Engineering diagnostics for vortex-induced stay vanes cracks in a Francis turbine

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Abstract. Despite the fact that vortex-induced vibration (VIV) in hydraulic turbines components (especially in stay vanes) is a well-known phenomenon, it still remains challenging for operation and maintenance teams in several power plants around the world. Since the first publication of a similar problem in 1967, literature shows that at least 27 other turbines witnessed strong stay vane vibrations associated with vortex shedding. Recurrent stay vane cracks in a 250 MW Francis turbine in Brazil motivated an engineering study involving prototype measurements, structural and Computational Fluid Dynamics (CFD) analysis in order to determine a proper geometry modification that could eliminate the periodic vortex wake generated at the stay vanes trailing edge. First cracks appeared in 1978 just after the machine was put into operation. A study published in 1982 associated these cracks with dynamic excitations caused by the water flow at high flow conditions. New stay vane profiles were proposed and executed as well as improved welding recommendations. Cracks however, continued to appear requiring welding repairs roughly every two years. Although Voith Hydro was not the original equipment manufacturer for these units, the necessary information was available to study the issue and propose and execute new stay vane profiles. This paper details the approach taken for the study. First, indirect vibration measurements were used to determine vibration frequencies to help to characterize the affected mode shapes. These results were compared to finite element (FE) calculations. Strain gage measurements performed afterwards confirmed the conclusions of this analysis. Next, transient CFD calculations were run to reproduce the measured phenomenon and to serve as a basis for a new stay vane geometry. This modification was then implemented in the actual turbine stay vanes. A new set of indirect vibration measurements indicated the effectiveness of the proposed solution. Final confirmation will come from new strain gage measurements.
1. Introduction
On 1978 when the first of the planned six 250 MW Francis machines started to operate, strong vibrations appeared on the turbine for outputs greater than 205 MW and when the waterway was dewatered and inspected, cracks were found in some of the stay vanes. This phenomenon was associated with vortex-induced vibration (VIV) on the stay vanes. This turbine has 24 stay vanes divided in 3 different geometries: first includes stay vanes 1 to 21 (geometry 1), second corresponds to vanes 22 and 23 (geometry 2) and finally stay vane 24 (geometry 3).

![Figure 1](image.png)

Figure 1. Plan and section view of the stay ring.

Analysis performed at that time [1] indicated that although the excessive vibration could be measured on different types of vanes, the first group (1-21) was the most critical one. At higher turbine flow, these vanes were vibrating in the first bending mode and measured dynamic stress amplitude exceeded 40 MPa.

A trailing edge modification introduced by the original manufacturer on all the vanes was able to increase the frequency of the flow induced forces and the maximum measured stress amplitude dropped to 8 MPa. Unfortunately this was not enough to prevent the cracks from keeping coming back repair after repair, although at a slower rate. As the exact measurement locations at the monitored stay vanes are not presented in the original publication [1] it is not possible to confirm that they were not in a new resonance condition with the first torsional mode or even if other non-monitored stay vanes were. After dealing with the cost and time associated with several stay vane crack repairs (roughly each two years), the owner decided to include in the rehabilitation scope of the power plant an engineering study containing a full problem characterization involving vibration and strain measurements, structural and CFD analysis in order to develop a new and definitive geometry modification for all the stay vanes to finally eliminate this problem.

2. Brief review of stay vane cracks
Since the first published case of stay vane cracks on 1968 [2] at least 27 other cases of unusual vibrations in the stay ring of turbines of different manufacturers were reported [3], the most recent one
in 2015. This clearly indicates that either the phenomenon is not completely understood or the knowledge is not sufficiently spread.

**Stay vane vibrations**

![Graph showing Turbine output vs Year when cracks or noise were first noticed](image)

**Figure 2.** Reported cases of cracks or noises related to stay vanes.

In most cases the problem was related to hydraulic dynamic forces induced by vortex shedding associated with a resonance in the first bending or torsional mode and the solution involved modifications in the stay vane trailing edge.

To better understand this phenomenon it is important to point out why it started on the late sixties. Around 1960 the size and output of the hydraulic turbines were not much different from the ones of the early post-war years but by the end of this decade unit outputs had increased fourfold [4]. At that time the development was focused on the low head machines of Russia, North and South America which coupled with the high outputs implied in big turbines. The motivation for bigger and more powerful machines, then and now, is economics since this reduces the cost per megawatt.

The huge size of these turbines also forced a modification in the manufacturing process with the cast and bolted stay rings being replaced by welding constructions, with some welds performed at site. This led to smaller transition radius and more concerns regarding a proper stress relief.

Finally, a new mechanical design, see figure 3, was proposed which reduced the bending stress on the vanes alloying them to be thinner while keeping the same stress levels. This, however, lowered the natural frequencies of these stay vanes. A similar trend continues today as the more sophisticated mechanical calculation methods allow smaller stresses safety factors and, as a consequence, slender stay vanes.

![Comparison between old and new design of stay vanes](image)

**Figure 3.** Comparison between old and new design of stay vanes [1].
All these modifications together created the environment for the appearance of these cracks. It is worth mentioning also that due to the characteristics of the phenomenon, vortex shedding on the stay vanes cannot be analyzed on a standard turbine model test.

Most of the problems followed a certain pattern: noise or vibration appeared for distributor openings greater than 70% and although they were reason for concern they were not considered serious enough to prevent the machine to start commercial operation. However, in the first inspection, after around 2000 hours of operation, cracks were found at the connection between the stay vanes and the decks of the stay ring. For turbines with different geometries of stay vanes it was also quite common that cracks would appear only in some of them usually the most flexible ones. The solution in the majority of cases consisted in weld repair and modifications on the trailing edge. Since detailed transient flow calculations were not available, the whole process followed a trial and error approach and only after months of successful operation and few inspections, the problem could be considered solved. Due to these characteristics, situations where the crack propagation actually did not stop but rather had their propagation rate reduced are not uncommon.

3. Indirect vibration measurements
Hydroelectric units, especially the most powerful ones, are not easily available for measurements in Brazil. Their availability is monitored by the Electric Energy National Agency and every generation loss has a significant impact on the revenues. This is particularly critical when these measurements require the complete dewatering of the machine. Therefore, instead of installing strain gages on the stay vanes, this study started with Piezoelectric Acceleration Transducers mounted at the turbine head cover at different circumferential positions with special focus on the region with a higher number of cracks incidences.

First measurements were performed on Power Unit 06 in December 2013. Clear spectral components were identified on the signals as illustrated in figure 5. At 75% of the rated output, spectral components at approximately 155 Hz were very clear. At 80%, another component around 170 Hz became visible and, finally, at 90% of the rated power a third component close to 215 Hz could also be seen with its intensity increasing with the turbine output.

![Figure 4. Location of the acceleration transducers on the head cover.](image)

![Figure 5. Spectral components of vibration measured on Power Unit 06 close to stay vane 23 for different turbine outputs [mm/s RMS].](image)
Similar results were found in a second measurement campaign performed on Power Unit 01 in May 2014. Despite the slight changes in vibration frequency and amplitude, the behavior is essentially the same. As can be seen on figure 6, spectral components around 155, 170 and 210 Hz are still visible.

Figure 6. Spectral components of vibration measured on Power Unit 01 close to stay vane 23 for different turbine outputs [mm/s RMS].

4. Measurement of stay vane geometry at site
Vortex shedding phenomena are extremely sensitive to the stay vane trailing edge geometry, therefore to provide a reliable input for the CFD analysis, a site measurement of the actual geometry is a must.

In February 2014 when Power Unit 02 was stopped and dewatered for its modernization, four different stay vanes (21, 22, 23 and 24) were measured using a coordinate measuring machine.

Figure 7. Geometry measurement at site.

5. Modal analysis
Once the frequencies of vibration in the head cover were measured, the next step was to compare them with the natural frequencies of the stay vanes in water, to evaluate if they were related to some stay vane in resonance.

For each type of stay vane, a 3D model was created for the finite element analysis (FEA). This model corresponded to 1/26 of the stay ring. The actual geometry of the stay vane was taken from the 3D measurements at site. Software Ansys 14.5 was used in the analysis. The effect of the water around the stay vanes was taken into account by fluid-acoustic elements type FLUID 30 and the structure was modeled using SOLID 187. The first three mode shapes (for stay vane 21 as an example) are represented on figure 8, while the calculated natural frequencies are shown on the upper part of table 1. The lower part of the same table presents the natural frequencies obtained from site measurements (bump test). Comparison between the frequencies on table 1 with the frequencies identified by the indirect measurements indicates that most probably the 1st torsional modes of the stay vanes are the ones being excited.
Figure 8. Stay vane mode shapes: (a) first bending, (b) first torsional and (c) second bending.

Table 1. Stay vane natural frequencies calculated by Finite Element Analysis and measured at site.

| Natural Frequencies                                      | Stay Vanes |
|----------------------------------------------------------|------------|
|                                                           | 1-21       | 22&23  | 24  |
| 1st bending mode in air [Hz]                             | 83         | 83     | 81  |
| 1st bending mode in water [Hz]                           | 58         | 60     | 63  |
| 1st torsional mode in air [Hz]                           | 186        | 204    | 258 |
| 1st torsional mode in water [Hz]                         | 159        | 178    | 232 |
| 2nd bending mode in air [Hz]                             | 227        | 226    | 220 |
| 2nd bending mode in water [Hz]                           | 169        | 173    | 178 |
| 1st bending mode in air (experimental) [Hz]              | 88         | 88     | 88  |
| 1st torsional mode in air (experimental) [Hz]            | 190        | 207    | 250 |
| 2nd bending mode in air (experimental) [Hz]              | -          | 234    | 229 |

6. Transient Computational Fluid Dynamics (CFD) calculations

CFD analysis was performed for different stay vane geometries. On the first trimester of 2014, focus was on the existing profiles, later the same method was used to propose a modified geometry for each vane.

6.1. Current geometry

Unsteady CFD calculations were performed for the 3 stay vane geometries using a commercial CFD code based on the Finite Volume Method. First a stay vane profile according to the original manufacturing drawing (with blunt trailing edge) was considered on the simulations. For this geometry, URANS based turbulence modeling (k-ω SST) and unitary velocity as inflow condition, an organized vortex wake was obtained. By extrapolating the results to the expected turbine flow conditions [1], a frequency comparable to the stay vane’s first natural frequency was obtained.

Simulations were then performed for the 3 current geometries according to measurements presented on section 4. In this case URANS simulations were not able to capture a vortex wake. Hybrid RANS-LES methods were then adopted. The SAS-SST model was also not able to capture a vortex wake. The DES model presented more realistic results at this time for the 3 geometries. The computational grid contains approximately 3×10^6 volumes. The average y+ parameter on the stay vane wall was close to 1 and non-dimensional time step adopted (U,Δt/L) was ~1×10^3. A uniform velocity...
profile based on the measured mass flow value was adopted as inlet condition under different angles of attack. No tandem cascade or spiral case were taken into account in the computational domain.

The intensity of the calculated vortex wake increased from geometry 1 to 3. With exception of geometry 1, the higher harmonics (2 and 3) of the fundamental vortex-shedding frequency matched the measured vibration frequencies.

6.2. Modified geometry

Based on previous experiences and references, proposed modifications were limited to the trailing edge and made as small as possible. After some iterations, contours indicated on figure 10 were selected for implementation since they were able to suppress the periodic vortex shedding.

Figure 9. Z-Vorticity contour for the 2nd geometry group.

Figure 10. Modified geometry CFD Z-Vorticity results for geometries 1 (a), 2 (b) and 3 (c).
7. Strain gage measurements

Following its modernization schedule, Power Unit 02 was dewatered and available for strain gage measurements in October 2014. Stay vane 21 did not display any indication of von Karman excitation at that time but stay vanes 22 and 24 showed signs of resonance with the first torsional mode at 169 Hz and 208 Hz respectively. These results corroborate the conclusions inferred by the indirect measurements.

![Figure 11](image1.png)
**Figure 11.** Deformations measured at stay vane 22 before the modification as a function of the output [µS pk-pk].

![Figure 12](image2.png)
**Figure 12.** Deformations measured at stay vane 24 before the modification as a function of the output [µS pk-pk].

8. Indirect vibration measurements after the modifications

First indirect measurements performed after the stay vanes geometry modifications happened in July 2015 on Power Unit 01 and corresponding results are shown on figure 14. For comparison purposes, figure 13 indicates measured vibration on the head cover before these modifications.

![Figure 13](image3.png)
**Figure 13.** Vibrations at the head cover before the modifications [mm/s RMS].

![Figure 14](image4.png)
**Figure 14.** Vibrations at the head cover after the modifications [mm/s RMS].
The indirect measurements performed after the modifications occurred at a higher head than all the other measurements. Assuming that vibrations should start at approximately the same flow rates, it was expected then that by 230 MW vibrations in all stay vanes should have already been observed. Comparing figures 13 and 14, one can see that the high peaks highlighted on figure 13 do not appear on figure 14 indicating that stay vane resonances due to vortex shedding were most likely eliminated. Considering the uncertainties associated with indirect measurements and the fact that operating conditions were significantly different during the two measurements campaigns, a final confirmation of the success of the modifications will come only after the planned strain gage measurements on the stay vanes.

9. Next step
The final step of this study is the direct measurement of the stresses on the stay vanes after the modifications. Again, based on the power plant modernization schedule, these tests should occur at the end of April 2016 on Power Unit 04.

10. Conclusion
The main goal of this study was to eliminate the reoccurrence of cracks on the stay vanes without affecting the availability of the machine in the process. In order to achieve that a special focus was placed on indirect measurements as well as combining every test that required the dewatering of the turbine with previously scheduled maintenances associated with the modernization of the power plant. Although there is still one test pending, the strain gage measurements on the stay vanes after the modifications, some points can already be made:

- Indirect measurements together with detailed background information can give a very good indication about which mode of the stay vane is being affected by the vortex induced phenomena. This is a very useful asset to reduce the unavailability of the machine.
- A really effective flow analysis demands a transient calculation. Steady state simulations together with an adoption of a reference Strouhal number do not have the capability to capture second order effects which may be very important in vortex shedding phenomenon.
- Although not being able to perform all the measurements in the same Power Unit increases the uncertainty of the work, this paper demonstrated that with few compromises it is still possible to combine the availability requirements with the research study.

References

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