# Cavitation Erosion Prediction in Hydro Turbines from Onboard Vibrations


| **Author** | **Firm / Institution** | **City, Country** | **Lecturer (x)** |
|---|---|---|---|
| *Xavier Escaler* | *Technical University of Catalonia-UPC* | *Barcelona, Spain* | *x* |
| *Eduard Egusquiza* | *Technical University of Catalonia-UPC* | *Barcelona, Spain* | |
| *Mohamed Farhat* | *Swiss Federal Institute of Technology-EPFL* | *Lausanne, Switzerland* | |
| *François Avellan* | *Swiss Federal Institute of Technology-EPFL* | *Lausanne, Switzerland* | |


## Abstract

The current work intends to improve the cavitation erosion prediction methodology in hydro turbines by the use of onboard vibration measurements taken on the rotating shaft. For that, a measuring campaign in two similar Francis turbine prototypes has been carried out. In particular, one machine is a well known case of severe erosion on all the blades meanwhile the second machine does not present any erosion at all. A miniature telemetry system has been mounted on the shaft in both cases to transmit the signal from an onboard accelerometer to a receiving antenna. Simultaneously, measurements have also been taken on fixed parts of the machine such as the turbine guide bearing and the guide vane. First, the analysis of the results serves to validate the detection techniques applied to the onboard signals. Then, it is discussed whether the use of the vibrations from the shaft as an alternative to the bearing ones should be an advantage to infer the absolute erosive forces taking place on the runner blades.

## Résumé

La présente étude vise à améliorer la méthodologie de prédiction de l'érosion de cavitation dans les turbines hydrauliques à l'aide de la mesure des vibrations induites sur l'arbre. A cet effet, deux campagnes de mesure sur deux prototypes similaires de turbines Francis ont été réalisées. La première de ces turbines présente une érosion sévère sur l'ensemble des aubes alors que la seconde est exempte de toute érosion. Un système de télémétrie fixé sur l'arbre permet la transmission du signal de l'accéléromètre embarqué vers la partie fixe de la turbine à l'aide d'une antenne. Simultanément, les signaux d'accélération induites sur le palier guide et les aubes directrices sont également enregistrés. D'abord, une analyse des résultats permettant la validation de l'approche vibratoire pour la détection de la cavitation depuis l'arbre est présentée. Ensuite, l'avantage de l'utilisation des vibrations induites sur l'arbre au lieu de celles induites sur le palier guide pour l'estimation des forces agissant sur l'aubage est discuté.

## Introduction

The methodology for cavitation erosion prediction based on vibrations in actual hydro turbine prototypes is investigated in this paper. The work is focused on the methods to be used on operating machines that suffer from severe erosion on the runner blades. These techniques must serve to control the cavitation aggressiveness and its negative consequences at any time. This is of prime importance for the operators of such machines in order to avoid long periods of operation at harmful conditions without knowing the potential risk of damage associated to them. It must be noted that when a given turbine is prone to suffer cavitation erosion there is no immediate solution that can be efficient enough to prevent this hydrodynamic phenomenon. For instance, if the erosion is very strong, a protective coating can serve to slow down the damaging process but it will not be stopped. So, only a correct monitoring of the machine based on vibrations is at the present time the best solution (Abbot *et al.* in Ref 1 and Bajic and Keller in Ref 2).

Up to now, the measurement and analysis of high frequency vibrations has been successfully used to detect the presence of leading edge sheet cavitation on the blades in real Francis turbines (Farhat *et al.* in Ref 3, Bourdon *et al.* in Ref 4 and Escaler *et al.* in Ref 5). In this case, it is generally accepted that the best measuring location is the turbine guide bearing pedestal. However, these remote measurements only give a relative indication of the energy associated to the cavitation. Therefore, another step is needed to quantify the intensity of the cavitation attack on the material being eroded. For this, the current approach is based on the experimental determination of the transmissibility function between the location of the erosion and the measuring position (Bourdon *et al.* in Ref 6 and Bourdon in Ref 7). This is done from calibration tests which consist in the use of instrumented hammers to excite the machine and to measure the response. The main disadvantage of this technique is that during the tests the machine is still and empty of water. This means that, if these functions are to be used during operation to infer the forces being applied to the blade from vibrations on the bearing pedestal, the possible effects of the fluid film between the shaft and bearing and of the water added mass are not being taken into account. Furthermore, the cavitation is taking place in a rotating frame of reference and the measurements are from a fixed one. And finally, the bearing is also receiving noise from other sources of excitation. How all this can influence the results is not well known.

## Experimental investigation

### Prototypes

Two vertical Francis turbines belonging to different hydropower plants, called *FT1* and *FT2*, have been selected for the experimental investigation. A comparison of their main characteristics is presented in Table 1. It has been intended to find two machines of the same type with similar dimensions and power outputs but presenting a completely different cavitation behaviour. In this sense, *FT1* does not show any erosion at all in the runner blades, meanwhile *FT2* suffers erosive inlet cavitation that damages its blades in a significant way.

In particular, *FT1* can operate up to a maximum output power of about 14 MW and *FT2* only reaches 10 MW. *FT1* has a rotating speed, $N$, of 375 rpm and *FT2* of 250 rpm. In both cases, the number of blades, $Z_b$, is 15 and the number of guide vanes, $Z_v$, is 24. Previous results concerning cavitation erosion detection on *FT2* are available in Escaler *et al.* [Ref 8]. Besides, periodic inspections have proved that cavitation erosion is well localized next to the leading edge in the suction side of the blades In fact, after several years of operation advanced stages of mass loss occur due to inlet cavitation.

Exactly the same instrumentation and measuring procedure has been used in both prototypes.

**Table 1 Comparison of Francis Turbine 1 and 2 characteristics.**

| Prototype | Erosive Cavitation | *N* [rpm] | $Z_b$ | $Z_v$ | Max. Power (MW) |
|---|---|---|---|---|---|
| *FT1* | *No* | *375* | *15* | *24* | *14* |
| *FT2* | *Yes* | *250* | *15* | *24* | *11* |

## Instrumentation

Vibrations on the rotating shafts of the water turbines have been measured with an onboard mini telemetry system. The system is composed of very small and light modules that are mounted around the shaft next to the turbine guide bearing. The signal acquisition module for ICP sensors includes signal condition and anti-aliasing low pass filter. It has a 12 bit resolution and is controlled by an Encoder module. This Encoder is a multiplexer that generates a PCM output signal that is in turn connected to a radio frequency transmitter module. The powering of the modules is 5V DC generated by a battery. The receiver station includes a receiving antenna and a 4 channel decoder. Analogue output signals are available via BNC connectors. All channels are simultaneously sampled and the cut-off frequency for a two channel configuration is 6000 Hz per channel. The full system accuracy is about ± 0.5% without the sensors. For these tests the telemetry system has been equipped with a high frequency accelerometer having a mounted resonance frequency of about 52 kHz and fixed to the shaft with a cementing stud.

The same type of sensors has been used to monitor vibrations on the fixed parts of the machines. They have been conditioned and amplified prior to their recording. The instrumentation set-up has also comprised a photoelectric tachometer probe for contact-free detection of the rotating shaft. The analogue output signals have been simultaneously digitized and recorded at a sampling frequency of 48000 Hz.

## Measuring positions

The experimental set-up has comprised four vibration sensors. The accelerometer on the shaft, which is indicated by *SHAFT* in Figure 1, has been glued close to the top end of the turbine guide bearing pedestal. The rest of the accelerometers have been located on fixed parts of the machine. In particular, two of them have been placed on the upper part of the turbine guide bearing wall in radial directions at 90° one from the other. They are named by *A13* and *A14* and their location relative to the penstock is shown in the same Figure 1. The last accelerometer has been positioned on the top of one wicket gate in axial direction, the one located in position *D18* for *FT1* and in position *D13* at 90º in counterclockwise sense from the first for *FT2*.

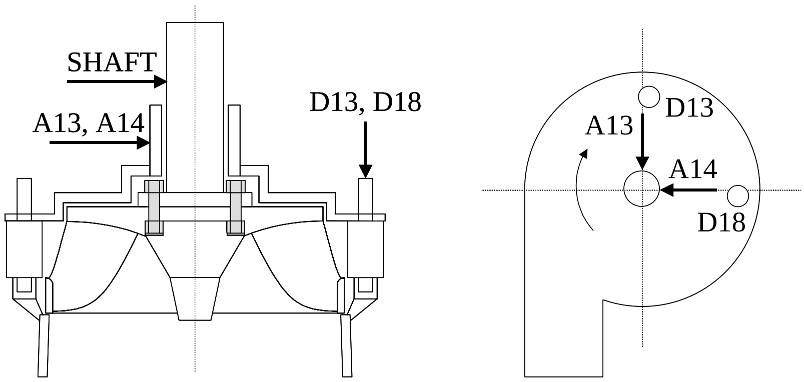


*Figure 1 Outline of a Francis Turbine indicating the location and direction of the vibration measurements.*

### Operating conditions

During the tests, each the turbine has been operated for a sufficient time span at various conditions from medium to maximum load. For instance, previous investigations carried out in *FT2* by Vizmanos *et al.* [Ref 9] had shown that maximum cavitation erosion was occurring at around full load in this particular machine. Therefore measurements in *FT2* have been carried out at fixed output power conditions corresponding to 7½, 8, 8½, 9, 9½ and 10 MW and for *FT1* the measurements have been taken at 9, 10, 11, 12, 13 and 14 MW.

## Results

The measured signals for the two prototypes have been computer analysed using the same signal processing calculations in both cases. A high frequency band from 3 to 6 kHz has been chosen for the first analysis due to the limitation of the telemetry system cut-off frequency. The overall RMS vibration levels in this narrow band have been calculated for all the signals. Then, the frequency content of the signals has been investigated from auto power spectra with a bandwidth of 20 kHz. And finally, these signals have been filtered in two frequency bands from 3 to 6 kHz and from 17 to 20 kHz in order to determine the main amplitude modulating frequencies.

### Overall levels

The overall RMS vibration levels are presented in Figure 2 as a function of output power for both machines. At first glance, it is observed that for *FT1* the values are more or less constant for all the operating conditions. On the contrary, values for *FT2* present a clear increasing tendency with load. In absolute terms, the vibrations are higher in the turbine guide bearing than in the shaft for both machines. The guide vane vibrations are the highest in *FT1* but are around the lowest in *FT2*. Although in both turbines the shaft vibration levels are around or below 0.5 $m/s^2$, the bearing vibrations levels are quite different with values just above 0.5 $m/s^2$ in *FT1* and ranging from 1 to 2.5 $m/s^2$ in *FT2*.

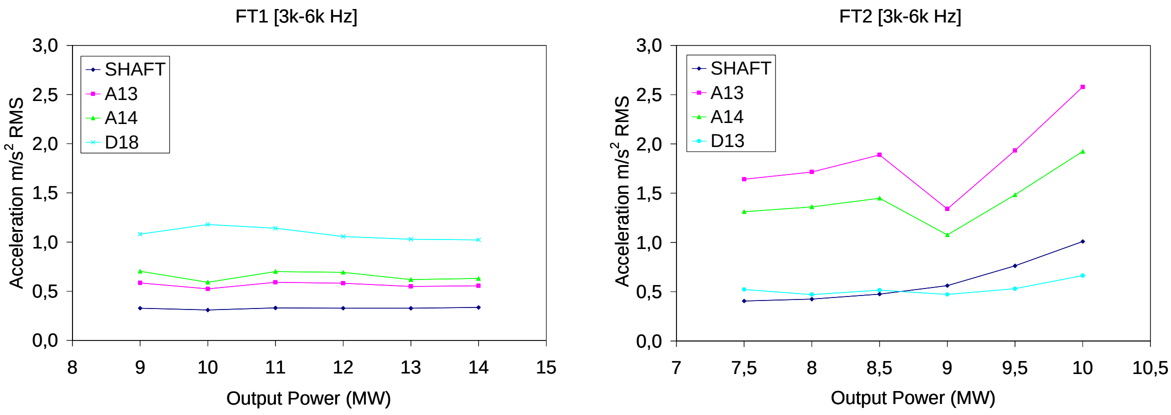


*Figure 2 Overall RMS levels of vibration signals filtered between 3 and 6 kHz as a function of output power for FT1 (left) and FT2 (right).*

## Frequency content

The auto power spectra of shaft vibrations in the frequency band from 1 to 6 kHz are shown on Figure 3. In *FT1* the amplitude of the frequencies along the bandwidth does not change significantly with machine load. Conversely, in *FT2* the shape of the spectra remains the same but the amplitude tends to increase with output load having its maximum at 10 MW. This behaviour begins to predominate above 3 kHz. In Figure 4 the same observations can be made for the vibrations measured in the bearing and reaching in this case the 20 kHz. A wide band excitation is detected by the bearing with the largest amplitude in the band from 5 to 15 kHz that also occurs at higher frequencies. Finally, the vibrations measured in the guide vane can be seen on Figure 5. In this case, completely different spectra are found between the two machines. Again, for *FT1* no significant amplitude change occurs with load and for *FT2* the increasing amplitudes are only observed above 5 kHz and for narrower bands than in the bearing.

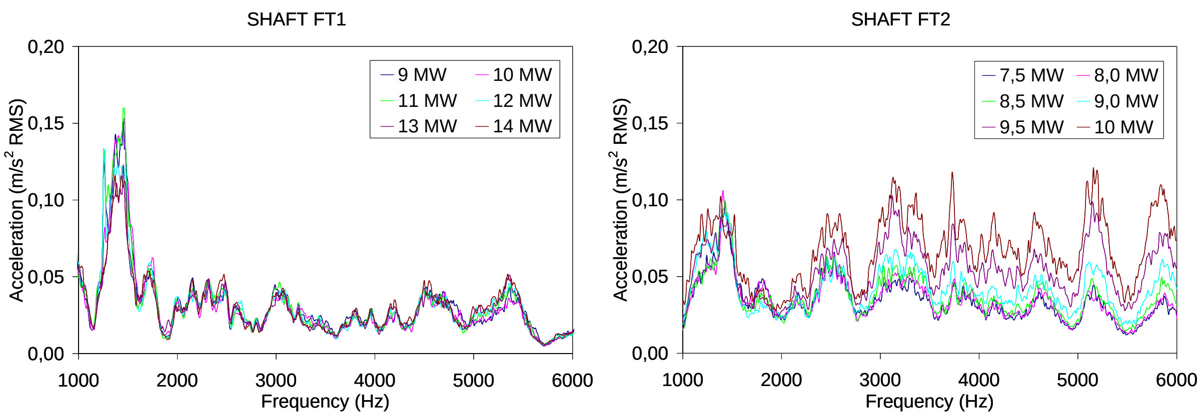


*Figure 3 Auto power spectra from 1 to 6 kHz of shaft vibrations as a function of output power for FT1 (left) and FT2 (right).*

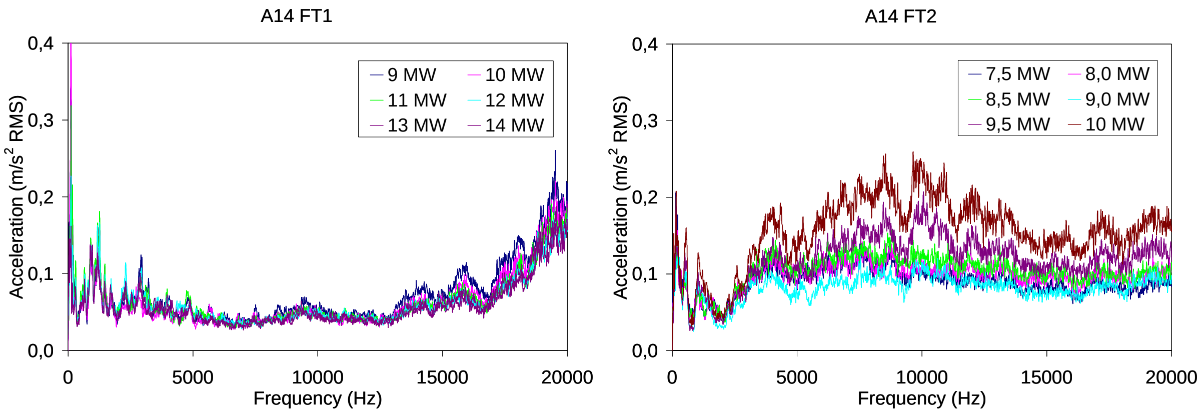


*Figure 4 Auto power spectra up to 20 kHz of bearing vibrations as a function of output power for FT1 (left) and FT2 (right).*

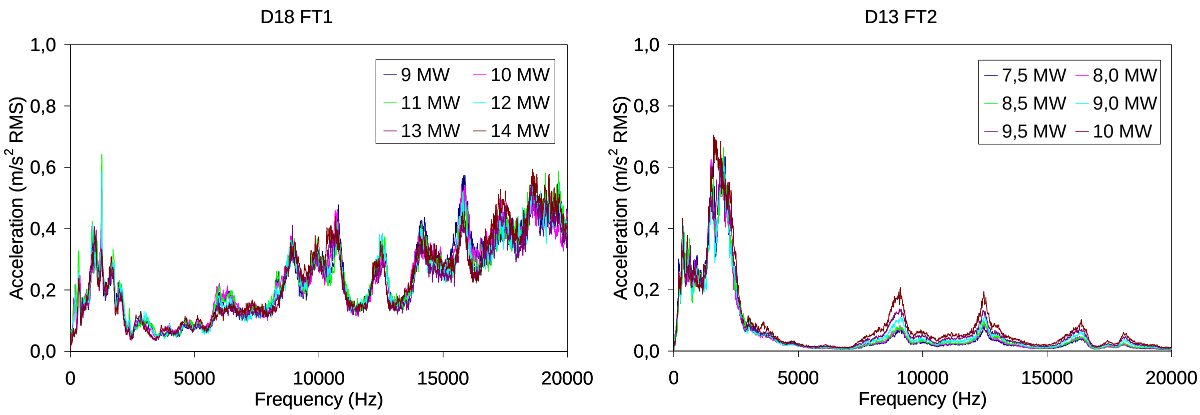


*Figure 5 Auto power spectra up to 20 kHz of guide vane vibrations as a function of output power for FT1 (left) and FT2 (right).*

## Amplitude demodulation

This analysis has consisted in determining the main frequencies that modulate the signals in amplitude by digitally computing the envelopes in certain frequency bands using the Hilbert transform. Now the hydrodynamic frequencies of interest are the fundamental frequency $f_f$ (=$N$/60), the blade passing frequency $f_b$ (=$f_f \times Z_b$), the guide vane passing frequency $f_v$ (=$f_f \times Z_v$) and any of their harmonics. As follows, the results corresponding to the frequency band from 3 to 6 kHz are presented for shaft vibrations in Figure 6, for bearing vibrations in Figure 7 and for guide vane vibrations in Figure 8. In these plots the reduced frequency, $f^*$ (=$f/f_f$), is represented on the abscissas axis to facilitate the identification of the main peaks.

Vibrations on turbine *FT1*, without cavitation erosion, are not modulated neither at $f_b$ nor at $f_v$. Only $f_f$ and its second and third harmonics are observed on the auto power spectra for all the measuring positions. Besides, the amplitude of the frequency peaks and of the entire spectrum do no present any significant change when comparing results in the same point for different operating conditions. As already stated, the amplitude of the demodulated signals is higher in the turbine bearing than in the shaft.

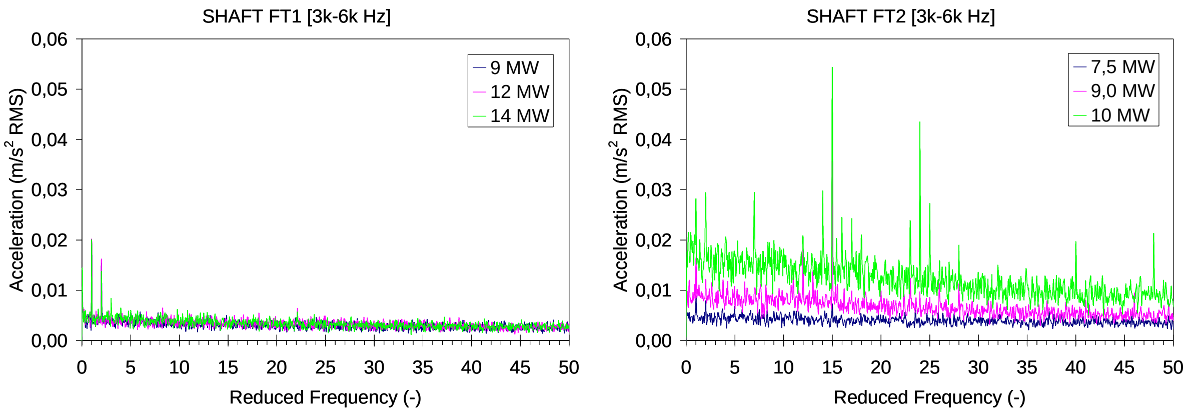


*Figure 6 Auto power spectra of demodulated filtered signals in the band from 3 to 6 kHz for shaft vibrations as a function of output power for FT1 (left) and FT2 (right).*

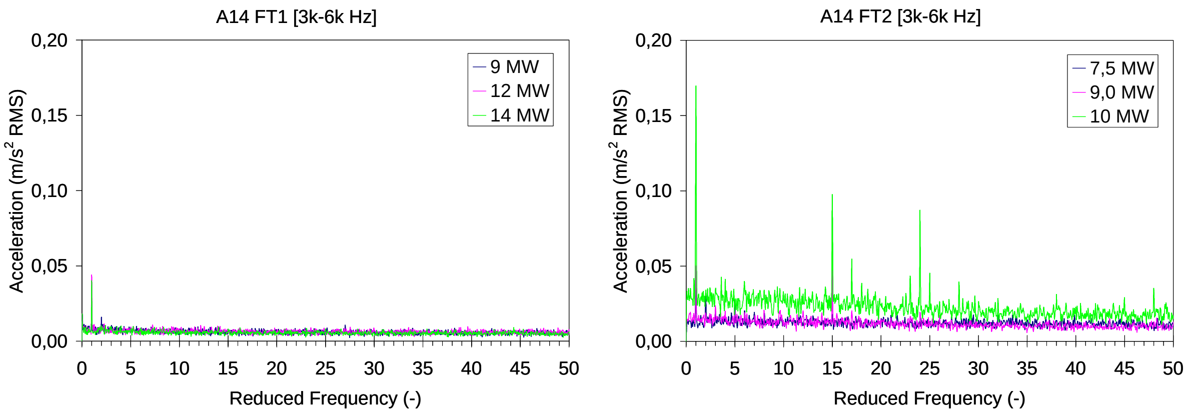


*Figure 7 Auto power spectra of demodulated filtered signals in the band from 3 to 6 kHz for bearing vibrations as a function of output power for FT1 (left) and FT2 (right).*

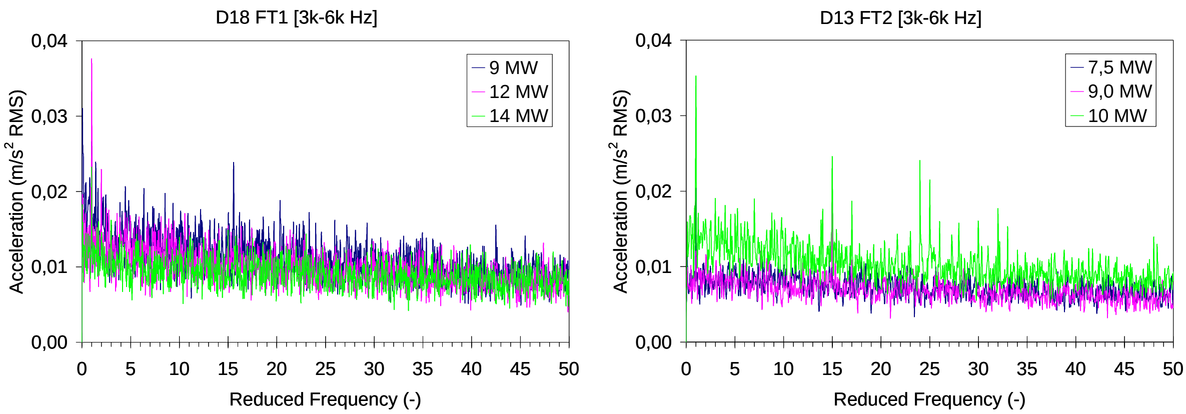


*Figure 8 Auto power spectra of demodulated filtered signals in the band from 3 to 6 kHz for guide vane vibrations as a function of output power for FT1 (left) and FT2 (right).*

The results from the turbine suffering severe cavitation erosion, *FT2*, are more informative than the ones corresponding to *FT1*. In all measured positions, as the machine load increases the presence of the main hydrodynamic frequencies $f_b$ and $f_v$ is more important. For instance, at maximum power output (10 MW) analogous qualitative results are obtained in the shaft and in the bearing. In both cases the main frequency peaks are $f_f$, $f_b$ and $f_v$. In the shaft, both $f_b$ and $f_v$ are modulated at $f_f$. In the bearing the maximum peak corresponds to $f_f$ and the same happens in the guide vane. If a higher frequency band is considered, for instance from 17 to 20 kHz, the $f_f$ peak is reduced and $f_b$ and $f_v$ predominate. For lower loads, these hydrodynamic frequencies tend to decrease and finally disappear. As for *FT1*, the amplitude of the spectra corresponding to the bearing are higher than in the shaft.

## Discussion

The comparison of the results obtained in two hydro turbines using the same experimental set-up and signal processing techniques for cavitation erosion detection permits to demonstrate the validity of the vibratory approach. The increase of the broadband high frequency vibrations indicates the appearance and development of erosive cavitation acting on the runner blades, which does not happen in a turbine exempt of this problem. Furthermore, the cavitation induced vibrations are modulated at hydrodynamic frequencies which are $f_b$ and/or $f_v$. If cavitation does not occur, then no trace of such frequency peaks is found in the vibrations. And finally, it is also proved that an onboard accelerometer mounted on the shaft just above the turbine guide bearing pedestal can be used for detection purposes since analogous qualitative results with the bearing ones are obtained even in a relatively low frequency range from 3 to 6 kHz.

The final goal of an erosive cavitation monitoring system is to quantify the hydrodynamic aggressiveness of the cavitation attack to the blade surface. For that, the use of transmissibility functions between the measuring position and the blades is needed. Up to now, these functions have only been obtained experimentally with the use of calibration tests. The main effort of the researchers has been dedicated to find the transmission from the turbine guide bearing to the blades with an instrumented hammer when the machine is still and empty of water because experiments with the machine in operation are too complex and expensive. Consequently, the bearing fluid film effect on the signal transmission and the fact that the excitation is rotating with the shaft during operation are not considered. So, the resulting transmissibility functions may not be accurate enough to be used to infer the absolute impact forces due to cavitation collapses. Moreover, it is possible that the bearing receives more noise through various transmission paths which is discussed as follows.

Calibration results carried out on a Francis turbine still and empty of water presented in Escaler *et al.* [Ref 10] have shown that an impulsive excitation acting on the blade is transmitted towards the upper part of the shaft, then to the guide bearing pedestal and finally to the guide vane. During its way to the guide vane crossing the bearing, the vibration is attenuated so that in the bearing a lower amplitude and delayed time signal is measured in reference to the shaft. However, the current results obtained with the machines in operation are in contradiction with this observation. In both turbines, *FT1* and *FT2*, the amplitude of the measured vibrations in the frequency band from 3 to 6 kHz is found to be higher in the bearing than in the shaft. These results would prove that in fact the bearing is receiving a certain noise level that could not be assumed to correspond to erosive cavitation activity on the blade surface. From this point of view, the suitability of using shaft vibrations for cavitation erosion prediction instead of bearing ones is reinforced. Additional advantages of such possibility are that the shaft position offers a direct and mechanical link with the excitation source without any fluid film whose transmission properties should not change with machine rotation and that the possible influence of measuring a rotating excitation from a fixed location is avoided when measuring directly in the relative frame of reference.

## Conclusions

Comparative tests have been carried out in two Francis turbines with one of them suffering from cavitation erosion on the runner blades.

A telemetry system with a high frequency accelerometer mounted on the rotating shaft has been successfully used to detect erosive cavitation by means of frequency domain techniques. The detection methodology is the same that has been used up to now for measurements on fixed positions such as turbine bearing pedestal and is based on the high frequency vibration content and the detection of hydrodynamic frequencies, $f_b$ and/or $f_v$, modulating the signals.

For cavitation erosion prediction, the shaft measuring location seems to be advantageous in relation to the bearing because it is more free from noise and the transmissibility function to the runner is easier to characterise with the machine still. However, further investigation is necessary to reinforce this initial assumption.

## References

Ref 1 Abbot, P.A., Morton D.W., Shanahan, T.B., "Hydroturbine Cavitation Detection Using Advanced Acoustic Emissions Techniques", *ASME Hydroacoustic Facilities, Instrumentation and Experimental Techniques*, 1991.

Ref 2 Bajic, B., Keller, A., "Spectrum Normalization Method in Vibro-Acoustical Diagnostic Measurements of Hydroturbine Cavitation", *Journal of Fluids Engineering*, Vol. 118, pp. 756-761, 1996.

Ref 3 Farhat, M., Bourdon, P., Lavigne, P., Simoneau, R., "The Hydrodynamic Aggressiveness of Cavitating Flows in Hydro Turbines", *ASME Fluids Engineering Division Summer Meeting,* June 1997.

Ref 4 Bourdon, P., Simoneau, R., Avellan, F., "Erosion Vibratory Fingerprint of Leading Edge Cavitation of NACA Profile, and of a Francis Model and Prototype Hydroturbine", *Proceedings of the ASME Symposium on Bubble Noise and Cavitation Erosion in Fluid Systems, ASME Winter Annual Meeting,* New Orleans, Louisiana, December 1993.

Ref 5 Escaler, X., Egusquiza, E., Mebarki, T., Avellan, F., Farhat, M., "Cavitation Detection and Erosion Prediction in Hydro Turbines", *Proceedings of the 9th of International Symposium on Transport Phenomena and Dynamics of Rotating Machinery,* Honolulu, Hawaii, February 2002.+

Ref 6 Bourdon, P., Farhat, M., Simoneau, R., Pereira, F., Dupont, Ph., Avellan, F., Dorey, J.M., "Cavitation Erosion Prediction on Francis Turbines Part 1: Measurements on the Prototype", *Proceedings of the 18th IAHR Symposium*, Valencia, 1996.

Ref 7 Bourdon, P., "La Détection Vibratoire de l'Érosion de Cavitation des Turbines Francis", *PhD Thesis*, EPFL, Lausanne, 2000.

Ref 8 Escaler, X., Egusquiza, E., Farhat, M., Avellan, F., "Vibration Cavitation Detection Using Onboard Measurements", *Proceedings of the 5th International Symposium on Cavitation,* Osaka, November 2003.

Ref 9 Vizmanos, C., Egusquiza, E., Jou, E., "Cavitation Detection in a Francis Turbine", *Conference Monitoring for Hydro Powerplants II*, Lausanne, July 1996.

Ref 10 Escaler, X., Egusquiza, E., Mebarki, T., Avellan, F., Farhat, M., 2002, "Field Assessment of Cavitation Detection Methods in Hydropower Plants", *Proceedings of the Hydraulic Machinery and Systems 21st IAHR Symposium*, Lausanne, September 2002.