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Failure investigation of a Pelton turbine runner

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Abstract

In this paper, an uncommon failure of a Pelton turbine has been analyzed. After the monitoring system detected a sudden increase in the vibration levels, the turbine was inspected. The inspection showed that a fragment of one bucket broke off during operation. Moreover there were several buckets with cracks, always located in the same side of the buckets. An analysis of the detached fragment revealed a fatigue problem.

After the damage was found, the vibration signatures measured by the monitoring system before damage, with damage and after repair, were analyzed. Before damage occurred, an excessive axial vibration and the excitation of several natural frequencies of the turbine were detected in the measured vibration.

In order to identify the origin of the problem the first task was to analyze the dynamic response of the turbine. A numerical model of the runner using the finite element method (FEM) was done. Experimental research using modal analysis techniques (EMA) was also carried out in the turbine runner. The results of the numerical model were compared with the experimental results obtained. With the validated numerical model natural frequencies and mode-shapes were determined and studied.

The next step was to determine experimentally the influence of the mounting conditions on the runner dynamics and the transmissibility of the runner vibrations to the machine bearings where the monitoring sensors are located.

From the results of this study it was concluded that the natural frequencies excited during machine operation had axial mode shapes indicating that axial forces were applied to the runner. In a Pelton turbine, this can only be produced by a misaligned jet.

To determine the influence of a misaligned jet on the bucket stresses, the dynamic behavior of the runner was performed. The dynamic force of the water jet was applied to the runner bucket. The results showed that with a misaligned jet the dynamic stress distribution increases in one side of the bucket with a maximum stress located where the cracks appeared.

1. Introduction

Pelton turbines are widely used in hydropower plants with high heads. They belong to the group of impulse turbines because only the kinetic energy of the fluid is changed across the runner. Unlike reaction turbines, such as Francis or Kaplan, Pelton turbines are installed in open casings and are not subjected to a pressure variation. This is translated into low installation costs and suitability for inspection. In order to convert all the pressure of the water into kinetic energy, a nozzle known as injector is installed at the end of the penstock. The high speed water jet generated is used to propel the runner [1].

The runner of a Pelton turbine basically consists of a disk with a number of buckets attached along its periphery (Figure 1). The high speed water jet impinges the buckets perpendicularly so that maximum torque is applied to the rotor. At the inlet of the bucket, the jet is divided into two equal streams by the splitter, which is a sharp edge dividing the inner surface of the bucket into two cavities. This helps balancing the axial loads applied to the runner. At the outlet, both streams exit the bucket with diminished velocity and a deviation of almost 180 degrees. In design operating conditions, the water jet must be directed in the tangential direction of the runner and aligned with its middle plane.

Typical damage types in PT are caused by silt erosion, cavitation and fatigue [. Erosion usually affects the inner surface of the buckets and the injector, due to the high velocities of the water jet, and is expected to appear in areas where the water has suspended particles. The amount of material removed from the affected surfaces mostly depends on the size, shape and composition of the grain [2]. Cavitation may occur because of an improper velocity distribution or to a wrong design of the bucket profile. However, the worst cases of damage come from fatigue problems. Fatigue cracks progressively weaken some locations of the runner and may end up breaking off the structure. Reparation costs can be extremely expensive.

Pelton turbines commonly suffer from fatigue problems because of the pulsating forces to which the runner is subjected. Every time a bucket receives the impingement of the water jet, a high concentration of stresses appears at the root of the bucket due to its cantilever structure. In the long run the periodic action of the jet may result in the appearance of fatigue cracks [3]. Typically the most affected area is the bottom part of the splitter, which has to be designed as to reduce the stress values. In addition to that, a factor that highly influences on the appearance of cracks is the dynamic response of the Pelton runner. If the frequency of the excitation is the same as the frequency of a natural bending mode of the runner, the displacements derived from the action of the force will increase, and, consequently, the bending stresses at the root of the buckets may be critical. In the end, the so called bending modal shapes must be carefully studied and considered

in the design process of the runner, as they are likely to be excited by the tangential force coming from the jet.

Structural damage caused by dynamic stresses has been largely studied in Pelton turbines [4]. An accurate design and a proper selection of the material, as well as quality control during manufacturing, are mandatory in order to minimize the effect of these stresses. The application of numerical methods also allows refined hydraulic designs in which the shape of the bucket is optimized and the dynamic forces calculated with good accuracy [5] [6] [7] [8].

Several references can be found reporting the fractures in the root area. In the last years, papers still analyze the causes of failure. Chavez et al. [9] found numerically the location of the most critical stresses, taking into account the pressure induced by the jet impact and the centrifugal forces, and checked it with an experimental case. As reported also by Ferreño et al. [10], material properties also highly influence on the appearance of cracks. Other authors, like Schmied et al. [11], studied the detuning methods that prevent the structure from reaching resonance.

2. Damage description

The case reported here corresponds to a Pelton turbine with a head of 770m and maximum power of 34MW. It is a horizontal shaft machine composed by a runner of 22 buckets and one injector. The diameter is of 1930mm.

Periodically, vibration monitoring is carried out in order to supervise the condition of the turbine. This procedure consists in measuring the vibrations on some the accessible points of the machine allowing the detection of abnormal vibrational behavior and, in some cases, incipient damage. For this case, data was measured in the bearings in the axial and radial directions. In one of the measures, the vibration monitoring system detected an increase in the overall vibration levels (Figure 3). The RMS (Root Mean Square) velocity values surpassed the alarm level so the machine was stopped and inspected. In Figure 4 the vibration spectra measured by the monitoring system is shown.

The inspection of the Pelton runner allowed noticing that several buckets had cracks in the same location of the cut-out (Figure 5) and that one bucket was broken. It is important to remark that all damage appeared in the same side of the runner. Figure 6a shows the damaged bucket after the failure. In Figure 6b the broken piece can be seen. After an analysis of the broken surfaces of both the bucket and the detached fragment, beach marks could be discerned between the cut-out and the rim. This showed that the damage took place because of a fatigue problem.

As indicated above fatigue problems may appear in Pelton runners because they are subjected to cyclic loads, which lead to a concentration of stresses that end up weakening some locations of the component. Until now, all the studies reporting Pelton turbine failures evidenced that the main stresses affecting the structure appear in the bucket's root. In this case damage appeared in one side of the upper part of the bucket. Lacking references on similar cases, an accurate study was carried out in order to determine the origin of the damage.

3. Analysis of the machine vibration

As mentioned before, the state of a machine can be determined by the analysis of its vibrational behavior. The analysis of the spectra shows at which frequencies the largest vibrations do appear and allows making a diagnosis of the actual condition of the machine. To do so, it is essential to understand the origin of the most common vibrations in Pelton turbines.

3.1. Origin of vibration

Vibrations generated by a Pelton turbine can be of mechanical origin (like unbalance) and of hydraulic origin. Pelton runners are mainly affected by centrifugal forces and by the impacts of the water jets. Centrifugal forces only depend on the distribution of the mass and on the rotational speed. The excitation force coming from the water jet is the most important and varies depending on many factors, such as the jet velocity, the geometry of the buckets, the rotational speed or the nozzle characteristics.

The frequency at which the jet impinges the structure depends on the rotating speed of the runner and on the number of buckets and can be defined by (1).

$$f_b = n \times z_b \times f_f \quad (1)$$

where f_b is the frequency of the vibration generated on the runner by the impact of the jet (bucket passing frequency), z_b is the number of buckets and f_f is the shaft rotating frequency. n stands for the harmonics of the exciting force, being $n=1,2,3\dots$. From the perspective of a reference system that rotates at the same speed of the runner, the force of the jet is applied to each one of the buckets with a phase shift. The force is basically directed perpendicular to the bucket (tangential to the runner), thus the bending modes are very prone to be excited. The resulting force in the axial direction is close to zero due to the symmetry of the bucket, being the streams on both sides of the splitter balanced. The force in the radial direction can be also neglected.

3.2. Monitoring data analysis

The analysis of the vibration spectrum before the failure gives some information about the state of the machine. In the bearings radial direction (Figure 7a) the vibration peaks at synchronous frequencies are generated by the runner unbalance and by the action of the jet. On the one hand, the rotational speed of the runner 600rpm can be seen at 10Hz, which can only be caused by an unbalance in the centrifugal forces (which depend on the mass of the structure). On the other hand, every time the runner completes a revolution it has been impinged by the water jet 22 times, which is seen as a peak at 220Hz (bucket passing frequency). In addition to that, the broad band frequency ranges 100-140Hz and 450-600Hz denote the excitation of natural frequencies. With much lower amplitude, a band of 600-700Hz is also excited.

In the axial bearings, some abnormal behavior could also be suspected because of the high levels of the vibration signature (Figure 7b). The vibration generated by the bucket passing frequency is higher than in the radial direction what is not normal considering that in a Pelton turbine no net axial forces should be affecting the structure. It can be claimed that some unwanted forces were exciting the runner before failure.

After the bucket was broken, the vibration signature of the machine manifested some significant changes (Figure 8a). The frequency peak related to unbalance reached high values, what evidenced the loss of part of the mass. Apart from that, the excitation of natural frequencies can also be observed in the spectra. After repair (Figure 8b) all the vibration levels decreased considerably.

The indications/symptoms suggested that an axial force was exciting the structure. The first assumption was that the runner and the nozzle were not aligned, what would create an unbalance

in the axial forces applied on the bucket. This would be prone to excite the axial modes of the runner.

4. Modal analysis of the Pelton runner

To determine the effect of a misaligned jet on the runner (and on the distribution of stresses on the buckets) a detailed analysis of the dynamic behavior was carried out. First the dynamic response of the runner was studied. Second a distribution of forces resembling the water jet impact was applied to the inner surface of the buckets in order to obtain the deformations and the stresses.

4.1. Numerical simulation

A numerical model was built-up and calibrated with experimental data. The geometry of the runner was provided by the company. Such geometry was obtained by scanning the surface of the real runner with a portable measurement arm Romer Absolute Arm Series 7525SI, whose volumetric precision was of $\pm 0.05\text{mm}$ and the scanning system precision of $\pm 0.008\text{mm}$, so the dimensions were quite accurate. The upcoming calculations were carried out by means of Modal Analysis from ANSYS Mechanical software. This program uses Finite Element Method (FEM) to simulate the modal behavior of any structure, whose mechanical properties have been previously defined. The meshing procedure of the geometry is an important step to find a balance between accurate results and low computational resources. For that reason, a mesh sensitivity analysis was carried out, and after that, the number of elements was set as 530000. The density of elements was increased in the critical areas such as the cut-out and the root of the buckets. The material chosen for this simulation was structural steel, with a density of 7850 kg/m^3 . A view of the mesh is displayed in Figure 9a.

At first glance, the modal shapes of a Pelton runner look quite complex. However, it is good to understand that its behavior is similar to a disk. The modal shapes of a disk-like structures can be classified according to the number of nodal diameters (ND) and nodal circles (NC). The naming makes reference to the inflexion points of the deformation, which can be distributed diametrically or circumferentially. These areas can also be discerned on the Pelton runner, both in the disk and the bucket areas (Figure 9b). The deformation of the buckets combines mostly axial and tangential deformation, and phase and counter-phase between both sides of the bucket. As frequencies increase, so does the complexity of the modes. However, for the range of frequencies we intend to study, the modes can be classified by axial modes, tangential modes and torsional modes. The deformation of the buckets in these modes is showed in Figure 10. In Figure 11 the first mode shapes found have been represented.

4.2. Experimental tests

To check the validity of the numerical model, an experimental modal analysis (EMA) was carried out for the real runner. The runner was hanged with ropes (free body) so that the results were not affected by the boundary conditions and the numerical model could be validated. Moreover, the hanged runner was completely accessible and multiple points could be measured for an accurate identification of the mode-shapes. This procedure consisted in exciting the structure with an instrumented hammer while measuring the response (vibration) with accelerometers. All sensors were connected to a Bruel&Kjaer acquisition system which recorded the signals in the time domain. Fast Fourier Transformation (FFT) was then used to convert the signals to the frequency domain. The combination of the hammer signal (input) with the accelerometers signals (output) results in the Frequency Response Function (FRF), which defines the ratio between the vibration

levels and the force value. Thanks to the roving method, the natural frequencies and mode shapes could be determined accurately.

Accelerometers were attached to the disk and to the buckets as to measure the vibrations in the axial, radial and tangential directions. Figure 12 shows the distribution of the sensors in one of the tests. KISTLER K-Shear accelerometers with a sensitivity of 100mV/g and acceleration range ± 50 g were used. The recording system was a LAN-XI Data Acquisition Hardware from B&K. In the post-processing, Pulse Reflex software from B&K was used to analyze and to obtain the Operating Deflection Shape (ODS) for each one of natural frequencies.

The experimental frequencies and mode-shapes were compared to the ones obtained in the numerical model. The comparison between the values of the first five natural frequencies is shown in Table 1. The relative difference does not exceed 3%, what means there is good correspondence between the theoretical and the real model.

In Figure 13 and Figure 14 the FRF's after tangential and axial impacts on the buckets are displayed. The excited frequencies depend on the direction of the impacts and on the position of the accelerometers. For the impacts in the tangential direction, the bending modes can be easily discerned in the range of 520-580Hz (Figure 13). However, when impinging the buckets in the axial direction, the frequency peaks correspond to the axial modes of the runner as well, as well as torsional modes. It is interesting to see that the first modes (2ND and 0ND) are less excited than higher modes because they involve the deformation of the disk (Figure 14).

Mode	Theoretical runner hanged [Hz]	Experimental runner hanged [Hz]	Difference [%]	Experimental runner attached (Hz)
2-ND	190,2	195,5	2,3	230
3-ND	346	356	2,8	373,5
4-ND	413,3	424	2,5	453
5-ND	438,8	449,5	2,4	480

Table 1. Comparison between numerical and experimental natural frequencies

4.3. Effect of the shaft and transmissibility

The tests carried out with the hanged runner were used to determine the different modes of a Pelton runner and to validate the numerical model. However, the real runner is attached to a shaft mounted on bearings. In order to determine the influence of the shaft on the natural frequencies of the Pelton turbine, EMA was also carried out with the runner mounted in the machine (real operating conditions). The access to the runner was possible while mounted, so accelerometers were placed on the runner, on the shaft and on the bearings. Impacts were also made on these components. After the measurements, the natural frequencies of the shaft and the transmissibility of the vibrations from the runner to the bearing (monitoring position) could be studied.

Figure 15 displays the structural response (FRF) of the runner and the bearing to axial impacts on the buckets. The frequencies are higher than the ones found for the suspended runner, which was expected because the stiffness of the structure increases when the runner is attached to the rotor. In this Figure it can be observed that the axially excited modes, are shifted from 420-490Hz to 490-540Hz. It can be seen that the runner vibrations are transmitted to the bearing.

In Figure 16, the FRF of the response of the runner and shaft to vertical impacts on the shaft can be observed. These peaks correspond to lateral frequencies of the rotor.

Looking at the spectra of the vibration from the monitoring locations (Figures 7 and 8) it can be deduced that the range of frequencies excited in the machine corresponds to the axial modes of the runner (450-600Hz), what indicates the existence of an axial force. Tangential modes (600-700Hz) were not as excited as the axial modes. The excitation of the lateral bending modes of the shaft (120Hz) also indicates the existence of such force. To adapt the numerical model to this case, the runner was attached to the shaft.

5. Dynamic analysis

In order to determine the cause of the failure the dynamic analysis was carried out by means of ANSYS Harmonic Response. This program allows obtaining deformation and stress values when the structure is subjected to cyclic loads. The output can be represented as a function of the frequency of excitation, thus obtaining peak values on resonance points. The action of the jet was simulated by applying a pressure distribution on the inner surface of the bucket in the validated numerical model. Several publications indicate the distribution of pressure in the buckets comparing experimental and numerical data [12, 13]. Results are presented in a non-dimensional form so that they can be extrapolated to different Pelton designs. In this case the pressure distribution on the bucket surface was taken from [8] and adapted to this case.

To do so, the pressure values were obtained by means of a pressure coefficient (C_p) that related the relative pressure on the surface with a reference pressure, which is described by (3):

$$C_p = \frac{p}{p_{ref}} = \frac{p_{rel}}{\frac{1}{2}\rho U_{ref}^2} \quad (2)$$

p_{rel} is the relative pressure, ρ is the density of the water and U_{ref} is the runner peripheral velocity, which was of 60 m/s. The worst loading case (that is when the jet is perpendicular to the bucket) was considered in this simulation. The damping ratio was obtained from the experimental data by means of the Half Power Bandwidth method and set to 0.1%.

Two situations have been contemplated one with a centered jet and another with a deviated jet. The exact degree of deviation was estimated from the site inspection. In Figure 17 the deformation of the runner with a tangential force generated by a centered jet has been represented. For the centered jet the highest concentration of stresses occurs in the root of the bucket in the splitter, which was expected (Figure 18a). However, with a misaligned, though applying the same pressure distribution but deviated in the bucket, the stresses in the cutout alarmingly increase.

Another important fact in this case is that the excitation frequency of 220Hz is very near the first natural frequency of the axial modes (230Hz) corresponding to a 2ND mode-shape. This leads to larger deformations in the axial direction. Moreover, the cutout area presents a sharp edge that makes it more vulnerable to the cyclic action of the stresses. The same behavior is present at other axial natural frequencies.

6. Conclusions

In this paper, an uncommon failure of a Pelton turbine has been analyzed in order to find out which were the causes that provoked it.

Pelton turbines are prone to have fatigue problems because of its operating mode; each bucket is subjected to a large dynamic force when passing in front of the high speed jet. This dynamic loading produces large bending stresses at the root of the buckets. As a consequence, the typical damage in Pelton turbines is due to fatigue cracks at the root of the buckets. In the present case fatigue cracks appeared at one side of the cutout of the buckets. In one of them the crack developed and a fragment blew off from the bucket while the turbine was in operation.

This damage generated an increase in the vibrations of the machine that was detected by the monitoring system. The analysis of the monitoring signals indicated large vibrations in the axial direction at the bucket passing frequency, an increase in unbalance and the excitation of natural frequencies. Taking into account that in these type of turbines axial vibrations should be very low (a centered jet splits into two equal parts compensating the axial forces) a jet deviation was thought to be the culprit. Another fact is that the cracks appeared all in the same side of the bucket. The observation of the broken part could prove that the damage occurred due to fatigue.

In order to identify the origin of the problem, the analysis of the dynamic response of the runner while in operation was carried out. Firstly, a numerical model of the runner structure was built up by means of Finite Element Method. The mode-shapes and frequencies were obtained and compared with experimental results. With the validated numerical model, the response of the runner was examined in detail. The influence of the shaft on the runner response was also studied. Three main groups of vibration modes were found: mode shapes with axial displacement; a second group with bucket displacement in the tangential direction and a third group with torsional deformation.

Finally, the dynamic behavior of the runner was calculated by applying the force produced by the jet on the runner. To simulate the force of the jet a pressure distribution was applied on the buckets. The value of the pressure distribution was extracted from available data in non-dimensional form. Applying this excitation on the numerical model (harmonic analysis) the deformations and stresses in the bucket could be determined.

Two cases were computed one with a centered jet and another with a misaligned jet. The results indicate that the centered jet generate a symmetric stress distribution with a maximum at the root of the splitter. A misaligned jet shows an asymmetric stress distribution with maximum values at the root and especially at the tip of the bucket where cracks appeared. Another factor that increased the axial deformation is that the bucket passing frequency was almost coincident with the first axial frequency of the runner.

Therefore, the failure was produced because the jet was misaligned with respect to the runner middle plane and the proximity between the excitation frequency and one of the natural frequencies of the runner.

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Figure Captions

Figure 1. Sketch of a Pelton runner and injector

Figure 2. Pelton turbine

Figure 3. Trend plot of the overall vibration values measured in the turbine bearing

Figure 4. Spectra acquired by the monitoring system in the turbine bearing

Figure 5. View of buckets with a crack

Figure 6. (a) View of the broken bucket (b) View of the detached fragment with the beach marks

Figure 7. Vibration signature measured in the machine bearing before failure (a) in radial direction and (b) in axial direction

Figure 8. Vibration signature measured in the machine bearing (a) with damage and (b) after repair

Figure 9. Views of (a) runner mesh and (b) runner displacement in modal shape 2-ND

Figure 10. Deformation in (a) tangential or bending mode, (b) axial mode and (c) torsional mode

Figure 11. Mode-shape of the first axial natural frequencies a) 2-ND; b) 3-ND; c) 4-ND

Figure 12. Accelerometers placed on the Pelton runner

Figure 13. FRF of the response of the runner suspended (in tangential direction) to tangential impacts

Figure 14. FRF of the response of the runner suspended (in axial direction) to axial impacts

Figure 15. FRF of the response of the runner mounted (in axial direction) to axial impacts

Figure 16. FRF of the axial response in the runner and in the bearing to vertical impacts on the shaft

Figure 17. Dynamic response of the runner at 220 Hz (stress distribution) with a centered jet

Figure 18. Stress distribution in a bucket a) with a centered and b) with a misaligned jet

Figure 1

Figure 2

Figure 3

Figure 4

Figure 5

Figure 6

Figure 7

Figure 8

Figure 9

Figure 10

Figure 11

Figure 12

Figure 13

Figure 14

Figure 15

Figure 16

Figure 17

Figure 18

Highlights

- Vibrations in a Pelton turbine before, with failure and after repair were analyzed.
- High axial vibrations and the excitation of wheel natural frequencies were detected
- The wheel structural dynamics was studied using a FEM model validated with experiment
- Deformations and stresses were calculated using a misaligned jet on the wheel.
- Maximum stresses occur where cracks were found. Jet misalignment produced the damage