Transient Analysis of Pelton Turbine Prototypes

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Abstract. One of the most critical operating conditions in hydraulic turbines is the start-up transient. This may be characterized by the presence of several resonances, which increase the vibration and stress levels on the runners. At present, its study is of interest due to the more frequent starts and stops of the turbines in the last years. The present study is focused on analysing the start-up transient of Pelton turbines. To do so, an experimental investigation was carried out in a horizontal shaft prototype unit. The procedure started by determining the natural frequencies and mode shapes of the turbine by means of impact testing. Several accelerometers were placed on the runner, on the shaft and on the bearings. With experimental modal analysis the main vibration modes of the rotor and the runner were found. Once the structural response of the turbine was determined, the start-up transient was studied. For the tests, different types of sensors were installed on the machine. Accelerometers, proximity probes and acoustic emission sensors were placed on the bearings and on the turbine casing. Moreover, an accelerometer and a strain gauge were installed on the shaft with an on-board system. Different situations are identified during the start-up. At the very beginning of the transient, the initial collision of the water jet on the still runner produces large vibration levels due to the excitation of many bucket modes. Once the turbine starts rotating, the vibration levels increase every time the jet excitation at the bucket passing frequency meets a natural frequency of the structure. The effect of the transient regarding the change in the vibration levels and runner deformations is presented in this paper. The sensitivity of the different sensors to measure the runner vibrations is analysed in order to optimize the detection of the transient effects. The best locations to install the sensors are also identified.

1. Introduction

In the last years, the fast growth of new renewable energies (NRE) has led hydropower plants to operate in a more flexible way [1]. Due to the random nature of renewable sources such as solar and wind, hydro turbines are required to undergo more starts and stops in order to make up for the derived power fluctuations and to ensure the power grid stability. Hydro turbines are thus subjected to more hydraulic transients, which can compromise the integrity of the machine components more rapidly.

One of the most common types of turbine is the Pelton turbine, which is used in sites with high heads and low discharges. Pelton turbines are subjected to large periodic loads coming from the interaction between the jet and the buckets [2,3]. The buckets must thus endure large dynamic stresses that eventually lead to cracks and damage due to the fatigue of the material [4,5]. In case the excitation coming from the jet enters into resonance with a natural frequency of the turbine, this situation is aggravated.

In this study, the effect of the start-up transient of a horizontal Pelton turbine has been evaluated. First, the modal behaviour of the machine has been studied to find the natural frequencies and modal shapes. To do so, a numerical model has been developed and validated with an Experimental Modal Analysis (EMA) carried out on the prototype. Next, the machine has been studied during the start-up transient. Several sensors including accelerometers and strain gauges have been installed and the results have been analysed.

2. Machine characteristics

The study has been performed on a horizontal prototype Pelton turbine. The characteristics of the machine have been listed in table 1. This consists of two runners, one shaft and the alternator. Two bearings support the structure. Each runner is operated by one horizontal jet.

Table 1. Main characteristics of the studied turbine.

Head	770,5 m	**Output**	35 MW
Speed	600 min^{-1}	**Nozzles**	1
$\boldsymbol{N_s}$	19	**Buckets**	22

3. Modal analysis

To understand the behavior of the machine in operation, it is mandatory to perform a modal analysis and to identify the natural frequencies and mode shapes of the turbine. To do so, a model has been developed numerically by means of Finite Element Method (FEM) and impact tests have been carried out on the real machine to validate the information.

3.1. Numerical model

The numerical model was made with the modal analysis module of ANSYS® software. The geometry of the runner was obtained from a 3D scanning of the real turbine and the rest of the components were developed with Computer Aided Design (CAD) programs. The mesh was selected after a sensitivity analysis, which proved that the model was accurate enough to provide stable frequency results.

The numerical simulation showed that the modes of the turbine can be divided into two groups: one for the rotor modes and another for the runner modes. The rotor modes appear in low frequencies (below 400Hz) and involve the deformation of all the components of the turbine. The runner modes appear in higher frequencies (above 400Hz) and only have deformation of the runner. The runner modes can also be classified depending on the direction of the bucket deformation (tangential, radial and axial) and on the number of nodal diameters (ND). As the number of ND increases, the vibration becomes more restricted to the buckets. The deformations and stresses in this case are more dangerous because the modal mass is small and the energy needed to excite the structure is much lower. All the information regarding the numerical model and results is explained in detail in [6].

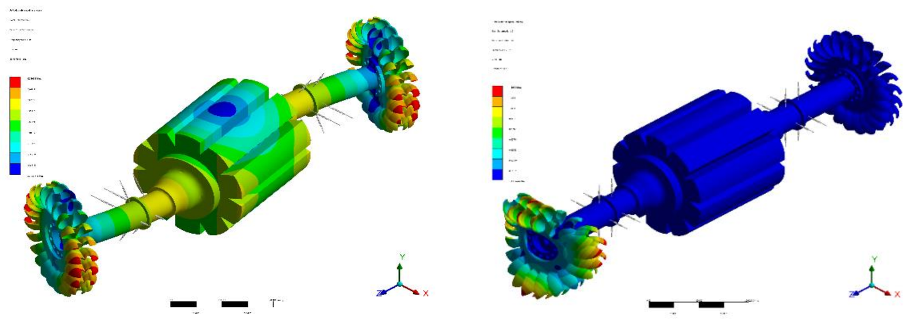

Figure 1. Left, typical bending mode of the turbine rotor and right, 2-ND runner mode in the axial direction.

3.2. Experimental tests

The impact tests were aimed at identifying the real frequencies and modes of the turbine. The casing of one runner was removed in order to reach the buckets. Several accelerometers were installed on different positions of the shaft and on the runner, and many series of impacts were performed with a hammer. The specifications of the sensors used and their positions for all the tests are detailed in [6]. The natural frequencies and mode shapes were obtained and suited well the numerical results. In table 2, the natural frequencies corresponding to the main types of rotor and runner modes are listed.

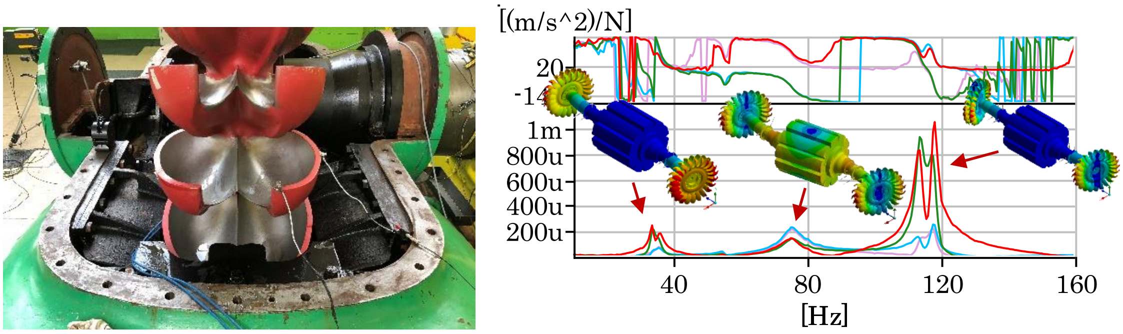

Figure 2. Left, accelerometers placed on the bucket tangentially and axially during the impact tests and right, rotor modes detected with the impacts.

Table 2. Main rotor and runner modes of the Pelton turbine studied.

Rotor modes	Freq. [Hz]	Runner modes	Freq. [Hz]
1st bending	30-35	Axial	223-534
2nd bending	75-84	Tangential	590-635
Torsional	54-57	Rim	640-676
3rd bending	115-120	Radial	722-1176

For the on-site tests while in operation, the vibration of the machine was recorded from different monitoring positions and with different sensors, as seen in figure 3 left. Accelerometers and proximity probes were installed on the bearings in vertical and horizontal direction. Acoustic emission sensors were also placed on one bearing and the turbine casing. In addition, strain gauges were installed on the shaft with an on-board system. Only the accelerometer signals will be discussed in this paper. The position of the accelerometers is shown in figure 3 right.

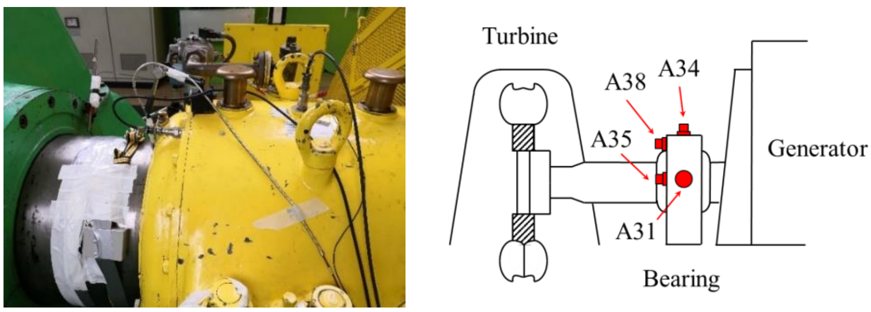

Figure 3. Left, sensors used for the on-site tests. Right, sketch of accelerometers' position

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4. Start-up transient

The start-up transient comprises the period between the opening of the nozzle and the turbine reaching nominal speed. The first water particles coming out of the nozzle collide sharply on the still buckets and, as the runner starts rotating and increases its velocity, the jet excitation enters into resonance with several natural frequencies. The components at most risk under these conditions are the runner buckets because of its direct interaction with the water jet. In case the main jet excitation or its harmonics enter into resonance with bucket-dominated modes, large vibrations on the buckets and, consequently, a more rapid deterioration are to be expected. Therefore, it is essential to study the structural response of the runner during the start-up transient with experimental data and to analyse the subsequent stress distribution with the help of numerical models. Eventually, the effect of increasing the number of starts and stops of the turbine can be assessed by means of fatigue analysis methods.

4.1. Evolution of vibration levels

In figure 4, the time signal recorded from start-up to maximum load has been represented in m/s^2. To set the machine in motion, the nozzle of the monitored runner was opened at 3,8% (segment 1). In the very beginning, the acceleration reaches its highest value. Then, the vibration decreases progressively until the runner is rotating at the nominal speed of 600 min^{-1} (2). After the alternator has been magnetically excited (3) and the nozzle of the second runner has been opened (4), the machine is connected to the electrical grid and put to operate at minimum load (5). This leads to an increase in the vibration. After some time, the load is steadily increased until reaching the maximum load (6,7). In this stage, the vibration of the machine is significantly higher.

On the right of figure 4, the start-up transient has been zoomed in. There is a clear initial period, in which the vibration transmitted to the accelerometer is maximum. This can be related to the first impingement of the water jet on the runner, which is similar to the sharp impact of a hammer and lasts for only for a few seconds. After that, the vibration of the machine is lowered and it starts to steadily decrease as the runner approaches its nominal speed.

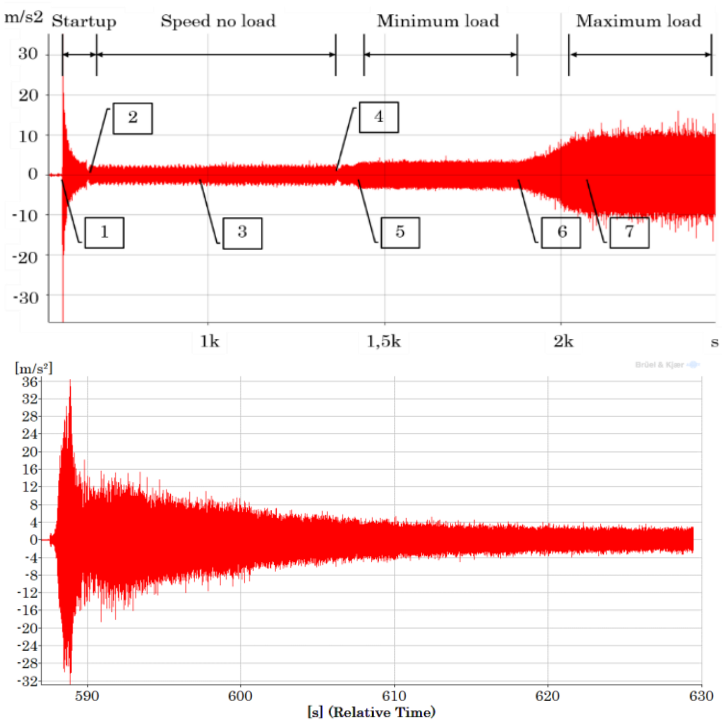
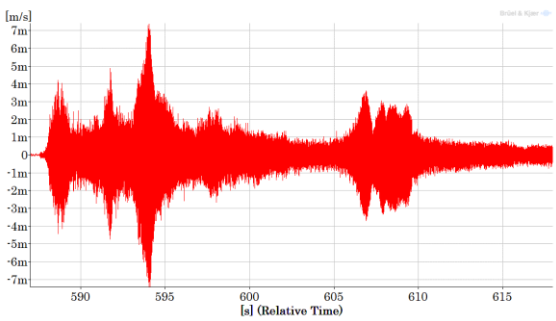


Figure 4. Top, time domain signal (in acceleration) measured from position A34 during a whole operating cycle. Bottom, zoom in the start-up transient.

In figure 5, the time-domain vibration signal recorded from the horizontal and vertical accelerometers during start-up is represented in velocity (mm/s). It is worth noting that the horizontal vibrations are larger than the vertical ones. This can be attributed to the direction in which the water jet impinges the runner and to the lower bearing stiffness in the horizontal direction. In the first seconds of the transient, the initial collision of the water particles on the bucket gives rise to an increase in the velocity. The maximum value attained is of 8 mm/s peak-to-peak. After that, the turbine starts rotating. There are several noticeable increases in the vibration during the speed-up, which are caused by the resonance of the bucket passing frequency with the natural frequencies of the rotating structure. The largest vibration reaches a value of over 14 mm/s peak-to peak.



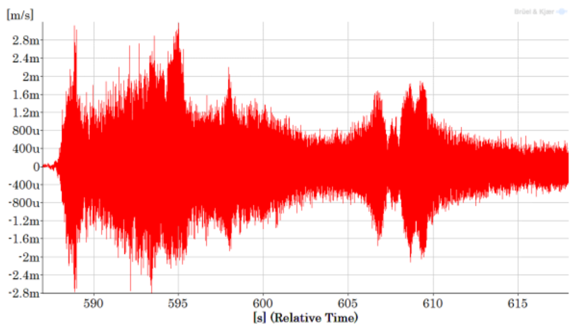


Figure 5. . Evolution of the vibration (mm/s) in horizontal A31 and vertical A34 direction measured in the turbine bearing during start-up.

In figure 6, the rms velocity levels during the transient have been plotted. According to the ISO20816-5:2018 Standard for the measurement and evaluation of machine vibration [7], the velocity measured from the turbine bearing must not exceed 3,1 mm/s to avoid big damage (figure 7). The trip and alarm rms limits have been marked in the chart with dashed lines. Even though these values are only applied for normal operation, they have been used as a reference to assess the vibration during the start-up. In the horizontal direction, the velocity rms is surpassed during the start-up in one occasion (4.5 mm/s rms).

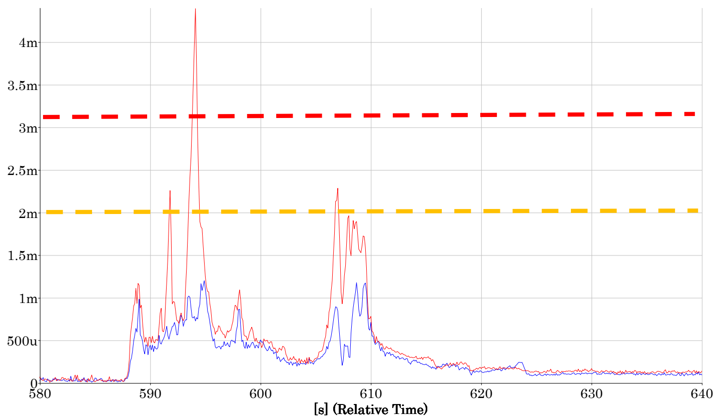
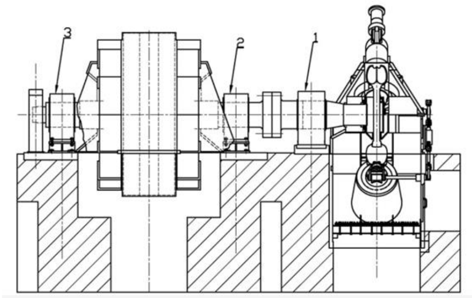


Figure 6. Variation of the rms level during start-up transient in horizontal A31 (red line) and vertical A34 direction (blue line).



Machine type:	Shaft vibration Sp-p, µm			Bearing vibration Vrms, mm/s		
Pelton Horizontal	T1	GE-DE	GE-NDE	T1	GE-DE	GE-NDE
Action limit 1 (A-B/C)	120	140	100	2,0		1,1
Action limit 2 (C/D)	190	210	160	3,1		1,8

Normal operating range for Pelton horizontal: from 10 % to 100 % of rated power

Figure 7. Vibration Standards for Pelton units [7].

In figure 8, the waterfall plot of the transient has been represented in acceleration (m/s^2) and in velocity (mm/s). The two aforementioned stages of the start-up can be clearly discerned. In the first one, the hit of the first water particles with the still buckets mainly excites frequencies above 400 Hz, which correspond to the natural frequencies of the runner. The second stage takes place when the runner starts rotating and lasts until it reaches its nominal speed. The main excitation affecting the runner is the jet-bucket interaction at the bucket passing frequency f_b, and can be expressed as shown in equation 1. z_b stands for the number of buckets and f_f for the rotational speed. In this turbine, the main excitation at nominal speed is found at 220 Hz.

$$f_b = z_b f_f \tag{1}$$

In figure 9, the waterfall is plotted in mm/s. Representing the signal in velocity enhances low frequencies over high frequencies. It is thus seen that mainly the resonances of f_b and its harmonics with rotor bending modes are important. The highest oscillation velocity takes place when f_b enters into resonance with a horizontal bending mode at 33 Hz. In terms of stress concentration, the excitation of this mode does not compromise the integrity of the structure since the curvature of the shaft is small. However, the excitation of a third horizontal bending mode near 120 Hz by the f_b may bear more problems in the structure because the oscillation is more localised in the runner. For this modal shape, the most affected area is the edge of the connection of the runner to the shaft.

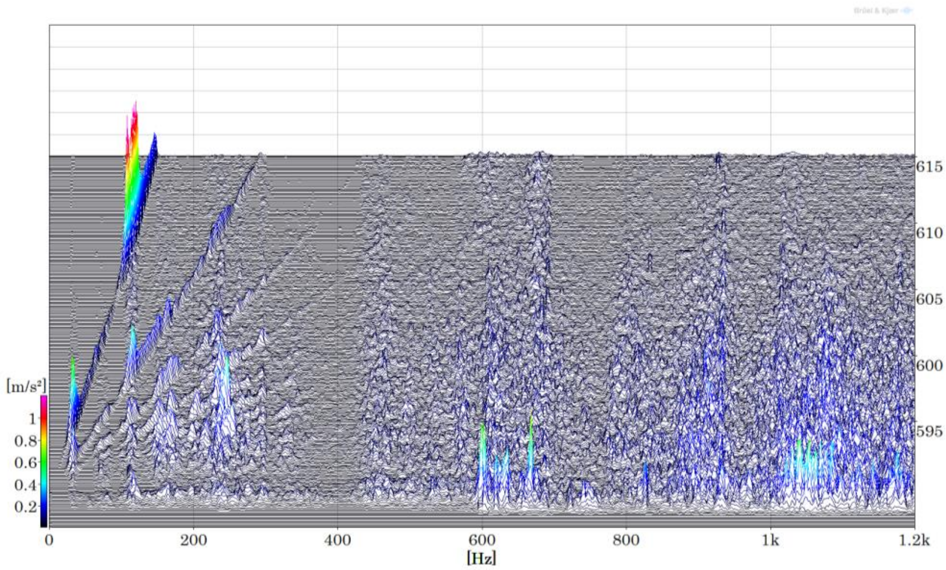


Figure 8. Spectra waterfall from position A31 in acceleration.

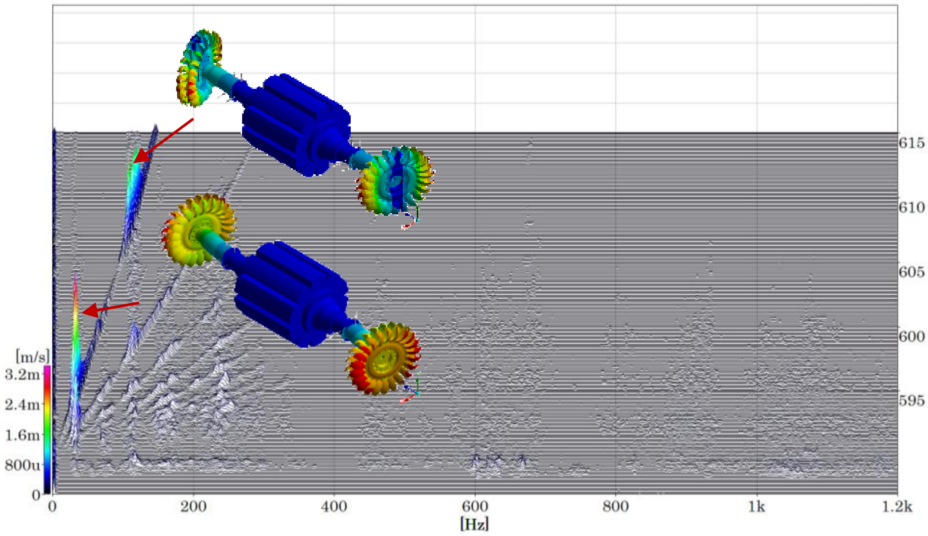


Figure 9. Spectra waterfall from position A31 in velocity (mm/s).

The waterfall plot during the first seconds of the start-up has been represented between 500 and 700 Hz in figure 10. The peaks that stand out correspond to the natural frequencies of the runner. With the information obtained from the impact tests, the frequency ranges corresponding to every type of mode have been limited. As it can be seen, the frequencies excited at the start of the operating cycle match the ones of the still machine.

The identified runner modes help understanding the characteristics of the jet force. The tangential modes are always excited because the jet is directed perpendicularly to the buckets. In addition, the rim modes are also predominant because the inner surface and rim of the buckets deflect the water jet. A single peak outstands in the range of the rim modes. This corresponds to the pure natural frequency of the bucket impinged by the jet at the beginning. Consequently, a great amount of the energy given by the jet is used to oscillate one single bucket and, as explained afore, the stress concentration is very high in such case due to the large amount of energy used to oscillate a small amount of modal mass. Nevertheless, when analysing the overall vibration levels, the values don't indicate a dangerous situation for the machine. For these bucket modes, though, the common failure areas such as the cut-out and the root are especially affected [8-10]. It is thus essential to study the stress condition and the transmission of the bucket vibrations to the monitoring locations taking into account the possibility of an increase in the number of starts and stops.

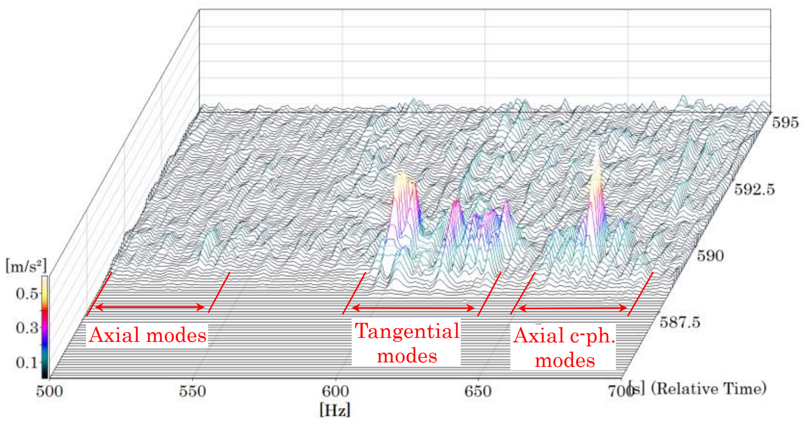


Figure 10. Zoom of initial start-up moment around bucket frequencies.

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5. Conclusions

In this study, the start-up transient of a horizontal Pelton turbine has been investigated in order to evaluate the effect of an increase in the number of starts and stops in the machine.

First, the modal behavior of the whole machine was analysed by means of a FEM numerical model and experimental data. Second, the vibrations of the machine were recorded by different sensors and from different locations during start-up.

The analysis of the start-up transient revealed that there are two different phases. In the first one, the first contact of the water jet with the runner buckets gives rise to a large oscillation. The energy is mainly used to excite the modes of the bucket hit by the jet, especially the tangential modes and the rim modes. Even though this situation may aggravate the stress concentration of the bucket, the overall rms vibration levels don't indicate a vibration problem in the turbine.

After the initial impact, the runner starts rotating and several resonances of the bucket passing frequency with natural frequencies of the rotor appear. The highest rms vibration levels take place when the first bending mode of the rotor is excited. Here the rms velocity overpasses the maximum value recommended by standards. The excitation of a third bending mode at 120 Hz may bear more consequences on the integrity of the machine due to its concentration of stresses on the connection between the shaft and the runner.

With the information obtained in this study, it would be convenient to perform a study on the effect of increasing the number of starts and stops in the turbine. Apart from the recommended rms values, a stress and fatigue analysis is recommended.

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