ROTOR RESPONSE ANALYSIS OF HYDROELECTRIC MACHINES

Luiz de Paula do Nascimento
Faculdade de Engenharia - UNESP – Campus de Ilha Solteira
Departamento de Engenharia Mecânica
Av. Brasil Centro, 56 - CEP 15.385-000 - Ilha Solteira - SP
E-mail: depaula@dem.feis.unesp.br

Abstract. *In this work the results of an experimental modal analysis carried out in a hydroelectric machine will be presented. The generating unit has nominal power of 15 MW and rotating speed of 375 rpm, and operates with a vertical turbine of Francis type. The natural frequencies, vibration modes and modal damping factors have been identified through blows onto the machine using an impact hammer. The results were obtained from the frequency response functions employing the LMS modal analysis package. In addition, the analysis in time domain has also been accomplished. The analysis has been carried out with still machine and with machine in operating condition. Hence, it has been possible verifying the change in the modal behavior of the machine, which happens due to certain dynamic effects that appear only when the machine is working, i.e., effect of the added mass of water contained by the turbine, effect of the oil film in the bearings, magnetic stiffness effect in the generator, seal effect at the labyrinths of the runner and the gyroscopic effect. In the analysis with machine in operating condition it has been used a signal processing technique that allows to attenuate the vibration levels at rotating frequency and its harmonics and to emphasize the resonance frequencies. An important contribution of this work is obtaining data respect to the damping of hydroelectric machinery, since little information about this subject has been found in the literature.*

Keywords: *modal analysis, hydroelectric machines, rotor-dynamic.*

1. INTRODUCTION

In general, the hydroelectric groups are big and not very accessible machines with complex vibratory behavior because they are exposed to dynamic exciting forces of mechanical, electrical and hydraulic origin. In view of that, a lot of developed works trying to analyze the vibratory behavior of those machines has been limited to theoretical researches, accomplished through simulations using mathematical models (Simon, 1982; Utecht, 1983; Childs, 1993). However, the models should be developed with a desirable accuracy, and for that, it is necessary to minimize the uncertainties of the dynamic parameters, which should be adjusted with base on experimental data, Nascimento e Egusquiza (1996). In these systems, among the modal parameters, the amount of damping acting on the bearings, turbine and another parts of the machine is not very well-known, Nascimento e Egusquiza (1997). The damping is important to attenuate the vibrations especially at resonant conditions. The determination of the damped natural frequencies and the respective modal damping factors has great interest when it is desirable the application of methods to establish the degree of dynamic stability of the systems. Hence, for prediction of the vibratory response when the machine is exposed to the several exciting forces, a correct mathematical discretization of the system and an exact determination of the damping are requested.

For monitoring and fault diagnosis of the machines it is very important to know previously the natural frequencies and to verify if there are amplified responses due to resonance. Furthermore, the identification of the vibration mode shapes is important to give an idea of the relative modal displacements that occur at several points of the shaft and where these displacements happen with higher magnitudes.

It is important to remind that this work is not intended to present insights about modal analysis itself, but just to show the procedures so that one can obtain modal data of that machine type using the conventional techniques.

2. MODAL RESPONSE IN TIME DOMAIN

The vibratory motion of a linear undamped system can be given by the equation,

$$[M]\{\ddot{y}\}+[K]\{y\}=\{F\} \tag{1}$$

where $[M]$ and $[K]$ are the mass and stiffness matrices and $(\ddot{y})$ e $\{y\}$ are the acceleration and displacement vectors, respectively, and $\{F\}$ is the external forces vector. The modal characteristics of the system are obtained by the homogeneous solution of Eq. (1), i.e., in taking $\{F\} = 0$. Admitting a harmonic solution for the homogeneous equation as,

$$\{y\}=\{u\}e^{\lambda t} \tag{2}$$

the equation of the eigenvalue problem can be given by,

$$([M]\lambda^2+[K])\{u\}=0 \tag{3}$$

The solution of the Eq. (3), for a system with n degree of freedom, will produce n eigenvalues as:

$$\begin{aligned} &\lambda_1 = i\omega_{n1} \ , \ \lambda_2 = i\omega_{n2} \ , \ \ldots , \lambda_n = i\omega_{nn} \\ &\lambda_{n+1} = -i\omega_{n1} \ , \lambda_{n+2} = -i\omega_{n2} \ , \ \ldots , \lambda_{2n} = -i\omega_{nn} \end{aligned} \tag{4}$$

where, ω_{nk} (k = 1, 2, ..., n) are the undamped natural frequencies of the system given in rad/s.

For each solution λ_k, the Eq. (3) will produce a real vector $\{u\}^k$ (eigenvector), which define the vibration mode shape of the system at ω_{nk} frequency. The displacements of a mode shape can be described by the equation,

$$\{y_k(t)\} = A_k u^k \cos(\omega_{nk}t - \varphi_k) \tag{5}$$

where A_k and φ_k can be obtained by the initial conditions of the motion. The general homogenous solution of the Eq. (1) can be given by the superposition of the motion of all vibration modes, i.e.,

$$\{y(t)\} = \sum_{k=1}^{n} A_k u^k \cos(\omega_{nk}t - \varphi_k) \tag{6}$$

In real systems, the condition of undamped doesn't correspond to the reality. Therefore, some general aspects when the damping is introduced in the equation of the motion will be stood out now.

Considering a viscous damping acting on the system, the homogeneous part of the equation of motion can be written as,

$$[M]\{\ddot{y}\}+[C]\{\dot{y}\}+[K]\{y\}=\{0\} \tag{7}$$

where $[C]$ is the damping matrix and $\{\dot{y}\}$ is the velocities vector. In this case, for solution of the eignvalue problem it is convenient taking a new vector of double order $2n$ to describe the motion of the system, coupling the displacement and velocity. Thus, the equation becomes,

$$\begin{bmatrix} [C] & [M] \\ [M] & [0] \end{bmatrix}\begin{Bmatrix} \{\dot{y}\} \\ \{\ddot{y}\} \end{Bmatrix}+\begin{bmatrix} [K] & [0] \\ [0] & [-M] \end{bmatrix}\begin{Bmatrix} \{y\} \\ \{\dot{y}\} \end{Bmatrix}=0 \tag{8}$$

As in the undamped system, admitting for Eq (8) a harmonic solution, according to Eq (2), the follow equation can be obtained,

$$\left([\widetilde{M}]\lambda+[\widetilde{K}]\right)\{u\}=0 \tag{9}$$

which will produce the eigenvalues,

$$\lambda_1=-\upsilon_1+i\omega_{d1}\,,\ \lambda_2=-\upsilon_2+i\omega_{d2}\,,\ \ldots,\ \lambda_n=-\upsilon_n+i\omega_{dn}$$
$$\lambda_{n+1}=-\upsilon_1-i\omega_{d1}\,,\ \lambda_{n+2}=-\upsilon_2-i\omega_{d2}\,,\ \ldots,\ \lambda_{n+n}=-\upsilon_n-i\omega_{dn} \qquad 10)$$

where ω_{dk} (k = 1, 2, ...,n) are the damped natural frequencies and the ν_k express the amount of modal damping. Introducing these eigenvalues into Eq. (9), it can be obtained the eigenvectors $\{u\}^k=\{u\}_R^k+i\{u\}_I^k$, which are complex vectors with *2n* order.

It can be demonstrated that the vibratory motion at a particular natural frequency ω_{dk} can be expressed by the following equation, Ohashi (1991),

$$\{y_k(t)\}=A_k e^{-\upsilon_k t}\left[u_R^k\cos\left(\omega_{dk}t-\varphi_k\right)-u_I^k\,\mathrm{sen}\left(\omega_{dk}t-\varphi_k\right)\right] \tag{11}$$

and the result of the superposition of the motion of all vibration modes will be,

$$\{y(t)\}=\sum_{k=1}^{n}\{y_k(t)\} \tag{12}$$

If the damping is introduced in the free vibration equation of the system, the eigenvectors $\{u\}^k$ will be complex and the maximum relative amplitude of each coordinate doesn't happen simultaneously, making the geometric visualization of the vibration modes so much difficult. Furthermore, the damped natural frequencies (ω_{dk}) have values lightly different than the undamped natural frequencies (ω_{nk}) in function of the amount of damping of the system. For systems with many degrees of freedom it is very difficult to establish an algebraic relationship among these frequencies.

From Eq. (5) it can be observed that if the damping is not regarded, the motion of the modes will be harmonic with constant displacements with the time. Nevertheless, in Eq. (11), corresponding to the damped system, the term $e^{-\nu t}$ is an exponential function ("envelope ") defined by the damping coefficient, which induces a reduction of the displacements with the time. The ν coefficient defines a measure of the modal damping in the time scale. The usual way to express the modal damping is through a non-dimensional factor (ξ) defined by,

$$\xi_k = \upsilon_k \omega_{nk} \tag{13}$$

The experimental damping factors can be obtained from the modal response after a blow onto the system. The exponential "envelope" will define the reduction of the displacement at a certain time or at a certain number of cycles of the motion. The modal damping factors can be obtained from the logarithmic decrement (δ_k) through equation,

$$\xi_k = \frac{\delta_k}{2\pi}\frac{\omega_{dk}}{\omega_{nk}} \tag{14}$$

The relationship among the frequencies (ω_{dk}/ω_{nk}) can be regarded equal to 1 when the damping of the system is very small. In general, this consideration should be assumed to determine the first approach of the experimental values of ξ_k since it is not possible obtaining the undamped natural frequencies of the system experimentally.

3. FREQUENCY RESPONSE FUNCTION

In an experimental modal analysis, the modal parameters of the machine can also be obtained through frequency response functions (FRF). They are functions in the frequency domain that express the relationship between a response ("output") and an exciting force ("input"). These functions will always depend on the position where the load will be applied and of the position where the response will be obtained, as it is shown in Fig. (1).

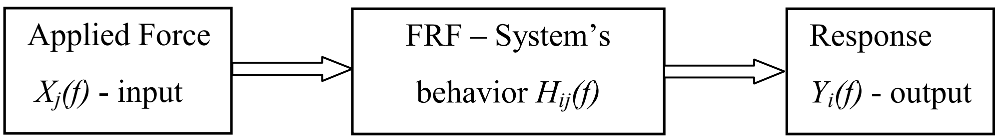

Figure 1. Frequency response function illustration

The frequency response function $H_{ij}(f)$ can be determined by the equation,

$$H_{ij}(f) = \frac{Y_i(f)}{X_j(f)} \tag{15}$$

where $X_j(f)$ e $Y_i(f)$ are the Fourier transform of signals originally recorded. In accordance with Bendat e Piersol (1980), the frequency response function can also be obtained in terms of spectral densities. Thus, considering the input and output signals with n samples, the frequency response function can write as,

$$H_{ij}(f) = \frac{G_{ij}(f)}{G_{jj}(f)} = \frac{\sum_{k=1}^{n} Y_{i(k)} X^*_{j(k)}}{\sum_{k=1}^{n} X_{j(k)} X^*_{j(k)}} \tag{16}$$

where $G_{ij}(f)$ is the cross-spectral density between the input and output signals, and $G_{jj}(f)$ is the spectral density of the input signal. The symbol (*) denotes the conjugate complex. The cross-spectral density has the phase information between the input and output signals.

4. EXPERIMENTAL PROCEDURES

Since the hydroelectric machines are big and very rigid systems, the more practice way of accomplishing modal analysis is through blows with impact hammer. In another manner, as exciting the machine with an electro-mechanical device (shaker), it would demand a powerful device and a sophisticated control system. Hence, the cost of installation of a shaker is very high and can be a very complicated task, demanding special adaptations (rigid structures to support the shaker) and a large period of time for installation.

The modal analysis presented in this work has been accomplished in a generating unit operating with a Francis type turbine of vertical shaft with three guide bearings and one thrust bearing. The nominal power of the machine is 15 MW with a rotating speed of 375 rpm. In the analysis with the still machine eight aligned accelerometers were placed on the shaft. With the machine running one accelerometer was located on each guide bearing. Several blows at different points of the shaft were done with a Dytran 5803A impact hammer of nearly 12 Kg. The signals were stored in a FM Racal V-Store tape record with 24 channels and later analyzed in the laboratory with a HP 35655A signal analyzer of several channels.

The data were analyzed at relatively low frequency range, from 0 to 200 Hz, where the most important natural frequencies of this type of machine should be found, and have high probability to be excited. The frequency response functions have been obtained through Eq. (16), as a result of an average of several impacts. To determine the modal characteristics of the machine, the frequency response functions from the signal analyzer were later processed through LMS modal analysis package. Furthermore, some results have been obtained from impact responses in the time domain using the logarithmic decrement.

5. EXPERIMENTAL RESULTS

The analysis has been accomplished with the still machine and with the machine in operation. In the first condition, it was possible to install sensors (accelerometers) along the shaft as already mentioned, resulting the more accurate analysis and with larger number of data. Nevertheless, although being a complicated task, it is more important to obtain the modal data of the machine in operation, corresponding to the real condition. When the machine is in operation, certain dynamic effects appear, which can modify the modal behavior. The effect of the added mass, the stiffness, damping and seal effect in the turbine emerge. In the generator, the effect of magnetic stiffness between the rotor and stator appears. Whit the rotation the gyroscopic effect also appears. Furthermore, the stiffness and damping characteristics of the oil film of the bearings are different when the machine is rotating. Moreover, the modal behavior can also vary depending on the different operation conditions (partial loads). Additional details about of these effects can be seen in Nascimento (1997). With operating machine the sensors were placed on the casing of the bearings, and in this condition the impacts onto the shaft must be very strong and carefully applied to avoid accidents.

5.1. Machine Still

Figure (2) shows the frequency response function and the respective phase curve between the exciting force (input) and response (output) obtained at a point close to the turbine. It can be seen clearly the first natural frequencies of the machine (21.5, 35, 47.5, 75 and 145 Hz), which are confirmed by the phase diagram.

The frequency response functions obtained at different points of the shaft and processed by the modal analysis package LMS produced the results presented in Fig. (3), which shows the second, third and fourth vibration modes of the machine and the respective natural frequencies (40.8, 48.9 and 77.5 Hz) and damping factors (8.55, 4.25 and 3.34%). In this case, it was not possible to obtain the first vibration mode due to its high damping factor. It is also observed that the damping of the second mode is significantly higher than the others.

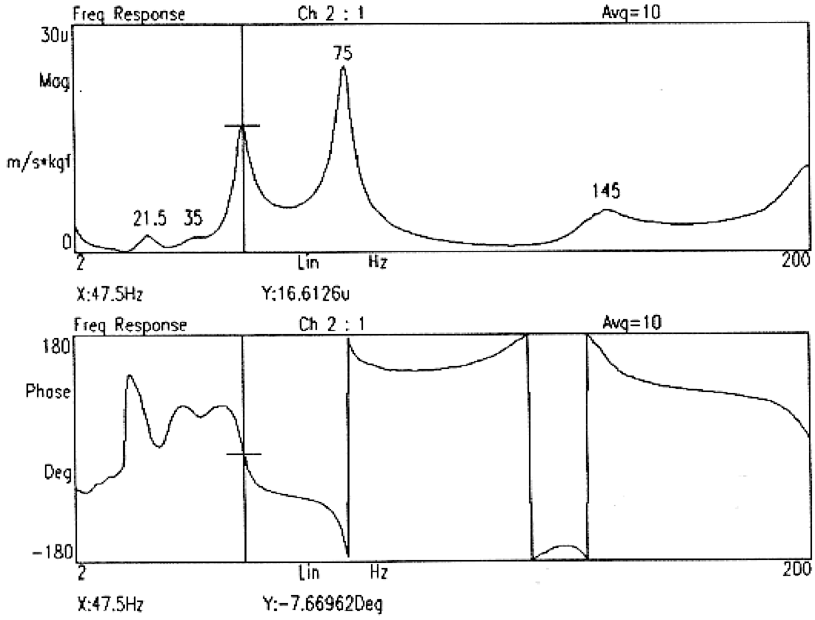

Figure 2. Frequency response function and phase diagram

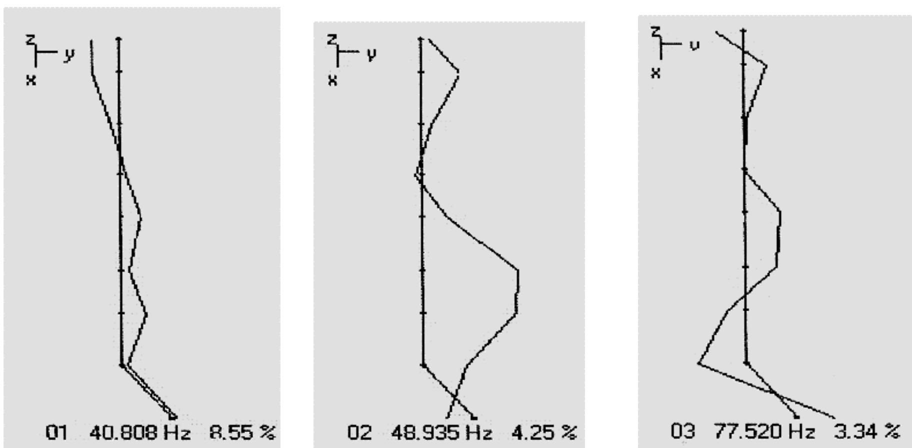

Figure 3. Second, third and fourth vibration mode shapes

In regard to damping, similar results of this measurement have been obtained through an analysis in time domain. The Fig. (4) shows the filtered responses to an impact at frequencies of 48,9 and 77,5 Hz, which are frequencies of the most excited vibration modes. In the upper curve it was taken a time period of 142,6 ms for analysis, corresponding to a seven periods of oscillation of the third mode (n = 0,1426 x 48,9 = 6,973). Considering the ratio of frequencies $\omega_{d3}/\omega_{n3} = 1$ and using the Eq. (14), it was found the damping factor $\xi_3 = 0{,}044$ or ξ_3 = 4,4 %. In a similar way, from the lower part of the Fig. (4), the damping factor of the fourth vibration mode was calculated at frequency of 77,5 Hz. The calculated value was ξ_4 = 3,47 %. As it can be observed, the two ways of determining the damping factors (frequency and time domain) have produced the same results practically.

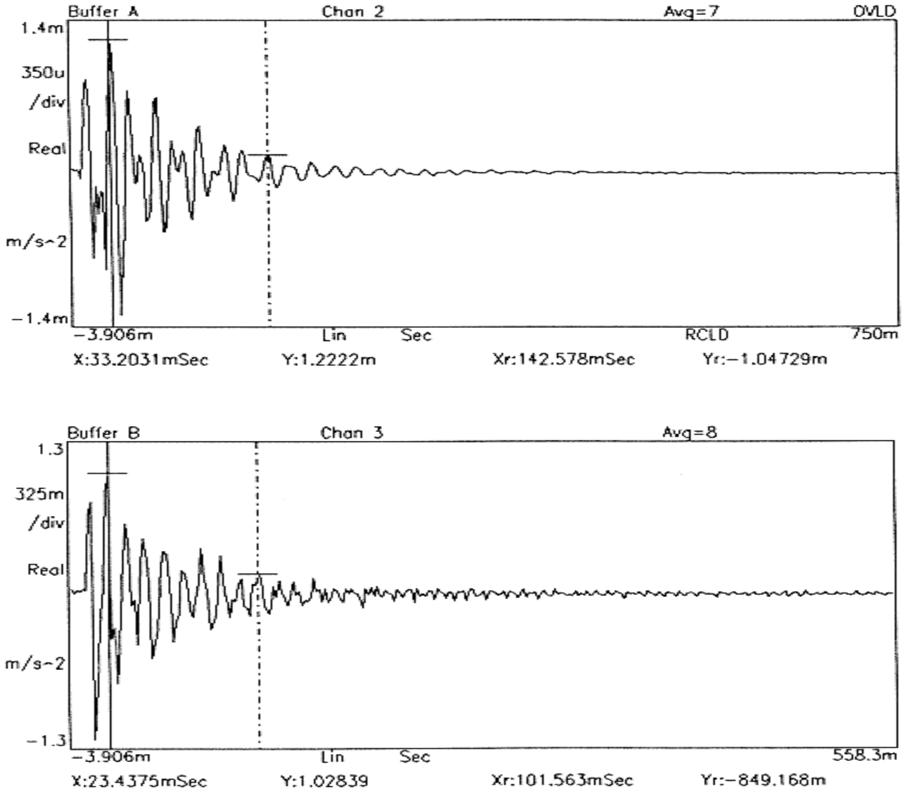

Figure 4. Curves of response in time with the still machine

5.2. Machine in Operation

The frequency response functions were not effective to identify natural frequencies through blows onto the shaft when the machine is in operation. In this condition, it is necessary blows with a lot of energy to obtain any modal response of the machine. To minimize the effect of the periodic component of rotation and its harmonicas it is necessary using very adjusted exponential window in the signal processing. As a consequence of this procedure the amplitudes at resonance frequencies don't come with clarity, mainly when the response to the blow is weak.

To improve the results and accurately determining the natural frequencies of the machine other technique has been used, in which the periodic components of the spectra are filtered out. In the filtered spectra the natural frequencies are clearly identified. The signal processing was carried out with a digital filter from cepstrum.

Figure (5) presents the modal response from filtered cepstrum with the machine in operation. It can be clearly observed the resonance's frequencies of 44.2, 70.7 and 128.5 Hz, corresponding to the third, fourth and fifth vibration modes. In comparison to the frequencies obtained with the still machine, it is noticed that all frequencies decrease of values (varying from 6 to 12%) in consequence of the dynamic effects that appear when the machine is operating, as previously mentioned.

It is very difficult to identify experimentally the damping in time domain when the machine is running since the periodic component of rotation and of its harmonicas, as well as the noise of turbulences are superimposed to the impact signal. However, using a digital filtering from cepstrum to attenuate the undesirable amplitudes, it was possible to identify the value of the damping factor of the third mode with the machine in operation. The upper part of Fig. (6) shows the response when the machine suffers a blow in the still condition and the lower part when the machine suffers a blow in the operating condition. Using the Eq. (14) it was obtained ξ_3 = 0,0443 when the machine is in the first condition and ξ_{3mot} = 0,0726 when the machine is in the second condition (it can be seen that the natural frequencies of the two conditions are different: 48.9 and 44.2 Hz, respectively). It is verified that the damping factor of this vibration mode is 64% higher when the machine is working. It can be supposed, therefore, that the values of the other modal damping factors are also higher when the machine is in operation.

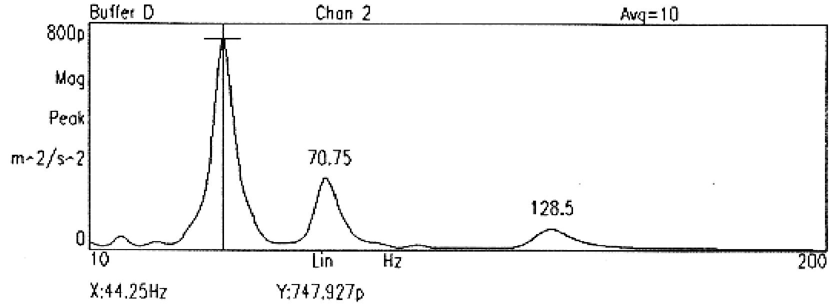

Figure 5. Modal response from filtered spectrum

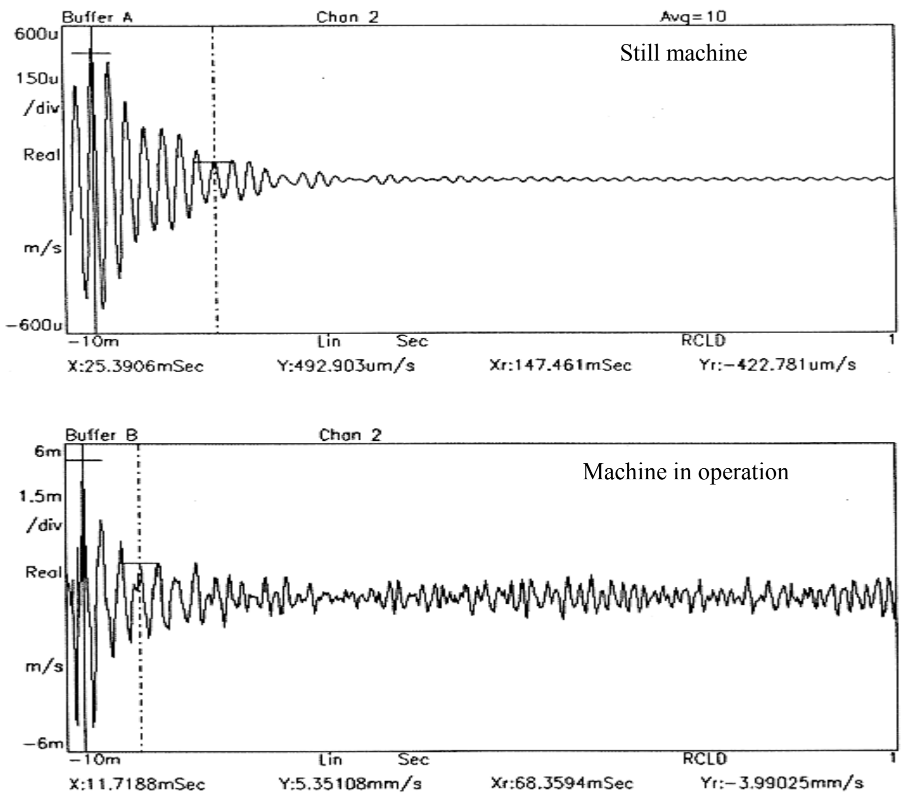

Figure 6. Response in time domain with still and operating conditions

6. CONCLUSIONS

The modal behavior of hydroelectric machines can be obtained with a reasonable accuracy through blows with an impact hammer. It is possible to identify natural frequencies, vibration modes and modal damping factors. Nevertheless, some difficult can be found to practice experimental modal analysis in big machines. The modal response can be very weak although strong blows are accomplished. In this case, in which a machine of 15 MW was analyzed, it was possible to obtain the modal behavior with very clear results.

The experimental analysis allowed obtaining important data regarding the damping of the hydroelectric units, which are unheard-of considering the lack of literature talking about of this subject. It has been identified modal damping factors through the frequency response functions using the LMS modal software package, as well as using analysis in time domain. Furthermore, it has been identified natural frequencies and vibration mode shapes. To get the modal characteristics with the machine operating it is necessary to employ a signal processing to eliminate the noise and components undesirable of the spectra. In this case, the signal filtering through edited cepstrum can be an interesting alternative.

It was observed that the damping is higher when the machines are rotating (above 60%). That indicates that the damping induced by the dynamic effects that appear when de machine is operating is very significant. It was also verified that the damping of the first and second vibration modes is much higher than the damping of the modes of superior order.

7. ACKNOWLEDGMENT

The author would like to acknowledge the "*FAPESP – Fundação de Amparo à Pesquisa de São Paulo*" and the "*FUNDUNESP – Fundação para o Desenvolvimento da Unesp*", which have granted funds for laboratory infrastructure and for publication of works like that.

8. REFERENCES

Bendat, J. S. & Piersol, A. G., 1980, "Engineering Applications of Correlation and Spectral Analysis", John Wiley & Sons.

Childs, D., 1993, "Turbomachinery Rotordynamics, Phenomena, Modelling & Analysis", John Wiley & Sons, New York.

Nascimento, L. P. & Egusquiza, E., 1996, "Application of Rotor Response Analysis to Fault Dettection in Hydro Powerplants", XVIII IAHR – International Association for Hydraulic Research Symposyum, pp. 1162-1171, Valencia, Spain.

Nascimento, L. P. & Egusquiza, E., 1997, "Parámetros que Interfieren en la Rotodinámica de Grupos Hidroeléctricos", Copimera'97, Santiago, Chile, Vol. 1, pp. 35-40.

Ohashi, H., Editor, 1991, "Vibration and Oscilation of Hydraulic Machinery", Hydraulic Machinery Book Series, University Press, Cambridge.

Simon, F., 1982, "On the Computational of the Dynamic Behaviour of Shaft Systems in Hydro-electric Power Stations", Voith Research and Construction, Vol. 28e, pp. 4.1-4.9.

Utecht, M., 1983, "Vibration Analysis for the Itaipu Generator Turbine Units", CIGRÉ Symposium, Rio de Janeiro, pp. (210-10) 1-6.