

MATHEMATICAL MODELATION AND RUNNING PROGRAM FOR DYNAMIC EVALUATION OF AN HYDROENERGETIC AGGREGATE

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Abstracts:

In order to investigate the technical situation of a hydro power aggregate, the paper presents an analytical method of vibration level evaluation, by simulating and modeling the phenomena [1,3]. As a result of processing the signals, there are presented the results of modeling - 10 own vibration ways and their features regarding the stress of the ball bearing and the deformation of the shaft (arbor) [2,4].

1. INTRODUCTION

The dynamic evaluation of hydro energetic aggregate, with rotation elements (rotators), by vibration monitoring is determinant. Two essential purposes are presented:

- keeping low dynamic forces, which means the extension of the hydro energetic aggregate time life
- failure tendency detecting for "preventing and at object" maintenance strategy application.

Dynamic forces could not be measured directly, which means that vibration parameters have to be follow through:

- the amplitude - for measuring the failure
- the frequency - for detection of the failure source
- the phase - indicating the force direction
- the form

The vibration is the result of both, force action and system flexibility. The amplitude and the frequency vibration are currently are studied.

Each element of a hydro energetic aggregate could be characterized by a determinant parameter, as a measure of it's performance. This parameter varies during the element's existence in negative direction (degeneration), because of some irreversible processes, named "depreciation" or „ageing”, till a critical value is attempt, which determine the failure of the element.

2. MATHEMATICAL MODELATION FOR VIBRATION ANALYSE OF A HYDRO ENERGETIC AGGREGATE

Vibration could be seen as aleatory variable, with increasing values most of the time, during the observation period (figure 1).

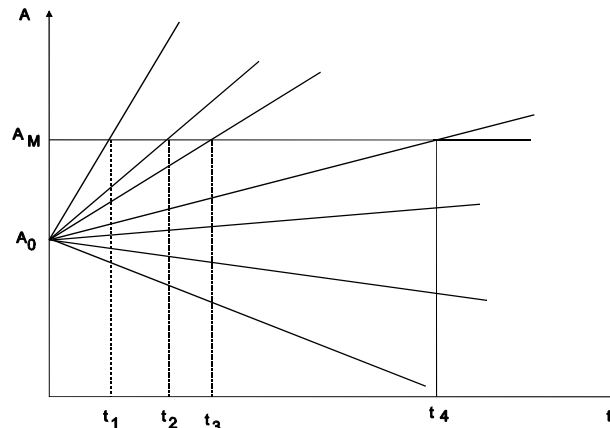


Figure 1. Qualitative evolution of amplitude- time dependence for the hydro energetic aggregate

A_0 -initial value of the amplitude

A_M -maximal admitted value

The overtaking of the admitted vibration amplitude represents the derive of a parameter, which determine the application of the “preventing and at object”. maintenance strategy for this aggregate. After the parameter A overtake A_M value, the time (time of good running) values, $(t_1, t_2, t_3, \dots)$ are equivalent with a failure with parameter derive for the hydro energetic aggregate(HEA).

Statistic analyze of the timings (t_i) , for all the points where aggregate vibration monitoring is performed, lead to the determination of the parametric relative reliability of vibration (R_v) of the aggregate, as a component of bathe global reliability:

$$R_v = \text{Pr ob} (A_{01} \leq A_{M1}; A_{02} \leq A_{M2}; \dots; A_{0n} \leq A_{Mn}) = \prod_{i=1}^n \int_{A_{0i}}^{A_{Mi}} f(A_i) dA_i \quad (1)$$

where:

„n” - the number of the points where HE A aggregate vibration are monitorized ;

$f(A_i)$ – distribution density for A_i variable.

Considering a linear „A” variable – time evolution, we may admit:

$$A = A_0 + V_A \cdot t \quad (2)$$

where: v_A – the increasing rate of vibration amplitude (degradation rate).

Measuring the value of „A” parameter, at two different moments of time, sort able delayed, (t_1, t_2) , we have:

$$\begin{cases} A_1 = A_0 + V_A \cdot t_1 \\ A_2 = A_0 + V_A \cdot t_2 \end{cases} \quad (3)$$

Solving the system from (3), the following solutions are obtained:

$$A_0 = \frac{A_1 \cdot t_2 - A_2 \cdot t_1}{t_2 - t_1}; \quad V_A = \frac{A_2 - A_1}{t_2 - t_1}. \quad (4)$$

On another hand, knowing the admitted value, (A_M) and the rate value, (v_A), we can determine the period of time measured from the moment when “A” parameter, have the A_0 value to the moment when the predictive maintenance had to be done.

$$t_F = \frac{A_M - A_0}{V_A} \quad (5)$$

For each of the variables mentioned above, (A , v_A , t_F), statistic analyze could be applied in order to determine the selection characteristics and to establish the most accurate theoretical distribution of the empirique distribution. If we admit a ordinary distribution law, which is in fact the „natural” distribution of the mentioned variable values, it is necessary to calculate the parameters: medium value (m) and dispersion (σ^2).

For example, for „A” variable, we’ll determine:

$$\begin{cases} m_A^* = \sum_{i=1}^n \frac{A_i}{n} \\ (\sigma_A^*)^2 = \frac{1}{n-1} \sum_{i=1}^n (A_i - m_A^*)^2 \end{cases} \quad (6)$$

The distribution density of “A” variable could be expressed:

$$f_T(A) = C \cdot f(A) = \frac{C}{\sqrt{2\pi} \cdot \sigma_A} e^{-\frac{(A-m_A)^2}{2(\sigma_A)^2}} \quad (7)$$

The normal distribution is truncated, because the existence of the analyse system suppose the „A” parameter evolution, among the preestablished limits [A_0 , A_M]. The „C” contant could be determined from the Laplace (Φ) function values:

$$C = \frac{1}{\Phi\left(\frac{A_M - m_A}{\sigma_A}\right) - \Phi\left(\frac{A_0 - m_A}{\sigma_A}\right)} \quad (8)$$

The above expression refers to a certain value of the HEA load, knowing that the vibration level corresponds to the load level. Taking in account the two, „S” load bearing where the HEA have a normal functionality the relative parametric reliability at the vibration could be expressed:

$$R_v = \prod_{j=1}^S R_{vj} \quad (9)$$

3. RUNNING PROGRAM FOR HEA VIBRATION MODELING

3.1. Acquisition data program

Data acquisition is realised using an acquisition board PCI12000, which has a library of functions that could be included in a C++ program. The acquisition data program gives the tension value from the piezoelectric traductor and store them up on the hard disk, as data base files. These files are used in Matlab Program. The data acquisition has a rate of 20kHz (1/20000 s time). The program starts with the board initialisation, saving the file, the data acquisition and ens with funtion resetting and file closing.

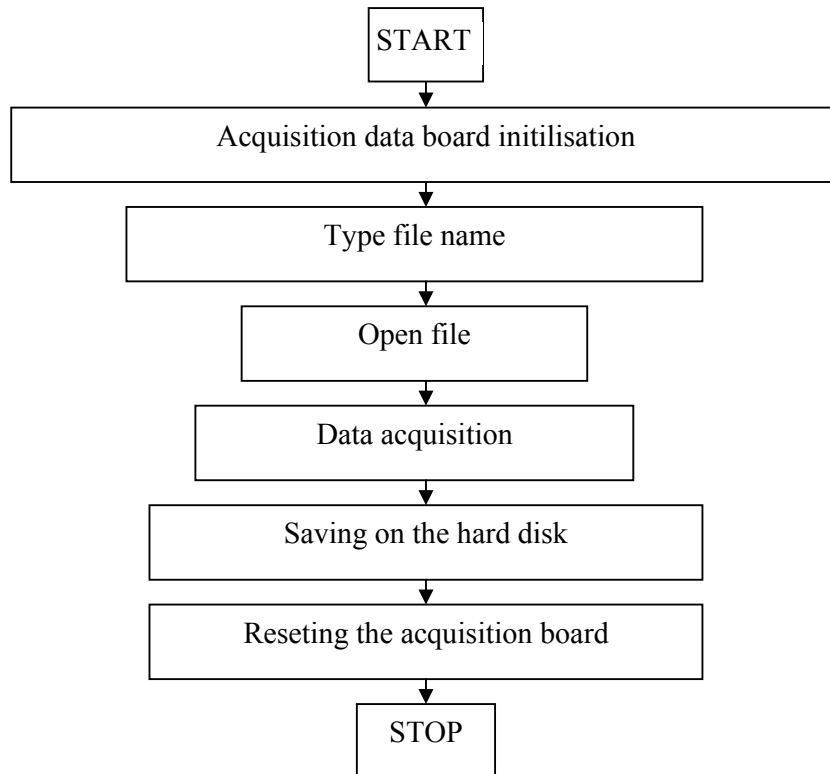


Fig. 2. Logic diagram of the acquisition data program.

3.2. Data Acquisition Analysing Program.

The relation between the amplitude of the oscilante accelerate movement and the electric signal given by the traductor could be obtained both by calculating or using the direct method, by oscilating movement generation with known parameters and analysing the signal.

The direct method was performed, using a vibration generator (vibration table), which generates sinusoidales oscillations at a frecquency of 78 Hz and an amplitude of 1 g (9,81 m/s²). The vibration table ("Metra Mess und Frequenztechnik", Germania) is delivered together with the piezoelectric traductors. The term of „power density spectrum” reflects the fact that FFT algoritm has as a result a a spectrum of narrow bands of frequency.

The program (containing the functions: signal diagram display, RMS value calculation, spectrum analyse and diagram of power density spectrum display, spectrum values calculation, spectrum diagram display), was performed in Matlab program. The logic diagram is presented in figure 3.

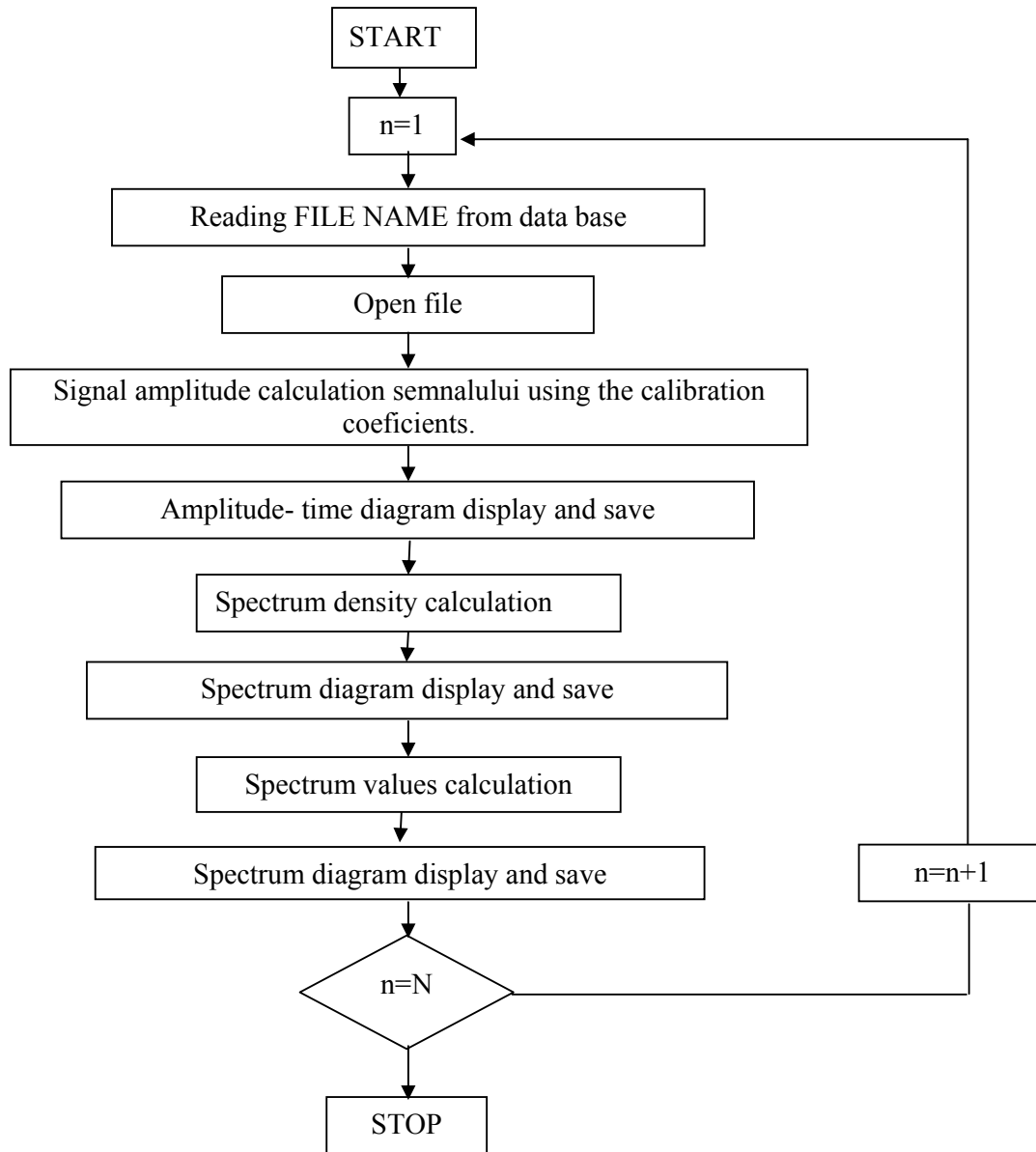


Figure 3. - Logic diagram of the data acquisition analyses program.

The proposed program could calculate:

- the acquired signal amplitude (m/s^2), using the calibration coefficients, determined as shown before;
- power spectrum density as a function of frequency;
- spectrum values which are defined by the next relationship:
- $C = \text{inv_fft}(\lg(\text{fft}(x)))$,

(10)

where :

- C- the spectrum value;
- Inv_fft – inversion of Fourier transformation;
- fft – rapid Fourier transformation;
- x – amplitude signal value in time.

4. HEA DYNAMIC DETERMINATION AND ANALYSING

4.1. Determination of the proper vibration modes.

The dynamic modelation of the turbine was realised in CATIA Program. After geometry definition, the material properties, elasticity coefficient, Poisson coefficient and density are associated with the studied element. For the analyse using finit elements, we pass in the structural analyse modul., where the finit element network and the boundary conditions are defined (place and type of the used bearing). After all the specification, the finit elemnt analyse is run, and the result are in both graphic and tabular form.

The parameters used to realise the dynamic model are:

- Material :
 - Young coefficient : $2 \cdot 10^{11} \text{ N/m}^2$;
 - Poisson coefficient: 0,266.
- Model structure :
 - Number of nodes : 1853;
 - Number of elements : 7328;
 - Number of variance: 5778;
 - Number of kinematic equations : 420
- Coordonates of the load centre :
 - Xg : 585,7 mm;
 - Yg : -585,3 mm;
 - Zg : -713,6 mm.
- Inertial tensor in origine :

2.450e+009 7.140e+006 9.616e+006

7.140e+006 6.749e+010 -4.198e+006

9.616e+006 -4.198e+006 6.750e+010

4.2. Simulation results

The simulation results were obtained taking in consideration the next parameters:

- Number of the propre calculate modes : 10;
- Number of iterations :44;
- Tolerance of the propre obtained value : $2.089 \cdot 10^{-4}$.

The propre modes are presented in table 1, and the coefficients of propre form of vibration are presented in table 2, where: Tx,Ty,Tz,Rx,Ry,Rz represent the variances: translation, and rotation, respectively.

Table 1. Characterization of the vibration proper modes

VPM number	Proper Frequency [Hz]	VPM characterization by the: :	
		frequency	deformation
1	3,3692	Very low	Axial vibrations of therotator- shaft ensemble
2	138,61	low	Radial vibrations in RB and TB; easy bending of the shaft
3	143,66	low	Radial vibrations in RB and TB; Hard bending of the shaft between RB and TB
4	447,36	medium	High radials vibrations in RAB, and smaller in RB and TB, sheft bending in RAB
5	462,55	Medium	Vibration promenade in LRA
6	528,82	Medium	Low vibrations in RB, bigger in TB, shaft bending between RB and TB
7	935,75	medium -high	Longitudinal vibrations, radial reduced in all bearings
8	961,26	medium -high	Strong vibrations in RB and TB on radial direction with shaft bending between RB and TB
9	1075,3	high	High longitudinal vibrations and radial vibrations in RB and TB, shaft bending between RB and TB
10	1093,0	High	Very strong vibrations in RB and TB on radial direction with shaft bending between RB and TB

VPM – vibration propre mode; RAB – radial axial bearing; RB – radial bearing; TB – turbine bearing.

Table 2. Values of the loadings and dynamic reaction in bearings

Number	Frequency [Hz]	Tx [%]	Ty [%]	Tz [%]	Rx [%]	Ry [%]	Rz [%]
1	3,3692	0.00	0.14	0.59	99.67	0.56	0.08
2	138,61	0.00	7.90	79.86	0.23	68.84	6.78
3	143,66	0.00	79.98	7.74	0.03	6.66	69.03
4	447,36	0.01	0.00	0.00	0.04	7.79	0.87
5	462,55	0.00	0.00	0.00	0.00	0.82	7.81
6	528,82	0.00	0.00	0.00	0.00	0.07	0.08
7	935,75	0.14	0.45	5.20	0.00	3.30	0.29
8	961,26	0.02	5.18	0.44	0.00	0.27	3.32
9	1075,3	87.15	0.00	0.01	0.00	0.00	0.00
10	1093,0	0.10	0.03	0.02	0.00	0.01	0.01

In the next figures are suggested the deformations of the ensemble at the propre calculated frequencies.

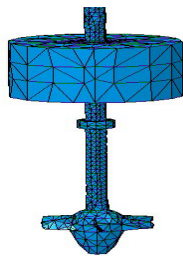


Figure 6. Mode1 –Shaft deformation at the frequency of 3,3692 Hz.

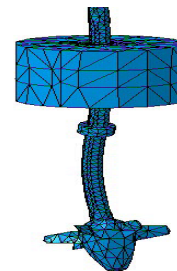


Figure 7. Mode 2- Shaft deformation at the frequency of 138,61 Hz.

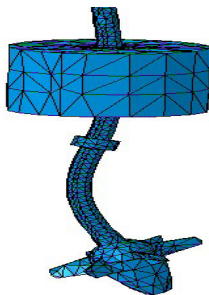


Figure 8. Mode 3- Shaft deformation at the frequency of 143,66 Hz.

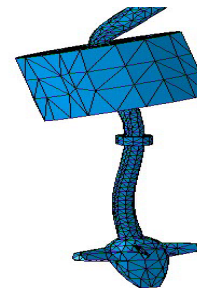


Figure 9. Mode 4 - Shaft deformation at the frequency of 447,36 Hz.

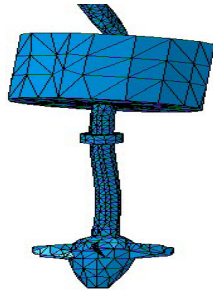


Figure 10. Mode 5- Shaft deformation at the frequency of 462,55 Hz.

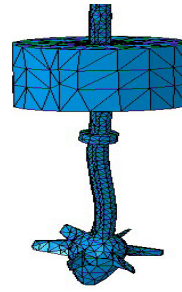


Figure 11. Mode 6 -Shaft deformation at the frequency of 528,82 Hz.

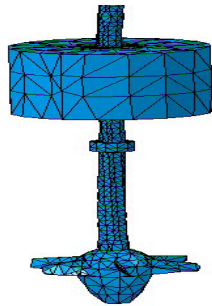


Figure 12. Mode 7- Shaft deformation at the frequency of 935,75 Hz.

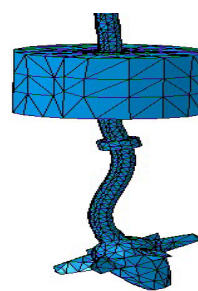


Figure 13. Mode 8- Shaft deformation at the frequency of 961,26 Hz.

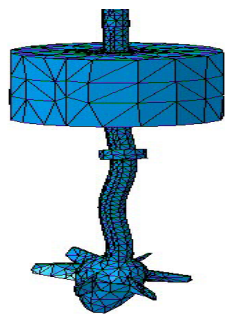


Figure 14. Mode9- Shaft deformation at the frequency of 1075,3 Hz.

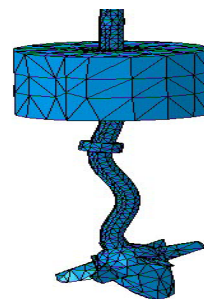


Figure 15. Mode10- Shaft deformation at the frequency of 1093,0 Hz.

5. CONCLUSIONS

1. For the simulation of the dynamic behavior of the hydroenergetic equipments, first we have to identify the right relationship between the model and the process.

2. In this paper for the dynamic behavior simulation it is underlined the necessity of building a dynamic model, based on the measurements of the process parameters, which means experimental identification. In this case the existence of any theoretical knowledge, in relation with the process structure or parameters values, could be used in order to reduce the search area in a „class” of models or parameter space.

3. The modelation of the experimental system functionality, in the case of using different bearing, as: pillow block bearing or drop hanger bearing, if the model is correct, could lead to the prediction of the system functionality also in some different conditions, than the experimental ones. Using models the system response could be established in boundary conditions which may lead to major failures.

4. In order to determine the propre modes of vibration (PVM) of an aggregate, a dynamic model and also a suitable calculating program(CATIA) will be used.

5. A number of 10 PVM are quite enough for the study of vibrational phenomena effects on the deformation degree of a rotator-shaft ensemble from the structure of a hydroenergetic aggregate, coresponding to 10 propre frequencies, which belong to five domaines(very low, low, medium,medium high, high) with determined loading and system deformation.

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