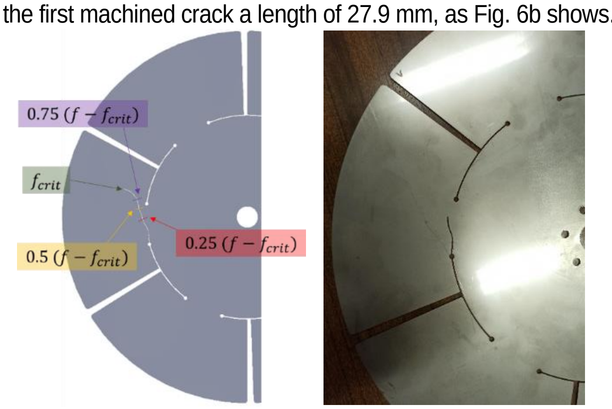
Additionally, five crack lengths that produce a 0, 25, 50, 75 and 100% decrease of $f - f_{crit}$ have been marked with different colors in Fig. 5, which have been named as f, $0.25\,(f - f_{crit})$, $0.5\,(f - f_{crit})$, $0.75\,(f - f_{crit})$ and f_{crit}, respectively. The evolution of the numerical mode shapes at each stage of the crack has also been depicted in the same figure, which clearly show how the initial 1ND mode turns out to become a torsion of the single cracked blade meanwhile the rest of the blades remain fixed.

2.4 Experimental Modal Analysis

Once the shape of the crack and the numerical evolution of the initial 1ND natural frequency were obtained using numerical simulations for each crack length, an experimental modal analysis based on the rowing-hammer technique was carried out to validate the numerical model and the results of the simulations. In order to monitor precisely the effect of the crack propagation on the modal response of the structure, 4 different crack lengths were considered. Hence, the original disk without any crack was machined 4 times to create the different cracks and for each case a complete experimental modal test was carried out.

The shape of the simulated crack is presented in Fig. 6a, in which the intermediate crack lengths have been marked over the total crack length corresponding to L_{crit}. This shape resembles reasonably the crack observed and simulated by Luo et al. [8] in a Kaplan turbine runner.

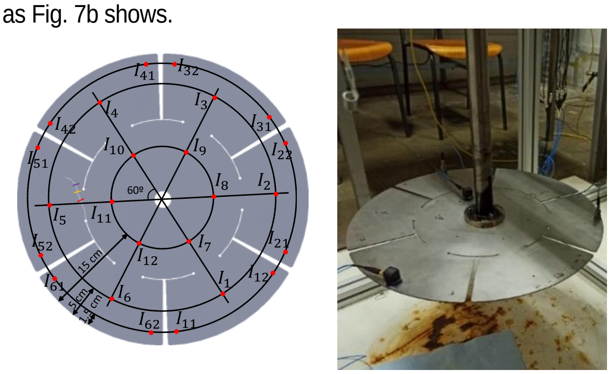
The intermediate cracks were sequentially machined, having the first machined crack a length of 27.9 mm, as Fig. 6b shows.



(a) (b)

Fig. 6. (a) Numerically simulated crack propagation and length of the intermediate cracks that have been sequentially machined in the real disk; (b) picture of the first $0.25\ (f - f_{crit})$ crack machined in the disk, presenting a ~~corresponding to~~ 25% of frequency reductionit~~s critical length~~.

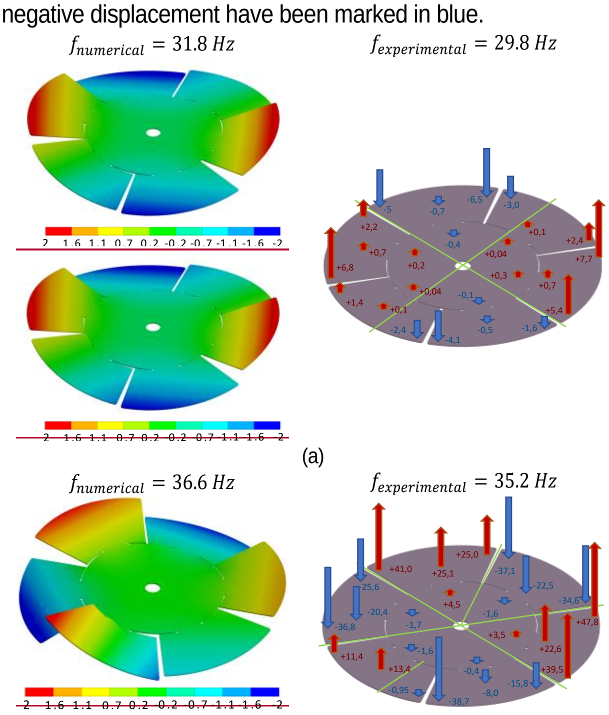
For each machined crack, experimental modal tests were performed using the rowing hammer technique using an instrumented impact hammer, to excite the disk, and 4 glued miniature accelerometers, to measure its ~~response .~~response. As Fig. 7a shows, a total of 4 points were impacted per blade at the different positions marked in red. In order to guarantee good measurement repeatability, each point was impacted at least 5 times. The accelerometers were always placed on the same locations, as Fig. 7b shows.

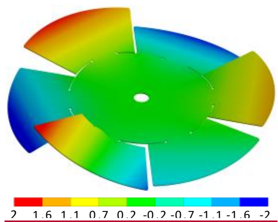


(a) (b)

Fig. 7. (a) Scheme of the impacted positions of the disk ~~excited~~ using the instrumented hammer; (b) photograph of the experimental setup used to measure the modal response of the disk placed in the test rig.

The measured excitation and response signals were analyzed using a Fast Fourier Transform (FFT) algorithm to transform the data from the temporal domain to the frequency domain. Consequently, the ratio of the output response of the structure to the applied hammer force was computed, obtaining an average frequency response function (FRF) for each series of impacts in a given position. The collection of all the FRFs at the measured locations contains information regarding the mode shapes of the structure [9]. By means of applying a force-exponential window to minimize signal leakage and using an RMS averaging method considering the 5 impacts done per point, the imaginary part of the FRF of each impact was obtained to determine the mode shape associated at each natural frequency. The obtained imaginary part of the FRFs allowed us to identify the mode shape associated with each frequency measured experimentally and compare it with the numerical simulation. Fig. 8a shows an example of the numerical and experimental mode shapes obtained with the first $0.25\ (f - f_{crit})$ crack, whereas Fig. 8b shows another comparison for the second crack at $0.5\ (f - f_{crit})$. The plots on the left-hand side show the relative vertical deformation of the disk obtained from the numerical modal analysis, using a color-map to represent in red the positive vertical displacements and in blue the negative ones. On the other hand, the plots on the right-hand side show the values obtained from the experimental modal analysis, where the aforementioned imaginary part of the averaged FRF at the different impact positions on the disk is displayed. To make the comparison easier, the values corresponding to a positive displacement have been marked in red, while the values corresponding to a negative displacement have been marked in blue.





(b)

Fig. 8. Comparison of the numerical and experimental mode shapes obtained at (a) 25% of frequency reduction and (b) at 50% of frequency reduction.

It can be seen how in Fig. 8a, the mode shape corresponding to $f_{numerical} = 31.8$ Hz presents a 2ND mode shape, in which the ~~–~~two nodal lines with zero displacement are marked~~depicted~~ in green. In Fig. 8b, the mode shape corresponding to $f_{numerical} = 36.6$ Hz has the particularity that the cracked blade suffers a torsion whilst the rest of blades present a bending, which corresponds to the BLADES mode. All in all, for both cases the shape and the frequency of the numerical and experimental modes match almost perfectly, which validates the present study.

3. Comparison of the numerical and experimental modal tests

After performing the experimental modal analysis of the disk with different crack lengths, the different natural frequencies and their corresponding mode shapes were compared with the numerical results. Fig. 9 presents the evolution of the different natural frequencies obtained numerically (dashed lines) and experimentally (continuous lines) for the first 4 modes of the structure, which are the 1ND, 2ND, BLADES and 3ND that are also depicted in the figure. As previously stated, the ND mode shapes present different nodal diameters with zero displacement on the disk-like structure, having the particularity that the cracked blade always presents a dominating torsion deformation with an amplitude depending on the length of the crack.

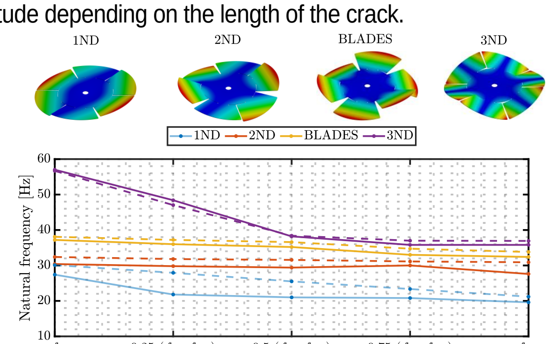


Fig. 9. Transition of the natural frequencies in the experimental and numerical models. Continuous line corresponds to experimental results and dashed line to numerical results.

It can be seen that the numerical and experimental trends of the frequencies for all modes are in very good agreement, despite ~~in~~ some ~~cases there are some~~ slight deviations~~,~~ possibly induced by several sources of modelling errors. Additionally, even though the crack was created based on the field of stresses induced by the 1ND mode of vibration, which has decreased its value a 28.47%, the most significant changes of frequency have been induced on the 3ND mode shape, as it has reduced its initial frequency by a 37.20%. By knowing in advance these frequency evolutions for the most sensitive modes as a function of the ~~length of the~~ blade's fatigue crack length~~for the most sensitive modes~~, a SHM system could be designed to monitor the~~m~~ crack length without the need of stopping and inspecting the runner. For example, based on the current experimental and numerical results, it seems that the 3ND mode might be a good candidate to be tracked in a real Kaplan turbine runner to assess the integrity of the structure due to its significant decrease at the initial stages of the crack formation and propagation in comparison with the rest of modes.

Based on the validated results, the authors proposal is to use this monitoring methodology during the start-ups of hydraulic turbines, as a wide range of runner frequencies might be excited during that process. Although the best measurement locations would be directly on the runner blades using on-board sensors, the complexity and cost of this type of measurement set-up makes it necessary to investigate the feasibility of using off-board sensors instead. This is the future work that will be continued at the laboratory test rig to advance in the present work objective.

4. Conclusions

During the apparition and development of an artificially machined crack on a disk-like structure resembling a Kaplan turbine runner, some measurable effects were identified on its dynamic response. By numerically simulating the evolution of the crack, significant changes in the modal properties of the disk were tracked, primarily resulting in a reduction of the lowest natural frequencies and the alteration of their mode shapes. Additionally, a validation of the numerical results was carried out comparing them with an experimental modal analysis performed with a hammer and accelerometers.

As long as the length of the crack increases, both numerical and experimental results show that the natural frequencies decrease due to the decrease of the stiffness of the disk. The associated mode shapes also change accordingly, in which the crack formation induces a torsional movement of the cracked blade independently of the mode shape. Concretely, the mode that has the highest natural frequency variation with the crack length is the 3ND with a 37.20% decrease of its initial value and thus evolving from 57 to 35.8 Hz when the crack reaches its critical length of 72.5 mm.

Up to now, these studies were conducted with the disk still and in air but, as future work, it is intended to repeat similar studies with the disk submerged in water both still and rotating. Additionally, this methodology could be tested in actual Kaplan turbine runners, in which more complex 3D crack surface growth methods will be needed to obtain numerically the crack path [10]. For that purpose, the monitoring of the natural frequencies could be done during the start-up of the turbine when the lowest modes of vibration of the runner might be sufficiently excited and measured with off-board accelerometers.

Acknowledgments

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Nomenclature

f : Natural frequency of a mode shape
k : Modal stiffness of a mode shape
m : Modal mass of a mode shape
k' : Critical stiffness
f_{crit} : Critical natural frequency of a mode shape
L_{crit} : Crack critical length
$f_{numerical}$: Natural frequency obtained through the numerical modal analysis
$f_{experimental}$: Natural frequency obtained through the experimental modal analysis

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Rafel Roig received his MS in Industrial Engineering from Universitat Politècnica de Catalunya (UPC) in 2017. From 2017 to 2020, he was working as design engineer. In 2020 he joined the Barcelona Fluids & Energy Lab (IFLUIDS) as PhD candidate. His research interests include Structural Dynamics, Fluid Structure Interaction, Finite Element Model and Experimental Testing.

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