

FIELD ASSESSMENT OF CAVITATION DETECTION METHODS IN HYDROPOWER PLANTS

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ABSTRACT

An experimental investigation on Francis turbines suffering from inlet cavitation erosion on the runner blades has permitted to assess several aspects of the actual cavitation erosion detection and prediction methods based on vibrations. Detection tests in three prototypes have consisted in the measurement of vibrations in turbine guide bearing, guide vanes and draft tube during operation. Discussion about RMS levels, auto power spectra and amplitude demodulation results is provided. A dynamic calibration has been carried out using an instrumented hammer. For that, one of the accelerometers has been mounted on the blade and the rest have been kept in their external positions. Frequency response functions between the blade and the remote sensors have been obtained and compared among the machines. The various transmission paths have been investigated.

RÉSUMÉ

Une étude expérimentale sur des turbines Francis qui sont exposées à l'érosion de cavitation d'entrée sur les aubes a permis de valider différents aspects des méthodes actuelles de détection et de prédiction de l'érosion de cavitation. Des essais de détection ont été effectués sur trois prototypes afin de mesurer les vibrations sur le palier guide turbine, les directrices mobiles et le tube d'aspiration au cours du fonctionnement. Les niveaux RMS, les auto-spectres et les résultats de la démodulation d'amplitude sont discutés. Un étalonnage dynamique a été réalisé avec un marteau instrumenté. Pour cela, un des accéléromètres a été monté sur l'aube et les autres sont restés sur les positions externes. Les fonctions de réponse en fréquence entre l'aube et les positions externes ont été calculées et comparées entre machines. Les différents chemins de transmission ont été étudiés.

NOMENCLATURE

Term	Symbol	Definition	Term	Symbol	Definition
Blade passing frequency	f_b	$f_b = Z_b \cdot f_f$ [Hz]	Rotating speed	N	[rpm]
Fundamental frequency	f_f	$f_f = N/60$ [Hz]	Number of runner blades	Z_b	[-]
Guide vane passing frequency	f_v	$f_v = Z_v \cdot f_f$ [Hz]	Number of guide vanes	Z_v	[-]

INTRODUCTION

Inlet cavitation affecting runner blades can produce severe erosion, which is a serious problem for operators of hydropower plants. To increase further the time between repairs, the operation condition at which the erosion is maximum should be avoided. For this, a reliable method to detect cavitation erosion is necessary. Up to now, several methods have been proposed to detect and predict cavitation erosion on prototypes, all of them based on vibrations. For instance, the latest works of authors like Farhat *et al.* (Ref. 1), Bourdon (Ref.

2) and Kaye (Ref. 3) have been devoted to the application of these methods to actual water turbines with satisfactory results. Basically, the main efforts have been dedicated to determine the most adequate measuring positions, the best detection techniques and the transmissibility functions to the eroded areas. This information is necessary to be able to quantify the cavitation intensity in absolute terms on the material under attack. Thus, cavitation erosion can be predicted if the type of material and its resistance are considered. To assess the feasibility of the present detection and prediction methods, experimental tests have been conducted on three prototypes suffering from leading edge cavitation erosion.

PROTOTYPES AND MEASURING CAMPAIGN

Three vertical Francis turbines have been chosen for the experimental investigation. Two of them, named as *HT1* and *HT2*, belong to hydropower plant *A* and are identical with the exception of the runner. Unit *HT2* keeps the original runner, made of cast iron, but unit *HT1* has recently been equipped with a new one made of stainless steel and with an improved design. These turbines operate up to a maximum output power of 65 MW per unit. The total nominal flow rate is 115 m³/s and the net head is around 122.5 m. In both units, *N* is 250 rpm, *Z_b* is 15 and *Z_v* is 24. The third prototype, *HT3*, is located in a different hydropower plant *B* that has already been described in Escaler *et al.* (Ref. 4) and Vizmanos *et al.* (Ref. 5). This unit is smaller than the former, with a maximum power of about 11 MW. In addition, its turbine bearing is made of rubber and works without oil film.

Cavitation erosion is well localized next to the leading edge in the suction side of the blades in the three runners. The picture in Fig. 1 shows the damage appeared after several years of operation in a spare runner. Advanced stages of mass loss occur due to inlet cavitation.

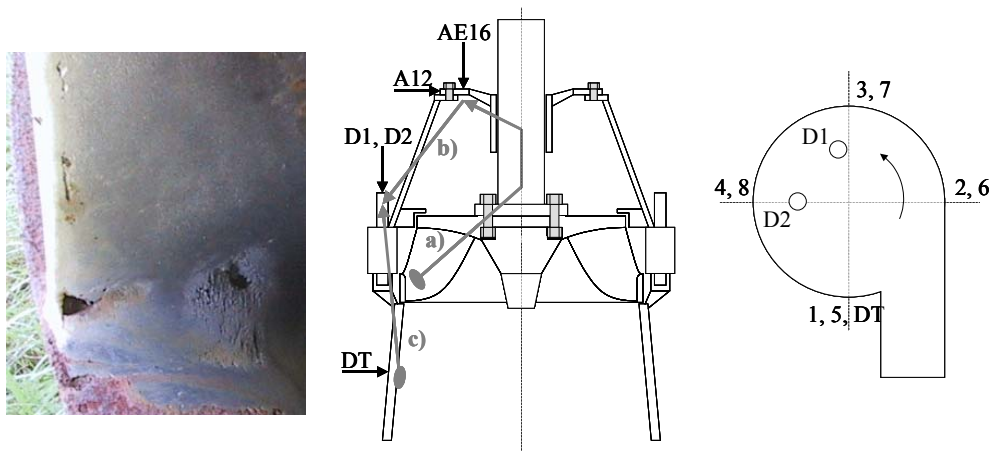


Fig. 1 Left) Picture of spare runner showing advanced cavitation. Center and right) Outline of the turbine with the measuring positions.

The instrumentation has comprised 4 miniature high frequency accelerometers with a mounted resonance frequency of about 52 kHz which have been fixed with cementing studs. For acoustic emission detection, a transducer with a resonance frequency of about 200kHz has been used. Contact-free detection of the rotating shaft has been achieved with a photoelectric tachometer probe. Moreover, the calibration tests have been carried out with an impact hammer with steel tip and having a frequency range up to 10kHz. The signals from all these sensors have been conditioned and amplified prior to their recording. Finally, the analogue outputs have been sampled simultaneously at a frequency rate of 100kHz with a 12-bit 1MHz A/D data acquisition system.

An outline of a transverse section for turbines *HT1* and *HT2* and the measured positions during operation is shown in Fig. 1. Position A12 corresponds to an accelerometer located in the turbine guide bearing at 90° from the penstock with radial orientation. Position AE16 indicates an acoustic emission sensor with axial orientation. The accelerometers located on the top of the wicket gate central arms have been named as D1 and D2. Finally, the position DT indicates the accelerometer mounted on the draft tube wall. It is important to note that the sensors have been mounted in the same positions for both units *HT1* and *HT2*. In unit *HT3*, only position A13 has been measured during operation.

For the dynamic calibration, the runner has been accessed from the draft tube man-door. The accelerometer DT has been mounted on one blade next to the eroded region and the accelerometer located in D2 has also been moved to the shaft. The photograph on the left of Fig. 2 shows cavitation erosion in the runner blades of unit *HT2*. On the right, the accelerometer mounted on the blade and the impact hammer are shown.

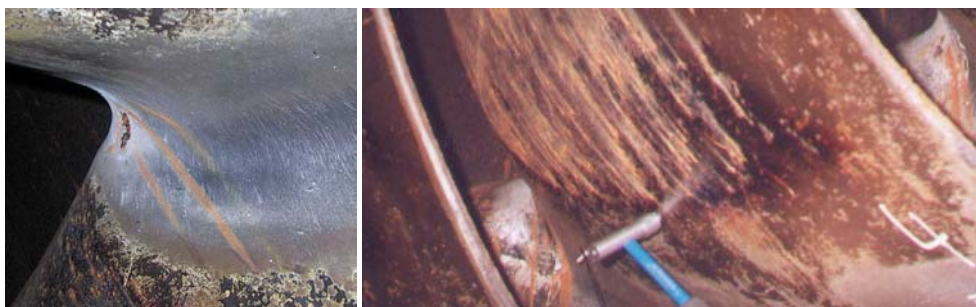


Fig. 2 Left) Cavitation erosion in unit *HT2* observed during calibration tests. Right) Accelerometer mounted next to the erosion and impact hammer.

Unit *HT2* has been operated at 20, 25, 30, 35, 40, 45, 50, 55, 60 and 65 MW. On the other hand, unit *HT1* has only been operated at 25, 35, 45, 55 and 65 MW. Unit *HT3* could only be measured during operation at 100% guide vane opening.

RESULTS

Detection technique

To begin with, the average RMS values of accelerations and acoustic emission are presented for unit *HT2* in Fig. 3. Data computed from 1 to 49k Hz and from 25k to 49k Hz are plotted on the left and on the right hand side, respectively, as a function of output power. The vibration levels follow similar trends in the turbine guide bearing and the guide vanes. However, vibrations measured in the guide vanes are much higher than in the turbine guide bearing. The maximum levels are detected at 30 and 60 MW. On the other hand, the measuring position in the draft tube shows a different behaviour.

The autopower spectra in any position except DT show that, when the load is changed, the entire frequency band is increased or decreased. Nevertheless, there are wideband regions in the high frequency range where the change is larger. This is particularly clear for sensors A12 and AE16. Some results are plotted on the left of Fig. 4 corresponding to positions D2 and A12 in units *HT1* and *HT2*. It is observed that A12 presents maximum amplitudes around 40kHz in unit *HT2*, meanwhile they occur around 30kHz in unit *HT1*. Furthermore, the spectrum for A12 at 30MW is slightly higher than the one at 60MW in unit *HT2*. In order to determine the origin of the vibrations, the signals have been band pass filtered between 30k and 40k Hz in order to compute the envelope using the Hilbert transform. As a result, the envelope's frequency content reveals well defined peaks at certain hydrodynamic frequencies.

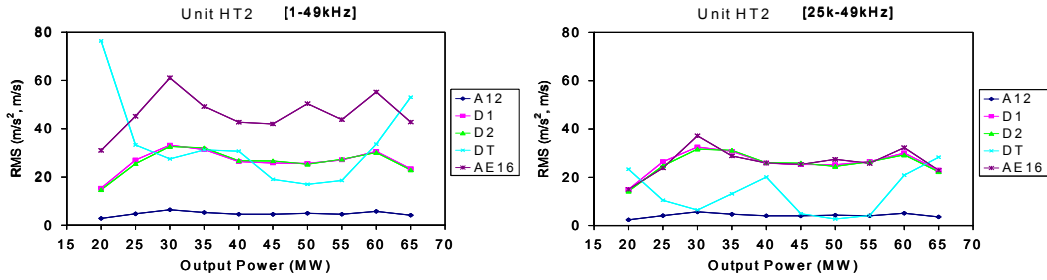


Fig. 3 RMS of accelerations and acoustic emission for [1-49k] and [25k-49k]Hz frequency bands measured in unit HT2 at various operating conditions.

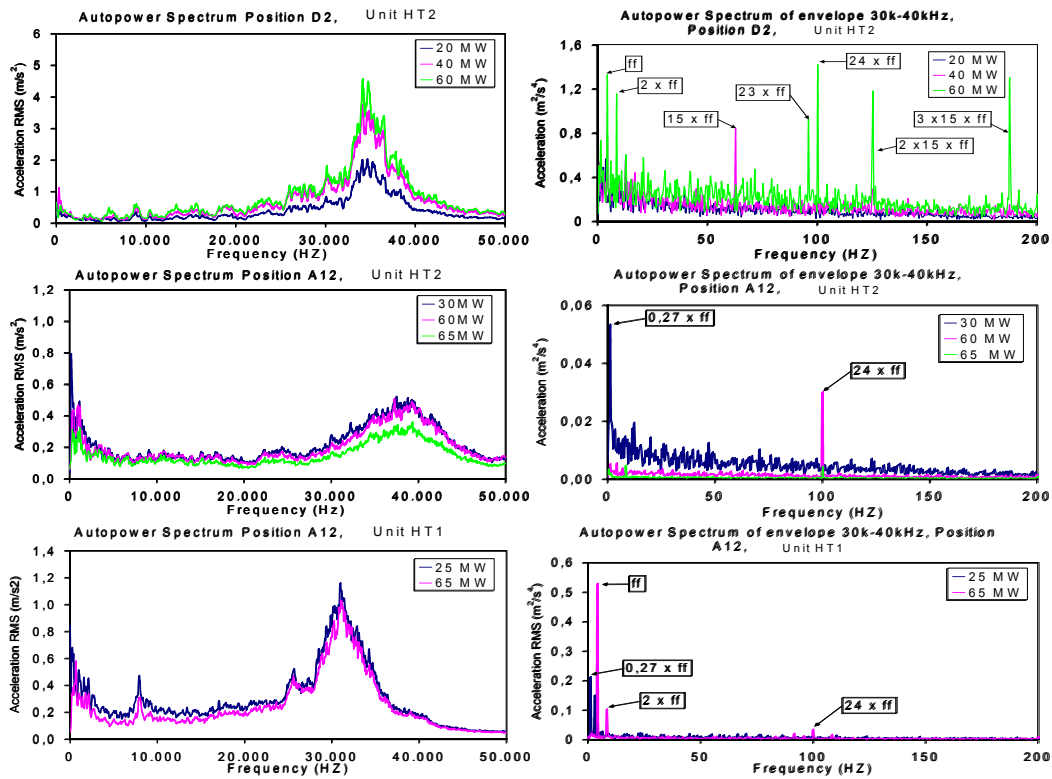


Fig. 4: Left) Autopower spectra of vibrations measured in position D2 in unit HT2 and position A12 in units HT1 and HT2. Right) Autopower spectra of vibration envelope in the frequency band from 30k to 40kHz for the same positions.

The results plotted on the right of Fig. 4 for position A12 in unit HT2 indicate that the vibration in the band from 30k to 40k Hz is mainly modulated at $0,27 f_f$ when the machine operates at low loads. On the contrary, the frequency peak that predominates at higher loads is $f_v (=24 f_f)$. The maximum amplitude for $0,27 f_f$ occurs at 30MW and for f_v at 60 MW. The AE16 sensor leads exactly to the same results. At 30 MW, the base level of the spectrum is highly increased as if strong turbulence were associated to this operating condition. If the guide vane sensor is considered, the frequency peak at $f_b (=15 f_f)$, some harmonics and f_f are also important.

Calibration

The tests have been carried out with the machines still and empty of water. The blade next to A12 has been impacted and the induced vibrations have been recorded in the same blade, in the shaft just in front of A12, in A12 itself and in D1. From Fig. 5, it can be stated that the initial excitation at the blade is transmitted towards the upper part of the shaft, then to the

guide bearing and finally to the guide vane. During its way up, the vibration is attenuated reaching the guide vane with a very low amplitude and delayed in time. The blade vibrates up to about 300 m/s^2 , the shaft reaches 10 m/s^2 , the bearing goes up to 5 m/s^2 and finally in the guide vane the vibration does not surpass 1 m/s^2 . Moreover, average values of time delays between measured responses have been calculated taking as a reference the first vibration on the blade, which are listed in Table 1. For instance in unit *HT1*, the accelerometer in the shaft begins to detect some vibration $0.87 \cdot 10^{-3} \text{ s}$ after the blade, the accelerometer in the guide bearing after $1 \cdot 10^{-3} \text{ s}$ and the one in the wicket gate after $3.7 \cdot 10^{-3} \text{ s}$.

In Fig. 6 the averaged frequency response function between the accelerometer on the blade and the accelerometer on the turbine guide bearing is shown, as well as the coherence function.

Table 1 Time delays from shock response measured on the blade to other measuring positions in prototypes *HT1* and *HT2*.

Time (seconds)	Unit <i>HT1</i>	Unit <i>HT2</i>
Shaft	$0.873 \cdot 10^{-3}$	$0.825 \cdot 10^{-3}$
Bearing A12	$1.003 \cdot 10^{-3}$	$0.875 \cdot 10^{-3}$
Guide vane D1	$3.668 \cdot 10^{-3}$	$1.425 \cdot 10^{-3}$

The reciprocal frequency response functions between blade and bearing are shown in Fig. 7. The blade to bearing function has been obtained impacting next to the sensor on the blade and considering its signal as the excitation and the signal from the bearing as the response. The reciprocal function has been obtained impacting next to the sensor on the bearing and considering its signal as the excitation and the response on the blade.

As drawn in Fig. 1, there is a mechanical connection between the runner and the shaft. Hence, the excitation coming from the blade reaches the shaft and crosses to the bearing pedestal. Therefore, the total transmissibility function from blade to bearing pedestal is due to the path from blade to shaft and from shaft to bearing. When comparing unit *HT2* against *HT3* in Fig. 8, sharp differences are found for the functions from blade to bearing but results are similar between the functions from blade to shaft.

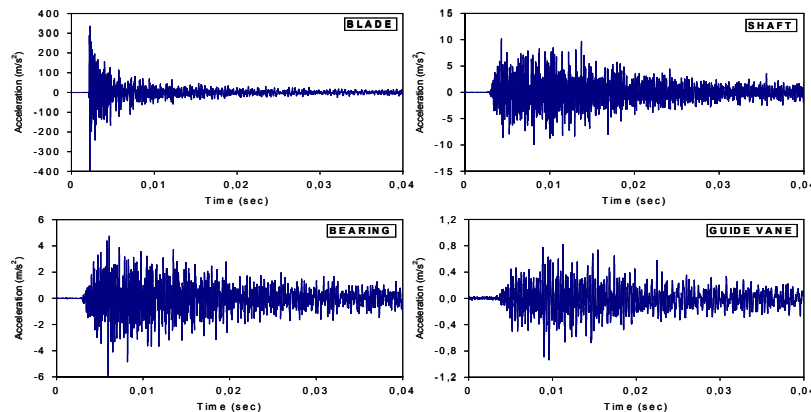


Fig. 5 Vibrations in blade, shaft, bearing and guide vane induced by a hammer blow on blade of unit *HT2*.

Discussion

Overall vibration levels in turbine guide bearing and wicket gates permit to find out the operating conditions at which maximum high frequency activity occurs inside the runner. This simple analysis provides an initial indication that some type of cavitation might exist. In opposition, the results from the draft tube does not follow the same trends.

The comparison of vibration signatures shows that the energy content changes in the entire frequency band with machine load, except in the draft tube. In particular, the high frequency region is very sensitive to the operating conditions. The maximum amplitudes of the spectra depend on the measuring position and their shape is different, as pointed out by Bajic and Keller in Ref. 6. Therefore, it is not possible beforehand to select the better band for detecting cavitation in a given sensor location. In addition, the vibration signatures measured with the same transducer in different machines are also different.

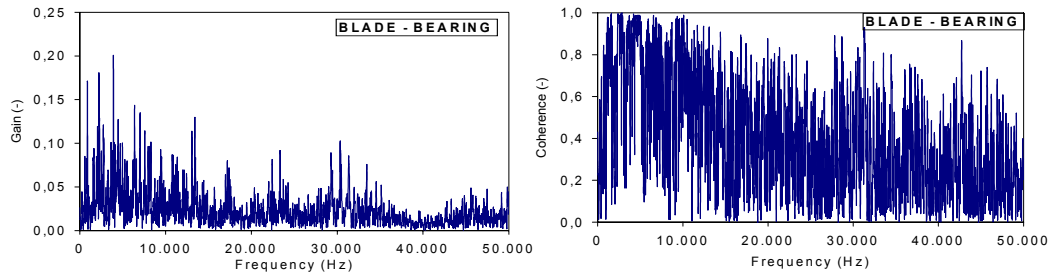


Fig. 6 Frequency response function and coherence between the accelerometer on the excited blade and the response on the guide bearing in unit HT2.

Amplitude demodulation techniques are proved to be very useful to determine the nature of the excitation. For instance in unit HT2 at 30 MW, the frequency peak at $0,27f_v$ suggests the presence of a partial load hub rope in the draft tube. On the other hand at 60 MW, the peak at f_v indicates that erosive inlet cavitation occurs on the blades. These frequencies are very well detected in the turbine guide bearing by the accelerometer and the acoustic emission sensor. Both sensors can be used to determine the exact operation condition at which the maximum cavitation aggressiveness occurs by detecting the maximum amplitude reached by the f_v peak as a function of load. The amplitude of f_v depends on measuring position and frequency band selected. Therefore, this parameter by itself is not a good indicator of the absolute cavitation intensity on the blades. Moreover, when comparing two machines with the same type of erosion, f_v is the hydrodynamic process that modulates the cavitation behaviour in unit HT2, meanwhile in HT3, it is the f_b . Bourdon (Ref. 2) has pointed out the design of the spiral casing as a reason to explain such discrepancy.

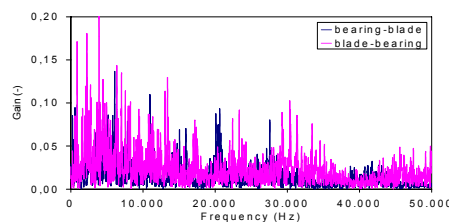


Fig. 7 Reciprocal frequency response function between blade and bearing in unit HT2.

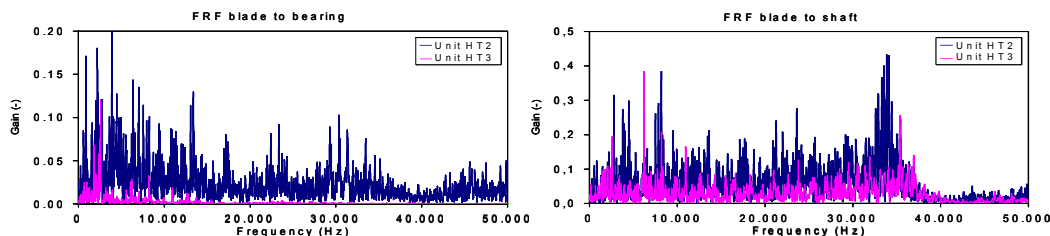


Fig. 8 Comparison of frequency response functions blade to bearing and blade to shaft between units HT2 and HT3.

The study of the transmissibility indicates that an excitation taking place on the blade wall goes up to the shaft, then crosses to the guide bearing and finally arrives to the guide vane

following the paths a) and b) indicated in Fig. 1. During its way the energy is progressively dissipated and the farther the measuring location the lower the vibration amplitude. Since during operation the vibration levels on guide vane are much higher than on turbine guide bearing, it indicates that the guide vane possibly detects non erosive cavitation and other sources of excitation through various transmission paths. One possibility is that some noise comes from the draft tube by the path c) in Fig. 1. Secondly, the wicket gates are directly wetted by the main flow and they are very close to the blade leading edges. The incoming water can act as a transmission media, as already pointed out by several authors, of cavitation noise generated by cavity collapses not occurring on the solid wall or next to it but in the main flow, which are not erosive. As a matter of fact the guide vane sensor detects higher vibration levels and additional modulating frequencies.

The coherence between the remote sensors and the one mounted on the blade is above 0.8 up to 15 kHz and decreases for higher frequencies. The hammer mass and hardness of the tip are the limiting factors to improve the transmissibility.

Based on the previous results, the principle of reciprocity (see Bourdon in Ref. 2) has been confirmed between the blade and the bearing in the prototypes under study since the reciprocal functions have similar amplitudes.

From the point of view of establishing a cavitation intensity quantification method by the use of frequency response functions, it would be very useful to assume that analogous functions are obtained for prototypes with similar characteristics. Unfortunately, the transmission from blade to bearing is found to be different between unit *HT2* and *HT3*. Nevertheless, the blade to shaft transmissibility is similar between both machines. Therefore, the differences are mainly due to the shaft to bearing contact.

CONCLUSIONS

The guide bearing pedestal has been the best measuring point for both detection and quantification of erosive inlet cavitation in the runner blades. Both high frequency accelerometers and acoustic emission sensors have been good transducers for detection purposes and provide similar results.

The best method to identify the hydrodynamic frequencies governing the cavity fluctuation has been the use of amplitude demodulation techniques applied to vibration signals filtered in wide frequency bands in the high frequency range. Operation conditions at which maximum erosive cavitation intensity takes place have been easily pinpointed by tracking the amplitude of f_v or f_b over the total series of machine loads.

The amplitude of f_v or f_b peak does not seem to be sufficient to quantify in absolute terms the erosive forces on the blades if the transmission properties are not considered, since this is very dependent on sensor used, measuring position, selected frequency band and particular prototype.

Impulsive forces on the blades are transmitted through the shaft and guide bearing, reaching the guide vane sensor with very low amplitude. Therefore, this confirms that during operation little noise due to erosive inlet cavitation should be felt by this sensor. On the other hand, it detects very well the collapses taking place in the main flow at the same time than other sources of noise. Therefore, guide vanes are a good detection location but they are not a good quantification location.

Frequency response functions with a good coherence level have been experimentally obtained using a medium instrumented hammer. In the prototypes under consideration, the principle of reciprocity appears to apply between the blade and the bearing.

The frequency response functions from blade to bearing with the machine still and empty of water are different between different machines, but the functions between the blade and the

region of the shaft next to the bearing appear to have similar gain values. This seems to indicate that this type of test might not be adequate to correctly determine the transmissibility properties between the shaft and the bearing when an oil film is formed.

ACKNOWLEDGEMENT

The authors would like to acknowledge ENDESA GENERACIÓN and the CICYT (DPI2000-0702) for their financial support and collaboration.

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