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EVALUATING THE LIFETIME OF HYDRO-MECHANICAL EQUIPMENT FOR LOW HEAD HYDROPOWER PLANTS

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Modern techniques for the estimation of lifetime for such equipments, allow the determination of the upper limit for safety in operation, the determination of areas and elements of high fatigue and also the required corrections in the refurbishment process. This paper is presenting a new and innovative way to determine the lifetime in case of a shaft for hydraulic turbine Kaplan. The analyzed element from the lifetime point of view is the turbine shaft, using for this an integrated system. The main parameters taken into consideration for durability calculation are: model geometry (load forces, local tensions), material properties (fundamental mechanical properties, surface quality and level of local tensions) and applied load for simulation (level of load, amplitude and load type).

Keywords: shaft, hydropower plants, lifetime remaining.

1. Introduction

Currently the majority of existing equipment in the operational hydroelectric power plants in our country have at least 25-30 years of activity. All these devices were built into the system unique replacement rule is extremely expensive.

The hydro-mechanical equipment is stressed by its own weight, water force and electromagnetic field, in static and dynamic regime.

In this paper the authors presents the status of research undertaken within the project SOP 62 557 EXCEL - Excellence postdoctoral research programs in priority areas of society based on knowledge of the topic on "Research on the remaining lifetime of hydromechanical equipment."

2. Integrated System

Research that need to be made have the final goal the establishment of a functional technological model framework based on data collected and evaluated to allow development of specific options for each equipment or part in question.

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Exponential development of computer programs has facilitated the parametric design software of the equipments with a rapid development (AutoCAD, Mechanical, Catia, ProEngineer etc.). In parallel with these programs, there were developed numerical simulation and finite element analysis programs devoted to calculations of resistance and particularly resistance to fatigue. Currently all CAD programs have embedded modules of finite element analysis models which may be generated.

All this has led, at present, to improve sustainability assessment by numerical simulation and testing. Such a process the following steps:

1. Virtual prototype. Development of the three-dimensional model (3D) using a CAD program. The resulting model (at a scale of 1:1) is a carrier information model:

- Geometric information: size, mass, volume, moments of inertia, etc.
- Information about the nature of the material (physical, chemical and mechanical) to the developed model is attached a material library provided by the program; if the material is new, there is the possibility of placing it in the database together with its properties.

2. Evaluation of loads: if the stress loads are known these are converted according to the parameters of interest. If there is a possibility of their acquisition by a data acquisition and processing system of the right equipment while in operation we have an ideal case.

3. FEM. The virtual prototype is transferred to a program compatible with finite element analysis. Here the model is tested at loads (stress) being put on the model (data from data acquisition system or estimated). The advantage of using finite element analysis program is to use time variable loads.

4. Meshing model and stress application is followed by numerical simulation load software according to laboratory tests. In this stage 90% of relevant information is obtained of the actual model behavior based on a virtual prototype decision-making, which may lead to the improvement of its performance.

5. The changes to the virtual prototype requires re-start and re-evaluation of its new features and therefore the pre-diagnosis of lifetime cycle.

6. Once the virtual prototype is going to meet the requirements, the development of physical prototype is starting, operational stress estimation and testing in the laboratory using a suitable testing program.

Based on what was presented above, the integrated system of design/diagnosis to availability of one equipment is shown in Fig. 1.

For equipments beeing in operation, such a system allows to: estimate the remaining life of hydro power plant's equipment in use; determine areas and equipment which is subject to heavy fatigue; determination of limit time in operation; identify the necessary corrections in the process of rehabilitation. This paper aims to present a methodology for life cycle approach to calculation of a turbine shaft that is fitting a hydraulic turbine type Kaplan. Subassembly analyzed in terms of life is the turbine shaft.

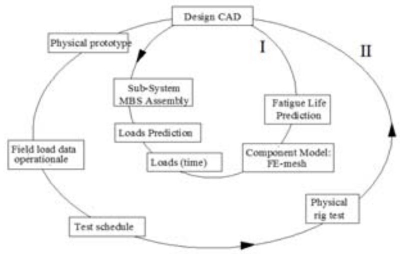


Fig. 1. Integrated system.

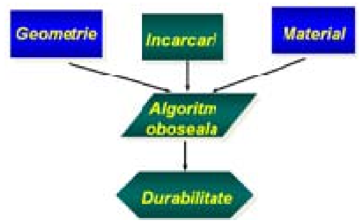


Fig. 2. Parameters of analysis.

The main parameters (Fig. 2) taken into account for calculation of sustainability are: model geometry and the application of load factor (concentrators, local tensions), material properties (mechanical properties of core and surface quality of the local tensions) and applied loads (the load, the magnitude and type of load).

For analisys teo methods was used:

1. Tension analysis. Starting from the stress loads applied to the model, the nominal tensions are evaluated and then the information about the local elastic tensions are obtained. In the end, an estimation of the lifetime expressed in relation with elastic stress from the important area.

2. Analysis of the strain. The sustainability assessment based on distortion shall be considered as the degradation as a function of local plastic deformed. Elasto-plastic behavior is described with a model of hysteresis loop. Degradation depends on the position, size and the order of hysteresis loops.

3. Geometrical Model

Geometrical model taken from the technical documentation was created as a 3D Model in Mechanical Desktop and then transferred as a file .stp to be retrieved in ANSYS (Fig. 3).

In achieving three-dimensional computer model, simplifications were made in the bearing geometry and clamping flanges. For the mesh there were used SOLID187 tetrahedral spatial element as follows:

- In the areas connecting the two flanges was imposed an automatic refining specialized algorithms based on valuation criteria;

- In the radial bearing area, an orderly meshing was done.

While modeling has always sought to create as uniform rectangular networks to limit calculation errors inherent in the program. The result is a model consisting of nodes 94 369 and 58 116 elements (Fig. 4).

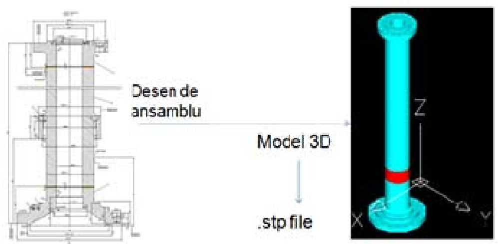


Fig. 3. 3D Model of Kaplan turbine shaft

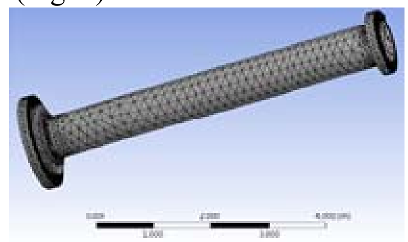


Fig. 4. Meshed Model

4. Material Properties

Using specialized programs of finite element analysis, requires a larger number of material properties in comparison with the classical calculation of fatigue.

The shaft material is, according to the designer, OLC 25X (improved quality carbon steel). Its equivalent in databases AISI / SAE ASTM is Steel 1020, for which Su = 455 MPa. Its main properties are presented in Table 1.

Table 1

Name	Simbol	Value	UM
Elastic Module	E	206800	MPa
Ultimate Strength	S_u	455	MPa
Fatigue Strength Coefficient	σ'_f	883	MPa
Fatigue Strength Exponent	b	-0,118	-
Fatigue Ductility Coefficient	ε'_f	0,16	-
Fatigue Ductility Exponent	c	-0,412	-
Cyclic Strength Coefficient	K’	1441	MPa
Cyclic Strain Hardening Exponent	n'	0,283	-

Using empirical methods in the literature we determined the fatigue curve for OLC 25X and cyclic stress-strain curve dependence.

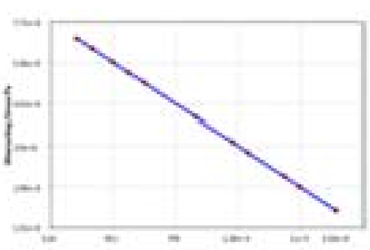


Fig.5. Fatigue curve for OLC 25X.

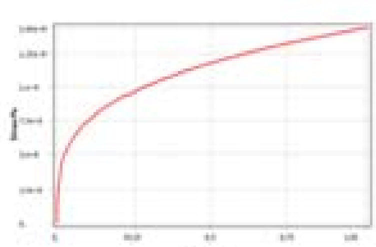
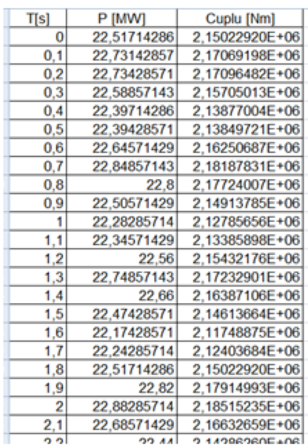


Fig. 6. Cyclic stress-strain curve dependence.

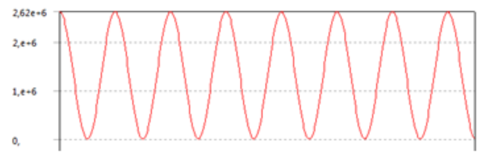
5. Stress Loads And Data Acquisition

The necessary parameter to define the stress analysis loads of the hydrogenerator shaft, is the torque. The torque is very difficult to measure, and therefore it has been calculated based on the delivered power from the central panel and then, by mathematical analysis the torque developed was estimated (Fig. 7).

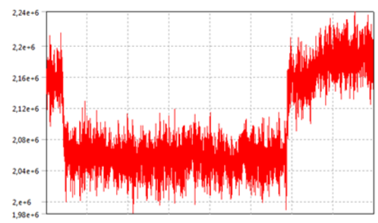


T[s]	P [MW]	Cuplu [Nm]
0	22,51714286	2,15022920E+06
0,1	22,73142857	2,17069198E+06
0,2	22,73428571	2,17096482E+06
0,3	22,58857143	2,15705013E+06
0,4	22,39714286	2,13877004E+06
0,5	22,39428571	2,13849721E+06
0,6	22,64571429	2,16250687E+06
0,7	22,84857143	2,18187831E+06
0,8	22,8	2,17724007E+06
0,9	22,50571429	2,14913785E+06
1	22,28285714	2,12785656E+06
1,1	22,34571429	2,13385898E+06
1,2	22,56	2,15432176E+06
1,3	22,74857143	2,17232901E+06
1,4	22,66	2,16387106E+06
1,5	22,47428571	2,14613664E+06
1,6	22,17428571	2,11748875E+06
1,7	22,24285714	2,12403684E+06
1,8	22,51714286	2,15022920E+06
1,9	22,82	2,17914993E+06
2	22,88285714	2,18515235E+06
2,1	22,68571429	2,16632659E+06

Fig. 7. Calculated and measured parameters



a) pulsatory cycle with constant amplitude



b) Random cycle from measurement data

Fig. 8. Applied stress cycles for the model

The main parameters of the hydro power unit used: Nominal Power (P = 23,5 MW); Nominal speed (n = 100 rot/min); Mechanical efficiency of shaft (99%); Generator efficiency (95%, considered constant to simplify the model); Overall efficiency (94.05%, from electrical power delivered to mechanical power at turbine shaft).

The torque was determined with the relation:

$$M = \eta_t \times P/\omega = \eta_t \times P \times 60/2\pi n = \eta_t \times P \times 30/\pi n$$

For the nominal power of the turbine (23,5 MW) a maximum torque of M=2,62466 x 10^6 Nm.

The time of the data acquisition was 10 minutes with a frequency of reading of 0,1 sec.

Fixing conditions imposed: For the radial bearing surface was permitted a axial movement, constraining a null movement in radial direction (Fig. 9). For mounting flange of the hydrogenerator, null displacements were imposed in all directions.

Torque is applied uniformly distributed on cylindrical surfaces of the 20 holes of shaft's mounting flange with the turbine (Fig. 10).

6. Mathematical Model and Simulation

6.1 Modal Analysis. The first preliminary analysis was the modal analysis, which determined the first 6 own ways (Fig. 11).

Fig. 9. Imposed fixing condition (Radial bearing)

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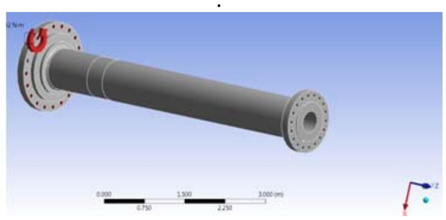


Fig. 10. External load

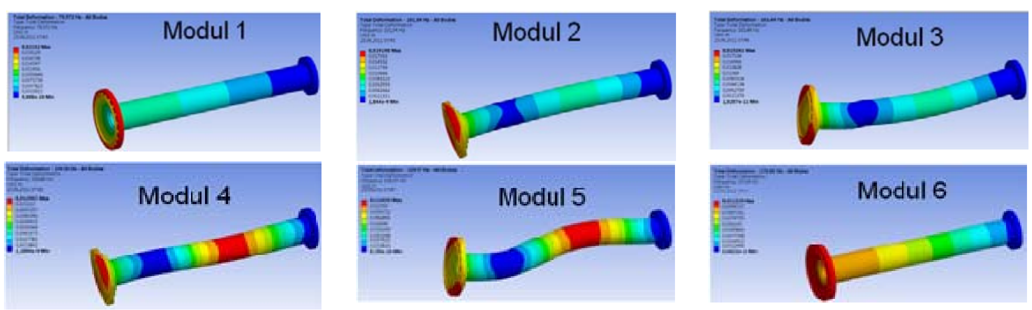
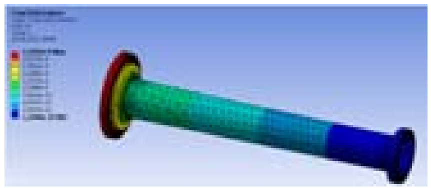


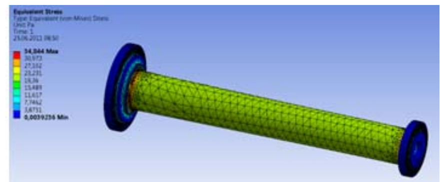
Fig. 11. Own ways.

6.2. Pulsatory cycle with constant amplitude. A second analysis that was submitted to the model was to charge it with a constant amplitude pulse cycle (fig. 8,a) equal to the nominal value of torque corresponding to maximum power turbines. The analysis was performed according to the two theories (stress life and foreign life), results are presented in Fig. 12.

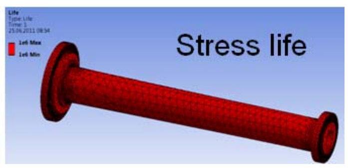
6.3. Random cycle. Cycle loading with random requests (data processing) (Fig. 8,b) was performed according to the theory of life stress (Fig. 13). The result was a lifetime cycle of N = 1.0152 e6 cycles, which is equivalent to 169,200 working hours. If random cycles are used, the diagram obtained must be presented with Rainflow method (Fig. 14).



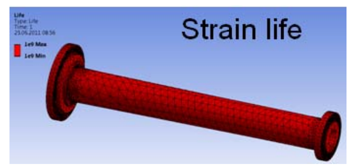
a) Overall deformation



b) Equivalent tensions



c) stress life: N=1e6 cicluri



d) strain life: N=1e9 cicluri

Fig. 12. Pulsatory cycle load with constant amplitude

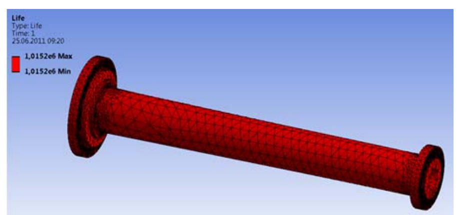
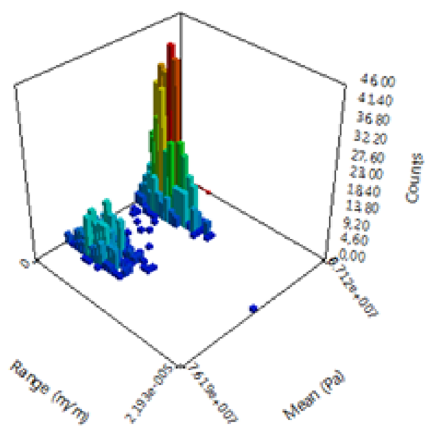
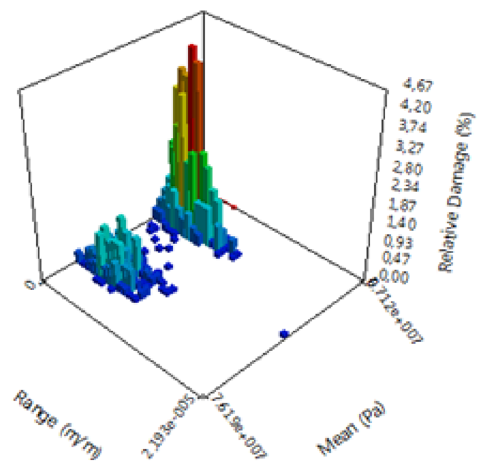


Fig. 13. Simulation results on random requests loading cycle.



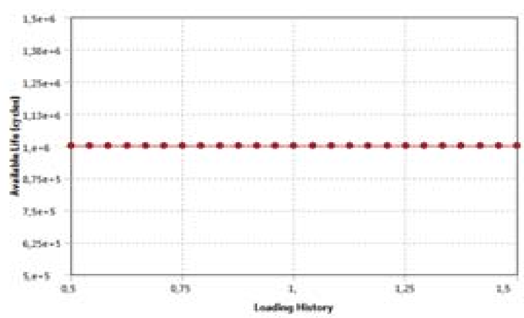
Rainflow Matrix: identification of numbers of cycles of same amplitude



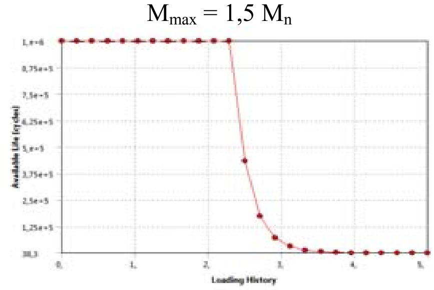
Relative deterioration matrix: equal amplitude cycles associated with provoqued relative deterioration

Fig. 14. Rainflow Diagram

6.4. Model sensibility analysis. Cycle loading with random requests (according to data processing) was performed according to the theory of life stress (Fig. 15).



$M_{max} = 1{,}5\ M_n$



$M_{max} = 5\ M_n$

Fig. 15. Model sensibility analysis

From the analysis with a load of 150% compared to the normal one, we can observe that the remaining lifetime is not modified.

In case of load of 500% of the nominal load (accident generator), a very strong diminishing of the life time is observed, and has to be taken into consideration in order to improve the model.

7. Conclusion

From the analysis of the own 6 modes, some conclusions were taken:

- The first six own frequencies are large enough to avoid resonance phenomena;
- The dynamic behavior of the model performed better, with no discontinuities in the obtained response;
- Model is consistent in terms of mass properties (18945 kg)

Estimation of lifetime according to the theory of "stress life" is consistent with reality (180000-200000 equivalent operating hours) if one takes into account the simplifications of the model and the fact that the simulation was performed for normal operation load, situation not very often in the real operation.

From this point of view it may be that the model is developed to meet actual operating conditions, but needs improvement, as will be shown below.

The mathematical model made, is corresponding both from structural and functional point of view, but for a more precise estimate of life is necessary:

- Taking into account transient phenomena during startup and shutdown of the hydro generator;
- Introducing the influence of turbine rotor (inertia) and flow of water (hydrodynamic forces);
- Consideration of incidents (accidents) in the model (inherent, on average 1-2 per year) involving major changes in priming the cracks (reducing the lifetime).
- Introducing vibration measurements into the mathematical model (vibration diagnosis)

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