13th International Research/Expert Conference
"Trends in the Development of Machinery and Associated Technology"
TMT 2009, Hammamet, Tunisia, 16-21 October 2009

THE ROTATING SHAFT BEHAVIOR IN TRANSIENT PERIOD AT THE GENERAL MODEL OF TURBOGENERATOR

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ABSTRACT
For the general model of a turbogenerator as a set of turbines and electric generator, in this paper the behaviour of the rotating shaft is analysed.
As such a complicated machine, the turbogenerator is presented by a dynamic model supported in different types of bearings also taking into consideration electromagnetic moment and the resistant moment of the working machine.
Experience has shown that the processes that occur during the transient period are of great importance for the system, therefore the rotating shaft is analysed at such a period.
The rotating shaft undergoes different types of oscillation depending in its geometry and type of support.
The adopted models – mechanical and its mathematical - are a good base for the dynamic analysis at the different models of general model.
Keywords: Turbogenerator, Transient Period, Rotating Shaft, Oscillation

1. INTRODUCTION

Turbogenerators as complicated machines that are used to develop power consist of several turbines (gas and/or steam) coupled to the electric generator. Their components, turbines and generators are subject ot rotordynamics. For the design of the rotating machines the torsional oscillations analysis are of vital importance, but in this paper a general model includes also bending oscillations.
The general dynamic model that has been built based on a model with a single non-central disk with necessary approximations for two types of support – rigid and elastic bearings and as well as for rigid and elastic foundation takes into consideration electromagnetic moment and resistant moment of the working machine [1,6].
The model also includes the oscillations of the elastic shaft depending upon its geometry, number of disks assembled on it, the type of support (bearings and foundation), excitation forces and gyroscopic effect.
For the dynamic analysis of the transient process involving mechanical, thermal and electromagnetic processes, the general model was simplified.

2. GENERAL MODEL OF TURBOGENERATOR

For dynamic analysis, the turbogenerator is expressed through a model representing an elastic shaft with several disks supported i several bearings and foundation. The general dynamic model of elastic shaft with *n-disks* supported in *(n+1)* bearings-foundation system is given in *Figure 1.*

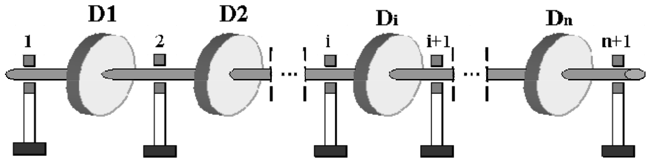


Figure 1. Dynamic model of an elastic shaft with n-disks supported in (n+1) bearings-foundations system

Based on the dynamic model in *Figure 1* can be adopted the turbogenerator consisting three turbines coupled with an electric generator supported in five bearing-foundation systems, *Figure 2.*

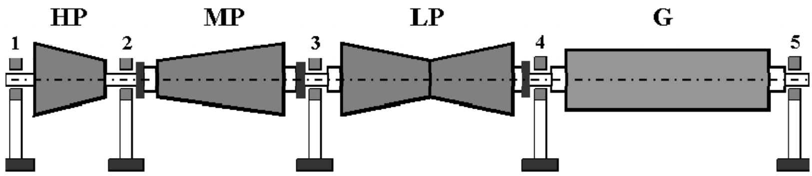
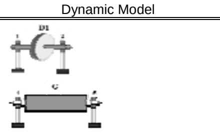
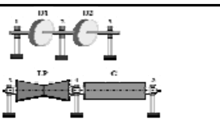
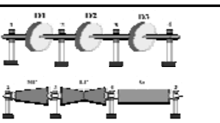
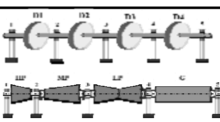


Figure 2. Adopted dynamic model for Turbogenerator
(HP,MP,LP-high, mid, low pressure turbine; G-electric generator; 1,2,3,4,5-bearing-foundation system)

Therefore, the general mathematical model is given [1,6]:

$$[M]\cdot[\ddot{q}]+([G]+[K])\cdot[\dot{q}]+[P]\cdot[q]=[Q] \qquad (1)$$

Table 1. Types of dynamic models for turbogenerator

	Dynamic Model	#	RB	EB	BwLsRF	BwLsEF	DoF
Elastic shaft with one disk supported in two bearings		1	×				5
		2		×			5
		3			×		9
		4				×	13
Elastic shaft with two disks supported in three bearings		5	×				10
		6		×			10
		7			×		16
		8				×	22
Elastic shaft with three disks supported in four bearings		9	×				15
		10		×			15
		11			×		23
		12				×	31
Elastic shaft with four disks supported in five bearings		13	×				20
		14		×			20
		15			×		30
		16				×	40

RB - *Rigid Bearing*
EB - *Elatic Bearing*
BwLsRF - *Bearing with Lubricant supported in Rigid Foundation*
BwLsEF - *Bearing with Lubricant supported in Elastic Foundation*
DoF - *Degree of Freedom*

Where: **[M]** is matrix for masses of the disks, masses of the bearings, polar and equatorial moments of inertia; **[G]** is matrix representing gyroscopic effect; **[K]** is matrix that presents damping coefficients for disks, shaft, lubricant and foundation; **[P]** is matrix for stiffness coefficients; **{q}** is vector for generalized coordinates with its first and second derivate; **{Q}** is vector of generalized load-external excitations.

Depending on number of turbines, number of bearings and the type of support the degree of freedom for the dynamic model is determined. DoF determines the number of differential equations of the system (1):

DoF = nq = (n x 2) + (n x 2) + (n x 1) + (nb x 2) + (nb x 2) (2)

3. PROCESS ANALYSIS AT TRANSIENT PERIOD

The transient processes occur at rotating machines with instantaneous changes in machine conditions as speed, load, usually during start-up or shut-down that is called transient period. At the electromechanical systems the transient processes appear as a result of electromagnetic and mechanical inertia of the system.

For analysis the dynamic model (*Table 1*) of elastic shaft with one disk supported in two bearings is adopted. Its mathematical model is:

$$\begin{aligned}
&m\ddot{\xi} + k_{C1}\dot{\xi} + k_2\dot{\xi}^2 + k_3\dot{\xi}^3 + p_{k11}\xi + p_{k14}\beta^* = me(\ddot{\varphi}\sin\varphi + \dot{\varphi}^2\cos\varphi)\\
&m\ddot{\eta} + k_{C1}\dot{\eta} + k_2\dot{\eta}^2 + k_3\dot{\eta}^3 + p_{k22}\eta + p_{k23}\beta^* = me(-\ddot{\varphi}\cos\varphi + \dot{\varphi}^2\sin\varphi)\\
&A\ddot{\alpha}^* + J\dot{\varphi}\dot{\beta}^* + J\ddot{\varphi}\beta^* + k_{C3}\dot{\alpha}^* + p_{k32}\eta + p_{k33}\alpha^* = \delta(A-J)[\ddot{\varphi}\cos(\varphi-\varepsilon) - \dot{\varphi}^2\sin(\varphi-\varepsilon)]\\
&A\ddot{\beta}^* - J\dot{\varphi}\dot{\alpha}^* - J\ddot{\varphi}\alpha^* + k_{C4}\dot{\beta}^* + p_{k41}\xi + p_{k44}\beta^* = \delta(A-J)[\ddot{\varphi}\sin(\varphi-\varepsilon) - \dot{\varphi}^2\cos(\varphi-\varepsilon)]\\
&J\ddot{\varphi} + c_1\varphi + c_2\varphi^2 + c_3\varphi^3 + b_1\dot{\varphi} + b_2\dot{\varphi}^2 + b_3\dot{\varphi}^3 = 0
\end{aligned} \qquad (3)$$

The mechanical transient processes will be analysed for changeable angular velocity considering:

$$\begin{aligned}
&\varphi = \varphi_0 + \varpi_0 t + \frac{1}{2}\varepsilon_0 t^2\\
&\dot{\varphi} = \varpi = \varpi_0 + \varepsilon_0 t\\
&\ddot{\varphi} = \varepsilon = \varepsilon_0
\end{aligned} \qquad (4)$$

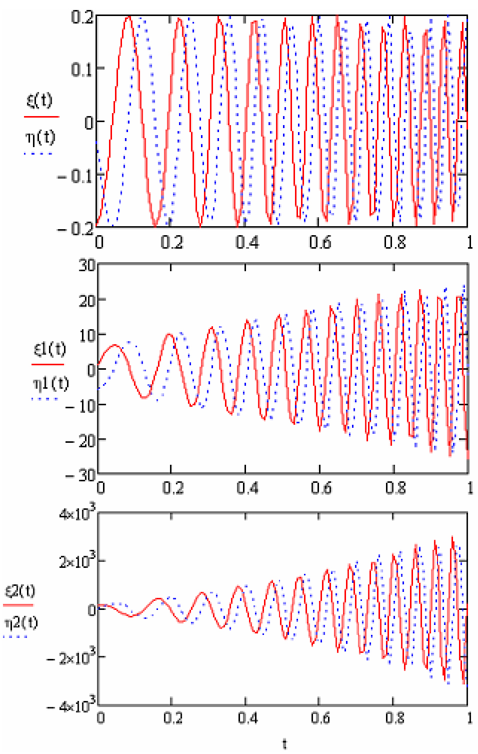


Figure 3. Coordinates, velocity and acceleration

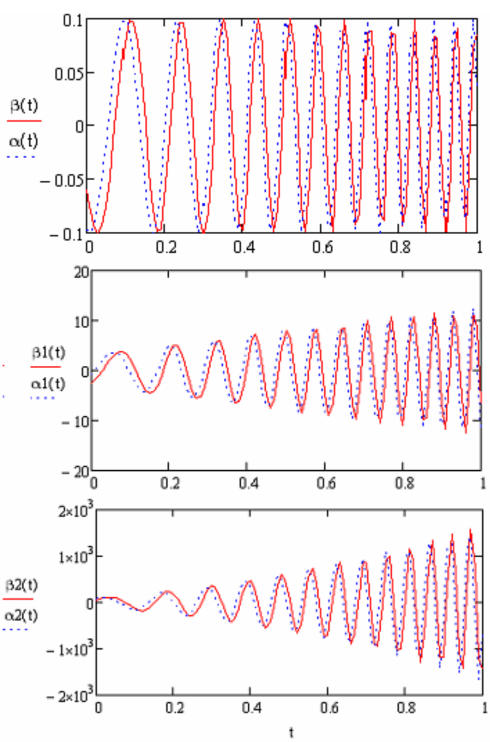


Figure 4. Nutation and Precession, their velocity and acceleration

Where: ***m*** is disk masses; ***A*** is disk's equatorial moment of inertia; ***J*** is disk polar moment of inertia; ***e*** is linear eccentricity (static des-equilibration); ***δ*** is angular eccentricity (dynamic des-equilibration);

α* is precession angle; ***β**** is nutation angle; ***φ, ω, ε*** are rotational angle/velocity/acceleration; ***c, b, k, p*** are stiffness, damping coefficients of shaft, bearings or foundation respectively.
Particular solutions will be found after several mathematical operations and approximations [1], while graphics are presented in Figure 3, 4 and 5.

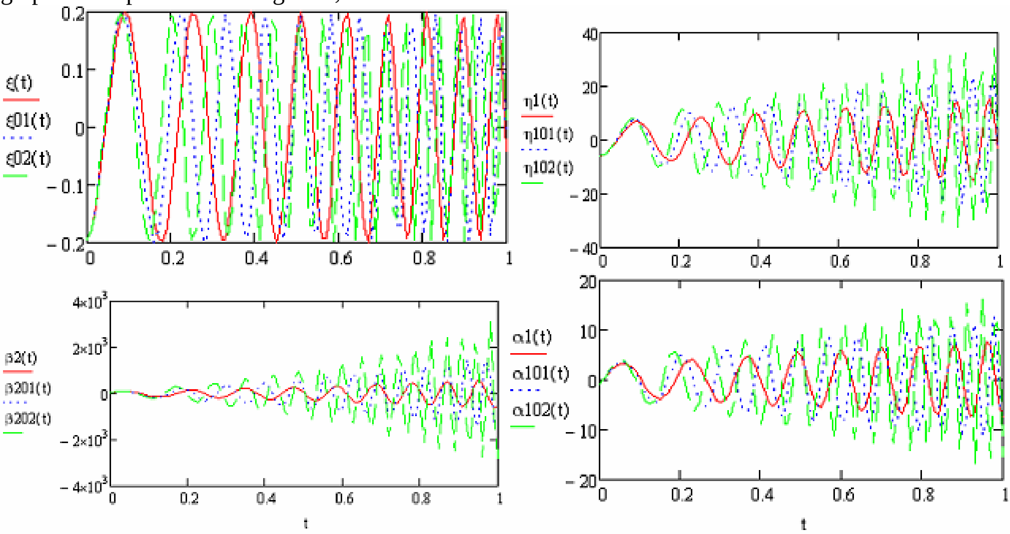


Figure 5. Compare of the particular solutions for different angular acceleration (50, 100 and 150 rad/s²)

4. CONCLUSIONS

Referring to the results graphically presented in *Figure 3, 4, 5* it can be noticed that:

- The values for the angle of precession and nutation and their components are lower than those for coordinates and their components (*Figure 3* and *Figure 4*);
- With increase of the initial acceleration angle (*Figure 5*) the values for all other quantities (coordinates and precession and nutation angles and their components) increase as well;
- For higher values of the time, the amplitudes are achieved quicker;

Therefore, it must be attempted that the period of transient processes be shorten during the cyclic working regime with often start-ups, but always keeping in mind that significant shortenings can bring to enormous increase of the inertial forces and dynamic loads, that may bring to damage of the system.

5. REFERENCES

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