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## Experimental investigation on the dynamic response of Pelton runners

To cite this article: Mònica Egusquiza *et al* 2019 *IOP Conf. Ser.: Earth Environ. Sci.* **240** 022062

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# Experimental investigation on the dynamic response of Pelton runners

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**Abstract.** An effective condition monitoring is essential to increase the availability of Pelton turbines and to avoid unexpected damage. The main challenge consists in determining the deformations and the stresses in the runner while in operation. To do so, it is essential to understand the dynamic response of the machine, and especially of the runner. Moreover, the monitoring locations have to be selected as to detect this response optimally. Having a deep knowledge of the natural modes of the runner, and how these can be excited while it is operating is mandatory to detect damage. The modal behavior of a Pelton runner can either be studied by means of numerical simulation or by experimental modal analysis (impact testing). However, none of these methods alone is able to describe with accuracy the dynamic behavior of the machine under real operating conditions. In this study, an experimental investigation of an existing Pelton turbine unit has been carried out. The modal response of the runner suspended, attached to the shaft and with the machine in operation has been studied. To do so, accelerometers, acoustic emission sensors and other types of sensors have been used. Signals were acquired during transients and with the machine operating at different loads. For a better understanding, a numerical model based on Finite Element Method (FEM) was created to represent the behavior of this type of turbines. At the end, the natural frequencies detected with the machine in operation were compared to those obtained numerically and by impact testing. Therefore, the effect of different factors on the runner response was analyzed. In addition, the best sensor and optimal location in describing the state of the machine were chosen. From the results obtained a possible improvement of the condition monitoring for these turbines is presented.

## 1. Introduction

Pelton turbines can be subjected to different types of damage during operation. Problems like silt erosion or cavitation erosion can lead to a considerable decrease in the efficiency of the turbine due to a modification of the inner surface of the buckets or of the injector nozzle [1][2][3]. When these are detected in a machine inspection, the affected areas must be reshaped to the original design. Other types of damage that can be found in Pelton turbines are caused by the fatigue of the material [4][5]. Since the buckets of the runner receive periodically the impact of the water jets, after some time of operation the high cycle alternating stresses lead to the appearance of small fissures. In case the magnitude of the forces is too large or the material has some imperfections, the effect of fatigue is magnified. If this remains undetected for a sufficient amount of time, the propagation of the cracks can



lead to failure (i.e. breakage of the buckets). When this happens, not only the reparation costs are high, but also those coming from the machine downtime.

This situation is aggravated nowadays, since power plant operators are interested in increasing the operating range of the turbines and in having more flexibility to start and stop the machine [6]. These new conditions inevitably lead to a higher risk of damage, because the machine operates longer in off-design conditions and the number of transients is increased. Since minimizing downtimes is also a priority for power plant operators, ensuring an extended lifetime of the turbine runners becomes a challenge.

In the current scenario, condition monitoring plays an essential role. Typically, this technique consists in placing accelerometers in different locations (e.g. bearings) in order to measure the vibration of the machine during operation. These signals are subsequently processed and analyzed to perform a general diagnostic of the state of the machine. A proper analysis can largely reduce the need to disassemble and inspect the turbine. However, while condition monitoring can help avoiding major damage in the machine, it is still a challenge to provide an accurate diagnostic of what is happening inside the turbine, due to its inaccessibility. To do so, the dynamic behavior of the Pelton turbine must be studied thoroughly and the locations of the sensors must be accurately determined. In this study, tests have been carried out in a real machine in order to determine which locations are better to detect abnormal operating conditions that can lead to damage. The experimental data is composed by impact testing in the machine, start-up vibration spectra and signatures. In addition, a model has been created numerically to help us understand the modal behavior of the turbine.

## 2. Machine characteristics and setup

The object of study is a prototype Pelton turbine located in Spain (see figure 1). The power plant houses two horizontal Pelton units with a rated power of 35MW each and a head of 756m. Each unit consists of a generator, a shaft and two Pelton runners, and two bearings support the whole rotating structure (see figure 4).

The runners have 22 buckets, a pitch diameter of 1900mm and are operated by a single jet. The nozzles are located under the runners in the horizontal direction.

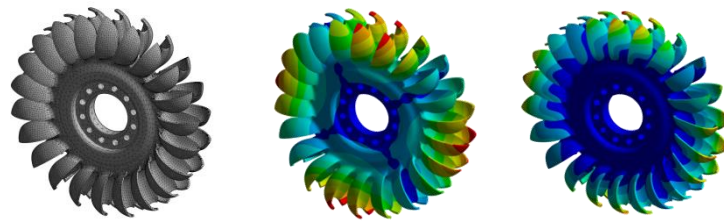


**Figure 1.** Pelton unit of the study.

## 3. Numerical model

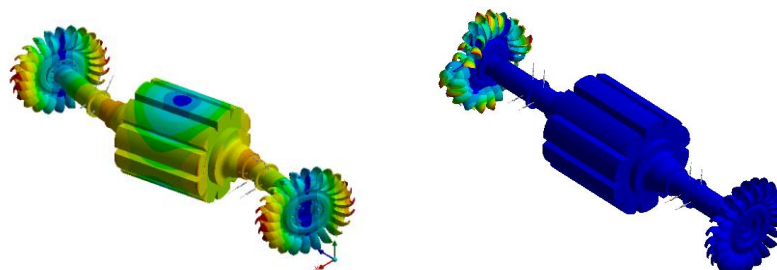
To understand the modal behaviour of the prototype Pelton turbine a numerical model was created. The geometry of the runner was obtained by scanning the surface of the prototype with a 3D scanner. The modal analysis was then performed with the Modal Analysis module of ANSYS Mechanical,

which uses Finite Element Methods (FEM). The meshed geometry is displayed in figure 2 left. Since an excessive number of mesh elements turns out to be computationally expensive and a low number may compromise the stability of the simulation results, a mesh sensitivity analysis was performed. The modal analysis helped identifying the natural frequencies and the mode shapes of the Pelton runner. In figure 2 middle it is the axial 2ND (two nodal diameters) that can be seen, in which the structure vibrates in the same direction of the shaft. In figure 2 right there is the 2ND tangential mode, in which the buckets bend in the tangential direction of the wheel. These modes are the most typical in Pelton runners and can be found in a range of frequencies and with different number of nodes. These and others are described in more detail in [7].



**Figure 2.** Left: runner mesh, middle: axial mode 2ND and right: tangential mode 2ND.

After this previous analysis of the modal behavior of the runner, the whole rotating structure of the machine was modeled, meaning that the two runners, the shaft and the generator were included. The stiffness of the bearings was simulated by attaching horizontal and vertical springs to the shaft. The results obtained from the simulation showed that the most important modes (which are the ones considered in this study) are located between 0Hz and 640Hz. In the lower frequency range one can find the modes of the rotor (between 30 and 84Hz) in which the whole structure vibrates with large amplitudes. Figure 3 left is a representation of a rotor mode at 75Hz. The generator, the shaft and both runners are involved in this vibration mode, what means that the modal mass is large. In a higher frequency range (110-530Hz), one can find the axial modes of the runner. It is worth noting that in higher frequencies the modes involve less modal mass, meaning that in the highest modes only the buckets vibrate. In figure 3 right there is a representation of a 3ND axial mode at 373Hz, which, unlike the rotor mode, doesn't transmit the vibration to the bearings. Finally, the so-called tangential or bending modes are located between 590 and 640Hz. These modes also involve only the vibration of the buckets.



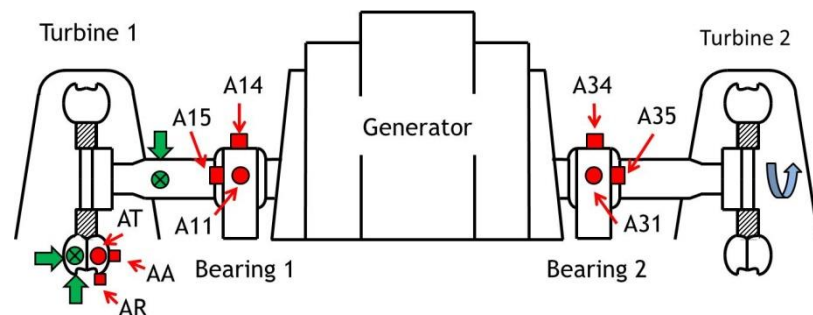
**Figure 3.** Left: modal shape at 75Hz and right: modal shape at 373Hz.

#### 4. Experimental tests and data

The experimental tests were used, on the one hand, to check the validity of the numerical model, and on the other hand, to see which frequencies are more excited when the turbine is in operation and which locations are the best to detect them.

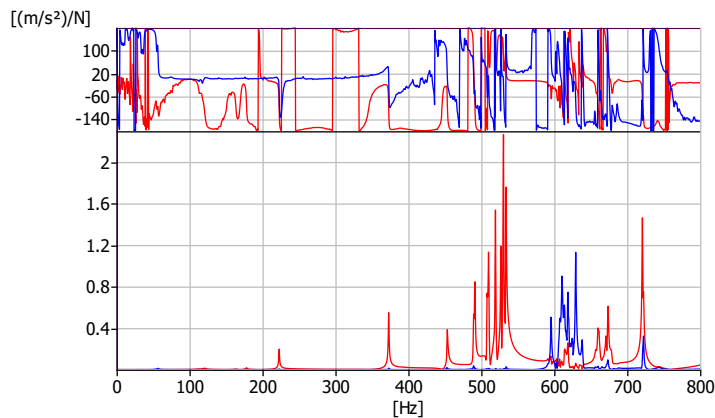
##### 4.1. Impact testing

Impact testing was carried out in the assembled machine in order to identify the different modes of the turbine. The procedure consists in performing impacts with an instrumented hammer (which records the force magnitude) on different locations of the machine and in recording the response of the structure (vibration amplitudes and phases) with accelerometers. The signals from each sensor are treated and combined to obtain the Frequency Response Function (FRF) and, eventually, to identify the frequencies and modal shapes of the machine. In these tests, the accelerometers were placed on the buckets of turbine 1 (AR for radial, AA for axial and AT for tangential) and on both bearings (A11/A31 for horizontal, A14/A34 for vertical and A15/A35 for axial) as represented in Figure 4. With this setup, the vibrations of the runner and the shaft and their transmission to the bearings were measured. The impacts (seen as green arrows in figure 4) were effectuated on different locations of the runner and the shaft to excite the rotor modes, the axial modes and the tangential modes. The recording modules and the software used in these tests were from PULSE Bruel&Kjaer.



**Figure 4.** Location of the accelerometers and the impacts on the machine.

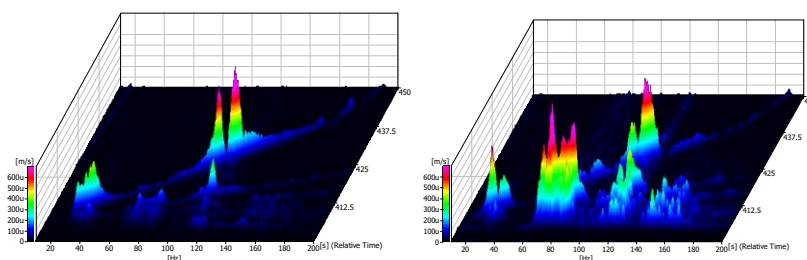
Figure 5 displays the response measured by accelerometers AA (red) and AT (blue) after doing axial impacts to the buckets. The peaks in acceleration/force represent the natural frequencies of the runner excited by the axial impacts. The axial modes can be clearly recognized between 110 and 540Hz and between 650 and 720Hz because of their high acceleration values and because they are only captured by the accelerometer in the axial direction. The tangential accelerometer instead, captures the tangential modes between 590 and 640Hz. Once the FRF's were obtained, the modal shapes were identified and checked with the numerical model, which proved to be a good representation of the prototype. For example, 1ND horizontal and 2ND axial modes were found in the tests at 113 and 223Hz, respectively, and were represented at 115,85 and 225Hz in the model.



**Figure 5.** In red the response of accelerometer AA and in blue the response of AT to axial impacts on the bucket. The top graph shows the accelerometer/hammer signal phase change and the bottom one shows the acceleration with respect to the force.

#### 4.2. Start-up spectra

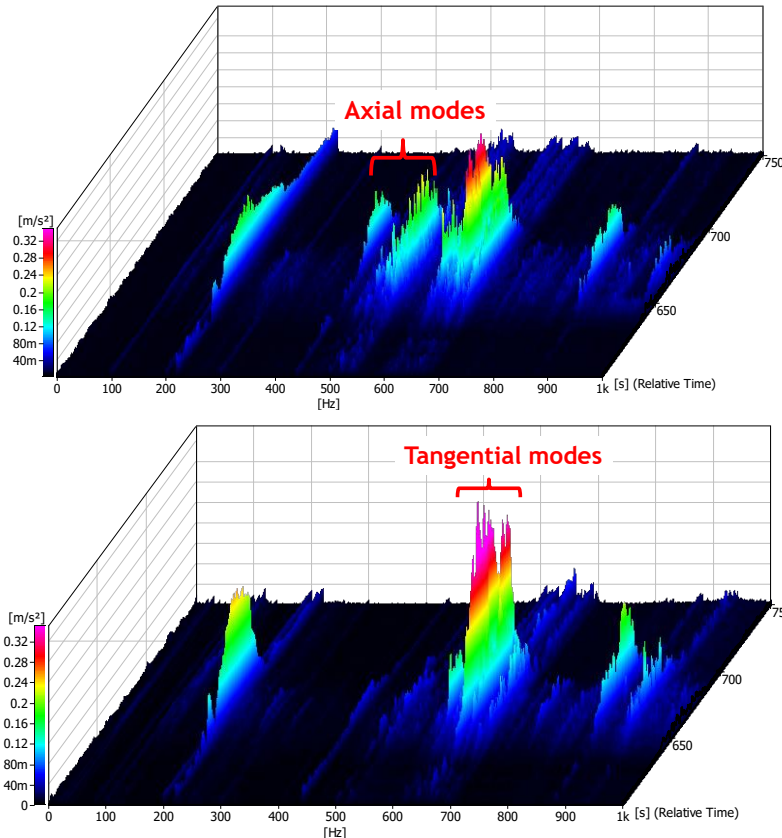
After doing the impact tests, the vibration spectra were recorded during the start-up of the machine. The accelerometers were placed on the same locations of the bearings as in the previous tests (A11, A14, A15, A31, A34 and A35). During the transient, several frequencies of the machine are excited and, if the respective modes are identified, important information can be extracted about the operating conditions of the machine. Figure 6 left shows the spectra waterfall during start-up from the accelerometer in the vertical direction. The peaks that can be discerned in the graph correspond to a rotor natural mode at 36Hz and the first vertical axial runner mode at 114,5Hz and 120,5Hz. When comparing this transient to the one obtained from the horizontal accelerometer, it can be seen that the deformation is higher in the latter. This can be attributed to the fact that the jet impacts the runner horizontally, thus exciting the modes in which the shaft is moving horizontally. Moreover, the stiffness of the casing of the bearing is lower in this direction than vertically and as a consequence the displacements are higher. Figure 6 right shows the spectra obtained from the axial direction. In this case the accelerometer can detect a larger number of frequencies. This is due to the deformation of the shaft in the lower frequencies. As a result, we can say that an accelerometer in the axial direction is the one that gives more information about the modes in the lower range of frequencies.



**Figure 6.** Spectra waterfall from the vertical A34 (left) and axial A35 direction (right).

In figure 7 the vibration from the horizontal and vertical directions can be compared. Tangential modes are better detected from the horizontal position due to the direction of the jet impact. However, it can be seen that the range of axial frequencies reaches higher values in the vertical direction. The excitation of the axial modes is due to a certain degree of deviation in the jet, what leads to the forces

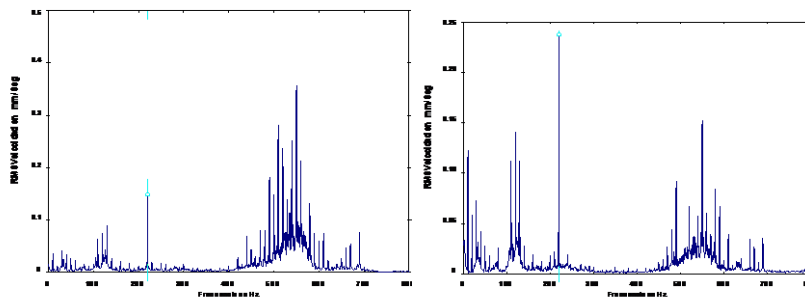
inside the bucket being unbalanced. In this case, the excitation of the horizontal jet in the axial direction will be better detected from the vertical position.



**Figure 7.** Spectra waterfall from the vertical A34 (top) and horizontal direction A31 (bottom).

#### 4.3. *Vibration signatures*

The vibration signatures of the machine have been recorded periodically during more than 10 years from accelerometers placed on both bearings. The proper analysis of these can provide valuable information about the state of the machine. In this study the signatures from the horizontal, vertical and axial directions have been compared. In figure 8 one can see the signatures from the vertical and horizontal directions. As expected, the detection of the frequencies excited is different depending on the direction of the sensors. Looking at the signature in the horizontal direction, one can see that the peak in the frequency of 220Hz is almost three times higher than the same in the vertical direction. This peak corresponds to the bucket passing frequency, which is the number of buckets (22 buckets) times the rotation frequency (10Hz). Provided that the jet impacts the runner in the horizontal direction, this vibration frequency is larger in this direction. As seen in the previous section, the range of the axial frequencies is higher in the vertical direction due to the position of the nozzle.



**Figure 8.** Signatures from the vertical (left) and horizontal (right) directions.

In all the signatures analyzed in this section, the range of frequencies related to the tangential modes reach low amplitudes. This doesn't mean that the tangential modes aren't excited, but that from the current monitoring locations these cannot be properly perceived.

## 5. Conclusions

An experimental investigation has been carried out in a real prototype of a Pelton turbine unit. Firstly, a numerical model was created in order to understand the dynamic behavior of the whole machine and to identify its natural frequencies and modal shapes. Next, impact tests were carried out in the Pelton unit in order to identify the modes of the real machine and to check the validity of the numerical model. After that, accelerometers were placed in the bearings in the three directions and the vibration spectra were recorded during the start-up of the machine. Lastly, the signatures obtained after years of monitoring were analyzed.

In this study the interaction between the numerical model and the experimental data has been essential to understand the modal behavior of a Pelton turbine. Frequencies and modes have been identified and classified. After that, the analysis of the spectra from different locations and directions has helped determining the best way to identify each type of mode. It has been proved that the location and direction of the jet determines which modes are more prone to be excited and thus the location of the sensors is tied to it. In the turbine studied the jet was located horizontally under the jet. As a result, shaft modes and the first axial modes of the runner were more excited in this direction than vertically. However, the shaft modes were best perceived from the axial direction, due to its bending deformation. When determining the best location to detect the axial modes of the runner, it has been seen that the sensor in the vertical direction is the most sensitive one. This is explained by the fact that an axial excitation coming from the jet tends to bend the runner and the shaft vertically. Finally, the tangential modes were analyzed in the signatures and it was proved that, while in the transient the horizontal accelerometer was able to detect them, none of the accelerometers during monitoring was able to detect them well.

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