Operation of The Bois plant hydraulic turbine in bypass operating mode: analysis of fatigue behaviour

Philippe Bryla*, Marc Pillou*, Mohamed Bennebach

Abstract

During load tripping of the hydraulic turbine at the Bois power plant, the turbined flow is directly transferred and discharged into an external canal without releasing an alert flow. The operation of the turbine in bypass mode, which consists in maintaining flow through the penstock and the turbine nozzles, was therefore envisaged to reduce flow variations in the short-circuited section of the scheme, in which people may be found, particularly during the summer season (safety stake).

To validate the feasibility of this type of operating mode and, specifically, to verify the structural integrity in operation of the components impacted by the jet (deflector, control rod, jack...), studies were jointly carried out by EDF and CETIM, and a test-calculation program was initiated. This program included preliminary calculations using a finite element model, a campaign of tests in operating conditions (with vibration measurements and dynamic extensometric measurements) and an analysis of the potential damage due to fatigue. In the end, the study made it possible to show that the damage due to fatigue resulting from the operation of the Bois turbine in bypass operating mode remained in an acceptable range.

© 2013 The Authors. Published by Elsevier Ltd. Open access under CC BY-NC-ND license.
Selection and peer-review under responsibility of CETIM

Keywords: fatigue damage, fatigue design, hydraulic turbine, simulation

* Corresponding author. Tel.: +33 472 827 769.
E-mail address: marc.pillou@edf.fr
1. Introduction

The Bois power plant, comprising a single Pelton turbine, turbines the meltwater of the Mer de Glace in Chamonix. The water intake is below the glacier. The intake structure does not have sufficient storage capacity to avoid water overflowing into a “short-circuited section” (SCC) in case of turbine load tripping.

The SCC is located in a highly touristic zone and it is highly probable that there will be people present in the canal itself. In order to reduce the risks to human safety caused by a sudden increase in the water level, the current operating instruction involves diverting an alert wave with a flow of 4 m³/s through the canal for one half hour prior to shutting down the structure. This duration corresponds to the travel time of the wave in the SCC.

At the end of the half hour, the persons alerted by the rise in the water level must have evacuated the SCC. The discharge from the Mer de Glace can then be fully diverted into the SCC without risking taking anyone by surprise. This current operating measure does not cover certain turbine load tripping situations. In this case, the fact that it is impossible to store the water at the intake structure means that all of this discharge must flow through the SCC. The discharge at the structure’s maximum capacity is 16 m³/s and can lead to major risks for a person in the canal.

The proposed solution consists in operating the turbine in “bypass” mode, i.e. sending the discharge minus the alert flow of 4 m³/s through the nozzles. The jets are deflected by the deflectors, for one half hour (duration of alert flow). This study analyses the structure's mechanical behavior (deflector / control system / nozzle tip support) in order to determine whether it can function in a prolonged manner without incurring damage. In particular, we studied the risk of damage to these components due to fatigue.

2. Working method

A 5-stage working method combining calculations and tests was implemented:

- Finite element modeling of the nozzle tip support + deflector structure with the mechanical loading of the deflector by the jet. Determination of the field of static stresses and the structure’s theoretical eigenmodes.
- Experimental model analysis with model-measurement comparison.
- Extensometric and vibration measurements (accelerometers) of the dynamic stress levels on the different transients (BD at different loads) and during prolonged operation in bypass mode.
- Based on the acquisitions from the strain gauges, comparison of the damage due to fatigue during prolonged operation on the deflectors with the damage due to load tripping at maximum capacity.
- Performance of fatigue calculations to predict the potential number of hours of operation in bypass mode, with a reasonable confidence index.

3. Finite element calculations

3.1. Static stress field calculation

The FE modeling of a nozzle tip support, a deflector and its control system to which we applied the partial hydraulic thrust of a jet, made it possible to determine the static stress field. The position of the deflector was such that the deflected water avoided impacting the runner.
These calculations reveal the zones in which the Tresca-equivalent stress is highest:

- **For the nozzle tip support**: rear section of the deflector fixing lugs under tension and front section of the deflector fixing lugs under compression;
- **For the deflector**: transition zone near the axis under tension.

Following the calculations, the positioning of the strain gauges was further refined (see §4). However, the low connecting radii in the most highly stressed zones of the nozzle tip support did not allow them to be equipped with strain gauges. The installation of gauges in the current section of the lugs (gauges n° II and V) was decided to recalibrate the finite element model on the test results.

3.2. Determination of the structure’s eigenmodes

The calculation of the first eigenmodes of the structure comprising the nozzle body, the nozzle tip support, the deflector and its control system was carried out prior to the tests. The objectives were:

- to understand the structure’s principal strain modes in order to select the zones to equip with strain gauges;
- to check that the initial calculated eigenfrequencies were high enough not to be excited by the hydraulic forces;
- following a calculation-measurement comparison, to allow the verification of the appropriateness of the calculation model’s boundary conditions.

The values derived from the finite element calculations are given in the table below.
Table 1: Eigenmodes for the structure calculated using the finite element model

| Mode # | Mode de déformation | Frequency (Hz) |
|--------|---------------------|----------------|
| 1      | Déformation tige et fond du vérin (plan xy) | 68              |
| 2      | Déformation tige et fond du vérin (plan xz) | 90              |
| 3      | Déformation tige et fond du vérin (plan xz) | 117             |
| 4      | Déformation tige et fond du vérin (plan xz) | 120             |
| 5      | Déformation tige et déflecteur (en opposition de phase) | 134             |
| 6      | Déformation d'ensemble – rotation autour de l'axe y | 144             |
| 7      | Déformation d'ensemble autour de l'axe y (tige du vérin et corps en opposition de phase) | 168             |
| 8      | Déformation d'ensemble autour de l'axe y (tige du vérin et corps en opposition de phase) | 175             |
| 9      | Déformation d'ensemble autour de l'axe y (tige du vérin et corps en opposition de phase) | 188             |
| 10     | Déformation d'ensemble autour de l'axe y (tige du vérin et corps en opposition de phase) | 229             |
| 11     | Déformation d'ensemble autour de l'axe x | 290             |
| 12     | Déformation d'ensemble dans le plan xz | 332             |
| 13     | Déformation de la tige du vérin et de la chaîne du déflecteur | 343             |
| 14     | Déformation d'ensemble dans le plan xz | 390             |
| 15     | Déformation de la chaîne du déflecteur et de la barre de support | 425             |

Mode # | Frequency (Hz) | Mode de déformation |
|--------|----------------|---------------------|
|        |                | Rod & cylinder base strain (xy plane) |
|        |                | Rod & cylinder base strain (xy plane) |
|        |                | Rod & cylinder base strain (xy plane) |
|        |                | Deflector swing |
|        |                | Cylinder rod and deflector swing (in 180° phase shift) |
|        |                | Structure mode – rotation around y axis |
|        |                | Cylinder rod and deflector swing (xy plane) |
|        |                | Cylinder rod and deflector swing (xz plane) |
|        |                | Structure swing (body and deflector in 180° phase shift) |
|        |                | Swing of structure around y axis (cylinder rod and body in 180° phase shift) |
|        |                | Rotation around x axis |
|        |                | Structure strain in the xy plane |
|        |                | Cylinder rod and deflector fork strain |
|        |                | Structure strain in the xz plane |
|        |                | Deflector fork and control rod strain |

4. Modal analysis and instrumented tests in operation

Tests made it possible to characterize the structure’s {nozzle + deflector} vibratory and mechanical behavior. These tests comprised the following stages:

- Experimental modal analysis of structure {nozzle + deflector + control system} when the turbine was stopped to identify the principal eigenmodes associated with the vibrations in operation and to identify their relative contributions.
- Measurement and recording of strain (gages) in operation in the most stressed zones of the deflector and its control system, which were localized by the experimental modal analysis and finite element modeling.
- Vibration measurements in operation on the deflectors and control jacks.
- Measurement of rotation speed in order to detect a possible rotation in the wrong direction of the unit in bypass operating mode.

CETIM was in charge of installing the strain gauges on the deflector and its control system and of signal acquisition during the tests in bypass operating mode.
4.1. Experimental modal analysis of the structure (nozzle + deflector)

The full experimental modal analysis of nozzles 2 and 5 (deflectors and their control jacks) was carried out by DTG/CMS. The structure was excited using a shock hammer in the YZ direction and the responses were measured using accelerometers positioned on other points.

![Figure 2.a](image1.png)  ![Figure 2.b](image2.png)

Figure 2.a: Overall FE model {nozzle + deflector + jack}  Figure 2.b: Experimental modal analysis model

![Figure 3](image3.png)

Figure 3: 1\(^{st}\) bending mode of the structure in a vertical plane

The first experimentally identified eigenmodes for the nozzles (in a complete manner for nozzles 2 and 5 more succinctly for the 4 other nozzles) are the following:

- 92 Hz: structure bending in the vertical plane \(Z\);
- 113 Hz: structure bending in the horizontal plane \(Y\);
- 270 Hz: lug bending in the vertical plane \(Z\).
4.2. Tests in bypass operating mode

4.2.1 Instrumentation

Structures 2 & 5 (nozzle + deflector) were equipped with instruments. Strain gauges were fixed in 13 different zones on each nozzle, representing 28 measurement channels (see figures below):

- **On the nozzle**: 4 tri-directional sets of gauges at the nozzle body – lug junction (I to IV) and 2 bi-directional sets of gauges on the top of the lugs (V & VI);
- **On the deflector**: 2 tri-directional sets of gauges at the fillet near the deflector’s axis of rotation (VII & VIII) and a bi-directional set of gauges on the external face of the deflector (IX);
- **On the cylinder rod**: 4 single gauges (X to XIII) parallel to the rod, in 90° increments.

Moreover, the deflectors and the cylinder housings were equipped with tri-axial accelerometers.

- **On the deflectors**, the accelerometer provided information on the displacement, which was then directly correlated with the strain measured by the gauges.
- **The vibratory measurement instrumentation on the cylinder housing** then confirmed or invalidated the jack’s eigenmodes identified by the SPRETEC calculation.

Careful attention was paid to the passage of the gauge connection cables to ensure that they would not be damaged by the pressurized water.
4.2.2 Performance of the tests

The vibration levels and stress levels were recorded for the following types of operation:
- Full load with normal blocking default (current deflector usage conditions): “P_{max} Normal BD”
- One quarter load in bypass operating mode (nozzles 2 & 5 open to 46%): “¼ bypass”
- Half-load in bypass operating mode (nozzles 2, 5, 1 and 4 open to 45%): “½ bypass”
- 75% of load in bypass operating mode (6 nozzles open to 45%): “¾ bypass”
- Full load in bypass operating mode (6 nozzles open to 92%): “P_{max} bypass”

The following sequence was followed for each test:
- opening of the isolation valve leading to the pressurizing of the nozzle,
- opening of nozzles,
- operation in steady regime,
- positioning of the deflector opposite the jet (passage in bypass operating mode),
- immediate closing of the nozzle in the case of the “Pmax BD Normal” test and after 1 to 2 minutes for the other tests,
- closing of the valve.

Strain measurement signal acquisition was carried out at 1200 Hz, with the frequency of the filters-amplifiers set at 500 Hz. Despite the fact that they were highly exposed to the impact of the water deflected by the deflectors, the gauges and cables did not suffer any damage, except for one gauge (electrical fault after the interruption of electrical continuity after the 3rd test). The acquisition of strain signals was thus carried out for all of the tests.

4.2.3 Dynamic analysis of strain gauges

For each operating test, the strain gauges quantified two magnitudes (see figure below):
- a level of mean stress: $S_m$;
- a stress range $\Delta S_m$.

![Figure 7: Example of a temporal signal of stresses (gauge VII on nozzle 2 for bypass operating mode tests at $\frac{1}{4} P_{max}$)](image)

The next table indicates all of the stress fluctuation levels and the mean stresses for gauges I, II, III, IV, VII & VIII (which are the 6 characteristic gauges in terms of damage), for each nozzle and each operating regime.

| Table 2: $S_m$ and $\Delta S_m$ levels for the different load cases tested |
|-----------------------------------------------|
| Gage | $\frac{1}{4}$ Charge | $\frac{1}{2}$ Charge | $\frac{3}{4}$ Charge | P$_{max}$BD normal | P$_{max}$Déchargeur |
|------|-----------------|-----------------|-----------------|-----------------|-----------------|
| RI   | $\Delta S_{max}$ | 12 | 15 | 12 | 15 | 12 | 15 | 12 | 15 | 12 | 15 | 12 | 15 | 12 | 15 |
| Sm   | $S_m$ | 85 | 60 | 70 | 65 | 60 | 66 | 60 | 65 | 60 | 65 | 60 | 65 | 60 | 65 | 60 |
| RII  | $\Delta S_{max}$ | 6 | 19 | 24 | 56 | 20 | 55 | 18 | 49 | 13 | 31 | 18 | 39 | 11 | 33 | 8  |
| SIII | $S_m$ | 70 | 65 | 66 | 68 | 65 | 68 | 66 | 70 | 67 | 74 | 68 | 71 | 69 | 73 | 44 |
| RIV  | $\Delta S_{max}$ | 5 | 7 | 11 | 5 | 11 | 13 | 5 | 11 | 13 | 11 | 13 | 13 | 11 | 13 | 11 |
| SIV  | $S_m$ | 55 | 50 | 49 | 56 | 49 | 51 | 49 | 51 | 50 | 53 | 50 | 52 | 50 | 53 | 43 |
| RVII | $\Delta S_{max}$ | 10 | 8 | 11 | 5 | 11 | 13 | 5 | 11 | 13 | 11 | 13 | 13 | 11 | 13 | 11 |
| S VII| $S_m$ | 49 | 44 | 48 | 42 | 49 | 44 | 48 | 44 | 49 | 45 | 48 | 44 | 49 | 45 | 37 |
| R VIII| $\Delta S_{max}$ | 11 | 8 | 11 | 5 | 11 | 13 | 5 | 11 | 13 | 11 | 13 | 13 | 11 | 13 | 11 |
| S VIII| $S_m$ | 36 | 33 | 37 | 31 | 37 | 32 | 38 | 36 | 37 | 38 | 36 | 37 | 38 | 37 | 38 |

We observed that the lowest fluctuations were generally detected for the $\frac{1}{4}$ load regime. The corresponding damage was therefore lower. Depending on the considered nozzle, the regime with the highest applied force was BD normal.
at $P_{\text{max}}$ or BD normal at ½ load.

Major fluctuations were measured on gauges VII and VIII (situated at the deflector/deflector axis junction), and they were likely to present high levels of damage. The frequency analysis was therefore focused on these zones which were considered as potentially critical.

The spectral analysis of gauges VII and VIII at different regimes (½, ¾ and $P_{\text{max}}$ bypass) revealed the following:

- Operation in bypass mode at ½ load is the most penalizing operating condition; this information was correlated with the accelerometer located on the deflector.
- Two domes (90 Hz and 120 Hz) corresponding to two eigenmodes identified during the experimental modal analysis (vertical and horizontal bending of structure) emerge from the background spectrum.
- The stresses measured at these frequencies of 90 Hz and 120 Hz, are respectively in the order of 2 MPa and 13 MPa effective (corresponding to a crest-to-crest displacement of 0.23 mm).

The spectra of the signals from gauges VII & VIII were strongly correlated with the crest-to-crest displacement measured by the accelerometer situated on the deflector.

4.2.4. The effect of stress concentration and positioning uncertainty

The comparison between the stress fields derived from the finite element model and from the positioning of the strain gauges shows that the linear deformations are not exactly measured at the point at which the model predicts the maximum stress. The figure below shows the average positioning of the gauges with respect to the finite element model’s peak stress in the case of gauge VII.

Figure 8: Superimposition of displacement and stresses measured at the level of the deflector (nozzle 2 – test in bypass operating mode at ½ $P_{\text{max}}$)
In order to estimate real damage in the zones of highest stress, we tried to estimate the level and range of the stresses in these zones directly from measurements. To do this, for each fillet, we defined an increasing coefficient for the stress measured by the strain gauges in order to take into account both the stress concentration effect between the instrumented zone and the fillet, and gauge positioning uncertainty. To this end, we made 2 assumptions:

- We assumed that the ANSYS calculation is representative of the stress concentrations, i.e. that the stress ratio between the instrumented zone and the zone with the highest applied force is representative of reality. We note that the $K_T$ stress concentration coefficient:
  
  $K_T = \frac{\sigma_{1}^{\text{max}}\text{(ANSYS in the fillet)}}{\sigma_{1}^{\text{avg}}\text{(ANSYS on the gauge at the instrumented point)}}$

- We took into account a positioning uncertainty of roughly 5 mm for the strain gauges. For each gauge, a stress reading was carried out on the finite element model results on a disk roughly 5 mm in diameter. This reading made it possible to evaluate the uncertainty due to the location of the gauge. This uncertainty was characterized by a coefficient $K_{\text{LOC}}$, corresponding to the ratio between the upper and lower maximum principal stresses on the uncertainty disk ($\sigma_{1}^{\text{sup}}, \sigma_{1}^{\text{inf}}$):
  
  $K_{\text{LOC}} = \frac{\sigma_{1}^{\text{sup}}\text{(ANSYS)}}{\sigma_{1}^{\text{inf}}\text{(ANSYS)}}$ ($\sigma_{1}$ designates the 1st principal stress, corresponding to the strain measurement direction).

Finally, we performed the damage calculation by increasing the stresses measured by the gauges by the coefficient $K_G = K_{\text{LOC}} \times K_T$. We considered that the damage thus calculated is the damage envelope for the stress concentration zone.

The table below indicates the coefficients $K_T$ and $K_{\text{LOC}}$ for the zones with the highest applied force. Depending on the zones, the coefficient $K_G$ ranges from roughly 1.5 to 3.

Table 3: Increasing coefficients for stresses to be applied in the zones with the highest applied forces

|        | Calcul EF | Measure | $\sigma_{\text{D,inf}}$ | $\sigma_{\text{D,sup}}$ | $\sigma_{\text{max}}$ | $\sigma_{\text{mov}}$ | Inj2 | Inj5 | $K_{\text{LOC}}$ | $K_T$ | $K_G$ |
|--------|-----------|---------|--------------------------|--------------------------|-----------------------|-----------------------|------|------|------------------|------|------|
| JI     | 56.3      | 61      | 83                       | 59                       | 79                    | 64                    | 1.08 | 1.41 | 1.52             |
| JII    | 53.6      | 57.6    | 78                       | 56                       | 66                    | 70                    | 1.07 | 1.39 | 1.49             |
| JVII   | 59.2      | 86.5    | 138.6                    | 69                       | 48                    | 44                    | 1.46 | 2.0 | 2.92             |
| JVIII  | 49.1      | 65.5    | 117                      | 54                       | 38                    | 36                    | 1.33 | 2.2 | 2.93             |
4.3. Conclusions on the test results

A first analysis of the displacements and stress levels gave reason to be confident about the static structural integrity of the structure (deflector + nozzle + control jack). In order to validate the feasibility of bypass operating mode, a fatigue analysis with a high number of cycles and a compared damage-type analysis were carried out. These calculations were made using the nCODE ® fatigue calculation software.

5. Analysis of the calculation-measurement differences and recalibration of the finite element model

5.1. Static analysis of the strain gauges

A comparison between the results from the finite element model and the stresses measured by the strain gauges showed that the mean stresses measured by the strain gauges were well below those obtained by the 1st version of the finite element calculation.
- According to this 1st calculation, the zone subject to tensile stress at the base of the lugs showed maximum stress in the order of 120 MPa, while the strain gauges (gauges I & II) indicated a stress of roughly 70 MPa.
- In the same way, the 1st finite element model gave a stress in the order of 180 MPa at the deflector/deflector axis junction, while the strain gauges indicated a stress of roughly 40 MPa (gauges VII & VIII). This difference between the calculations and the measurements is due to different causes:
  - The static thrust on the deflector was calculated by applying the theorem of momentum. Depending on the angle of deflection of the jet, the mechanical loading thus estimated could be overly high.
  - The dissipation of energy resulting from the water sent back by the deflector interfering with the jet was not taken into account in the loading applied to the finite element model.
  - The boundary conditions of the finite element model were also an increasing factor (recess installation of nozzle body connection, perfect pivot connection of the deflector on its axis of rotation, etc.). The result was that the linear deformations and energy dissipation were not taken into account at the level of the connections.

5.2. Resetting of the mechanical loading

The differences mentioned above led to resetting the calculation model’s loading hypotheses. A posteriori, the resetting of the calculation taking into account a reduction in the angle of deflection of the jet, made it possible to recalibrate the results of the model on those of the tests. The resetting of the deflector’s closing angle according to a minimum value ensuring sufficient jet deflection allowed the reduction of the jet’s hydrodynamic thrust and of the resulting stress levels in the entire structure. The new calculation takes into account a jet impact load of 135 kN instead of the 226 kN in the first calculation. The level of stress calculated in the current section of the lugs after this loading modification coincides with the mean static stress measured by gauges II & V.

5.3. Verification of the structure’s eigenmodes

The comparison of the eigenmodes calculated by finite element model with the experimental modal analysis revealed important differences:
- The modal analysis carried out on site by DTG revealed two model responses of the structure at 92 and 113 Hz, not identified by the finite element model. They are 2 bending modes for the nozzle body around the X and Z axes, rotation axes of the “recessed beam” formed by the nozzle body.
- Conversely, the experimental modal analysis did not reveal the calculated eigenmodes corresponding to the cylinder housing bending modes (88, 90, 117 Hz).
These differences are probably explained by the finite element model's boundary conditions (stiffness of the recessing supposed to be infinitely rigid) and by some modeling simplifications (needle modeled by added mass). This comparison clearly shows the contribution of experimental measurements that make it possible to both eliminate the restriction of overly conservative hypotheses and to recalibrate the calculation model. The perfect recessing condition can notably explain that the first bending mode was calculated at a frequency much higher than that observed during the experimental measurement (respectively 332 Hz and 380 Hz).

6. Calculations of fatigue-induced damage

6.1. Principle

The method used for the calculation of fatigue-induced damage is a stress based approach, based on a S-N curve, otherwise known as the Wohler curve. This type of approach is adapted to safety-related structures or components intended to operate for a very high number of cycles.

6.1.1. Occurrence of operation in bypass mode

The analysis of faults recorded over a 10-year period, leading to a possible operation in bypass mode from 05h00 to 22h00 from June to September, leads to an occurrence of 1 operation per year. To be on the safe side, we take into account two operations per year over a ten-year period, i.e. 10 hours of cumulative operation.

6.1.2. Wohler's curve

Wohler's curve defines the relationship between the stress amplitude and the number of cycles or life cycle for a given material, with a given probability of failure and mean stress. This curve is generally established experimentally for zero mean stress and a probability of failure of 50%. The fatigue-induced damage calculations were performed with a mean stress correction law and a probabilized Wohler curve. The transformation into pure alternating stress was carried out using the Goodman formula, which takes into account alternating stress, the material's nominal strength ($R_m$) and static stress:

$$\sigma_{alternating} = \frac{\sigma_{alternating}}{1 - \frac{\sigma_{static}}{R_m}}$$

At this stage, the alternating stress values were measured at 4 different points then averaged. The mean measured static stress was corroborated by the calculation model after its recalibration. As the real Wohler curve of the analyzed component was unknown, an envelope curve, according to Basquin's model ($S=C.N^b$), was built based on an estimate of an endurance limit, a slope $b$ and the consideration of certain influencing factors.

- We estimated the mean endurance limit at $10^4$ cycles as of the material's ultimate tensile strength: $\sigma_D = 0.45 \times R_m = 0.45 \times 480 = 216 \text{ MPa}$.
- With the basic hypothesis of a normal distribution of fatigue resistances and a coefficient of variation estimated at 10%, a probabilized endurance limit at -3 standard deviations was estimated by:

$$\sigma_{Dp} = 216 - 3 \times 0.1 \times 216 = 151 \text{ MPa}.$$

- Two additional reductions to this probabilized limit of roughly 30% to take into account the surface condition effect (molded part) and 30% for the scale effect (size of the part) were also taken into account:

$$\sigma_{Dpref} = 151 \times 0.7 \times 0.7 = 74 \text{ MPa};$$

- Slope $b$ is estimated at -0.2 (steep slope corresponding to Eurocode recommendations).
The generic curve thus created is a steep curve, making it possible to increase the value of fatigue-induced damage.

6.1.3. Miner's rule and counting of fatigue cycles

On the most highly stressed zones, we carried out Rainflow-type counting, which consists in building a histogram of cycles according to the mean stress $S_m$ and for the stress range $\Delta S_m$ using the stress's temporal signal $S(t)$. This counting is performed for each operating regime. Based on this histogram, we then built a histogram of associated damages for each pair (mean stress $S_m$ & stress range $\Delta S_m$) using the Wöhler curve and Miner's rule. A Goodman correction of mean stress was applied to take into account the cycles containing a static component. The total damage was obtained by the linear aggregation of partial damages, and we considered that failure due to fatigue is reached when the overall damage is equal to 1.

6.2. Analysis of the results

The damage was calculated for a signal duration of 60 seconds, corresponding to the steady regime after opening the nozzles and positioning the deflectors. This duration was taken as the reference for all the compared cases. In order to easily compare the relative level of damage at each load with the reference damage corresponding to normal load tripping at $P_{\text{max}}$, we calculated for each load case, the relative damage ratio: $\rho = \frac{D(i)}{D_{\text{ref}}}$ (designating the damage for 60 seconds of operation in bypass mode at load $i$ and $D_{\text{ref}}$ designating the cumulative damage for a normal BD stop at $P_{\text{max}}$ also called the “reference damage”).

The zones that accumulate the most fatigue-induced damage are the fillets of the deflector at its axis, whose stresses are deduced from gauges VII and VIII. The figure below makes it possible to visually compare the temporal signals of raw stress for the different loads tested on the gauge with the highest applied force.
Reference case: BD normal at $P_{\text{max}}$:

Operation in bypass mode at $\frac{1}{4} \times P_{\text{max}}$:

Operation in bypass mode at $\frac{1}{2} \times P_{\text{max}}$:

Operation in bypass mode at $\frac{3}{4} \times P_{\text{max}}$:

Operation in bypass mode at $P_{\text{max}}$:

Figure 12: “Raw” temporal signals of stress in the most highly loaded zone

The 2 following figures make it possible to compare the relative damage levels for the different bypass operating modes (from $\frac{1}{4} \times P_{\text{max}}$ to $P_{\text{max}}$).

Figure 13: Compared damage levels in the most highly loaded zone

We observe that:

- The damage for 60 seconds at $P_{\text{max}}$ in bypass operating mode is of the same order of size as the reference damage.

- The damage for 60 seconds at $\frac{1}{4} \times P_{\text{max}}$ in bypass operating mode is much lower than the reference damage.

- The damage for 60 seconds at $\frac{3}{4} \times P_{\text{max}}$ in bypass operating mode is significantly higher than the reference damage (ratio of 2 to 10 depending on the case).
The damage for 60 seconds at ½ \( P_{\text{max}} \) is always the highest, with a ratio of 6 to 40 compared with the reference damage.

However, the highest damage estimated according to the measurements (gauge VIII nozzle 2 at ½ \( P_{\text{max}} \)) remains very low in absolute value: roughly \( 1.3 \times 10^{-4} \) over a one minute operating period. Given the very conservative hypotheses made for the increasing coefficient for stress, on one hand, and the choice of the Wöhler curve, on the other, we can consider that this level of damage represents a very conservative upwardly-adjusted value for the real damage.

The extrapolation for 10 hours of operation in bypass mode (corresponding to 10 years of operation) leads to an upwardly-adjusted value for damage: \( D_{\text{max}}(10 \text{ years}) < 10^{-1} \). We observe that this upwardly-adjusted value remains largely below 1, which guarantees the absence of the risk of deflector fatigue failure, even with operation systematically at the most penalizing load ½ \( P_{\text{max}} \).

6.3. Specific case of the zone under compression loading

The zone situated at the front section of the lugs, which is under compression loading, could not be equipped with instruments. As a result, we didn’t directly have the alternating stress value for this zone. However, we know that the static peak stress, calculated after the recalibration of the finite element model, is close to 310 MPa. This very sporadic, high value exceeds the material’s theoretical elastic limit \( R_{\text{p0.2}} = 240 \text{ MPa} \).

This is due to the angular shape, which creates stress concentrations. Compression loading reduces the risk of cracking, while a local yielding phenomenon makes it possible to envisage an increase in the elastic limit via the local strain hardening of the material.

The nature of the applied force and their levels lead us to suppose that this zone is not fatigue critical. Nevertheless, during the changing of the nozzle tip supports, we are considering blasting with micro beads to locally increase hardness.

7. Conclusions

7.1. Summary of the study results

The different investigations carried out to understand the mechanical behavior of the nozzle + deflector structure of the Bois turbine operating in bypass mode, revealed different phenomena in a coherent and cross-cutting way:

- The static stress levels derived from a finite element calculation were acceptable after recalibration of the FE model using tests.
- The structure’s modal response in operation did not reveal any abnormal resonance phenomena that could be harmful to prolonged operation on the deflectors.
- The study of fatigue with a high number of cycles, based on the processing of signals from gauges placed in the most highly loaded areas and upwardly adjusted by a coefficient integrating an uncertainty with regard to the position of the gauges and stress concentration, demonstrates the material’s capacity to accept a number of cycles corresponding to at least 2 bypass operating sequences per year over 10 years (or a total of 10 hours of operation in bypass operating mode).
- Depending on the machine loading during the operations in bypass mode, the compared fatigue-induced damage study in the most highly loaded zone shows that 10 years of operation in bypass mode are equivalent to a number of load triggers at \( P_{\text{max}} \) ranging from 600 (operation at \( P_{\text{max}} \)) and 24,000 (operation at ½ \( P_{\text{max}} \)).

7.2. Recommendations

In view of these investigations, and subject to the absence of pre-existing faults in the material, the operation of the Pelton turbine in bypass operating mode in the Bois power plant may be envisaged. The systematic bleeding of these zones allowed the verification of the absence of emerging defects. Nevertheless,
given the age of the molded parts currently in place and the manufacturing imperfections inherent to the manufacturing period, we cannot exclude the presence of internal faults near the surface in areas with a low radius of curvature. This uncertainty must be considered as an important limit to the study that was presented. This study does not take into account the potential presence of a sizeable manufacturing defect, close to the surface and capable of spreading with fatigue according to kinetics that could best be described by a Paris Law type propagation approach rather than a Wöhler curve approach.

In view of this residual uncertainty, we recommend that a visual inspection be systematically performed and that the most highly loaded zones be bled after each long period of operation in bypass mode. The aim of this inspection is to compensate for the limits of the current study.

7.3. **Proposal for improvement**

In the long term, it will be possible to eliminate this stress linked to the poor quality of the material. We plan to replace the nozzle tip support and the deflectors with molded elements in 16.4 grade stainless steel. The surface condition of the stress concentration zones will be closely monitored in order to minimize the risk of developing cracking in these zones. Finally, a campaign of Wöhler curve characterization tests for this grade will be launched in 2012 to enable a more precise evaluation (which should be less severe) of the level of damage expected for a high number of operations in bypass mode. This choice of material will make the systematic inspection of each part after each operation in bypass mode unnecessary.