

VIBRATION-TERMAL ANALYSIS OF ROTORS

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Abstract: The appearance of an uneven temperature field in the rotor occurs due to uneven heating or cooling and due to asymmetric distribution in the structure of the rotor masses. As a consequence of the uneven distribution of the temperature field in the rotor, deformations (thermal deflection) occur, which result in a change in vibration levels and unstable system operation. These deformations are manifested as additional deflections of the rotor, so that the state of balance of the system changes significantly. Vibration analysis and monitoring are one of the most reliable ways to diagnose the existence of thermal deflection. This paper deals with the phenomena of thermal deflection, the appropriate mathematical model is given, and the emphasis is on diagnosing this problem using vibration analysis.

Keywords: vibration, thermal analysis, diagnostics, rotors

1. INTRODUCTION

The appearance of an uneven temperature field in the rotor of the turbogenerator occurs due to uneven heating or cooling and due to asymmetric distribution in the structure of the rotor masses. As a consequence of the uneven distribution of the temperature field in the rotor, its deformations occur, which result in a change in vibration levels and unstable operation of the system. These deformations are manifested as additional deflections of the rotor, so that the state of balance of the system changes significantly. Vibration analysis and monitoring are one of the most reliable ways to diagnose the existence of thermal deflection [1..5]. By monitoring the trend of 1X vibration component, due imbalance, comparing with the trend of process parameters (primarily reactive power) the existence of this defect can be diagnosed. Thermal imbalance during stationary operation of the generator causes a change in the amplitude and phase of the 1X vibration component. The paper deals with the phenomena of thermal deflection, the appropriate mathematical model is given and the emphasis is on diagnosing this defect by vibration analysis.

It should be noted that the thermal balancing process is carried out by changing the intensity and location of cooling on the rotor, which is achieved by dampening the flow of coolant. These interventions can be reported on the basis of detailed analyzes of the causes in the complex heating and cooling system in turbogenerators. [5..8]

2. FORMAT TEMPERATURE FIELD OF TURBOGENERATOR ROTOR

During the operation of synchronous generators, a large part of energy is lost and converted into heat. Losses consist of losses in the stator and rotor windings, then in the core and losses in bearings and seals where the kinetic energy is converted into heat. The examination revealed empirical dependences of the lost energy converted into heat and other types of energy and power of synchronous generators (Fig. 1).

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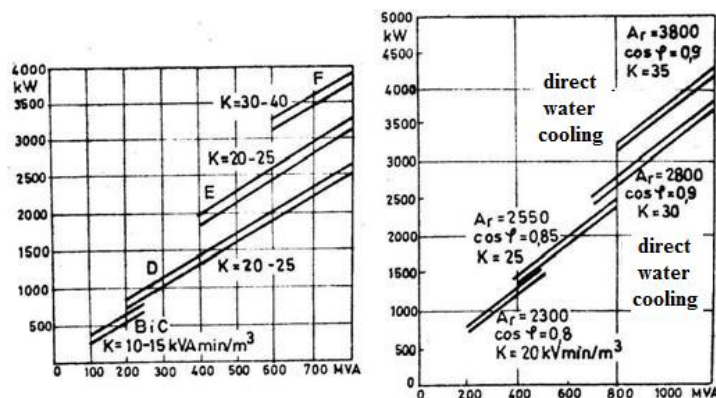


Fig. 1. Losses: a) stator winding, b) in the rotor of synchronous generators depending on power, type of cooling and efficiency factor K

It should be noted that when changing the mode of operation of the generator and uneven cooling on the rotor circumference due to various possible causes (errors in processing, asymmetry and unevenness of cooling openings and grooves for windings, dust, cracks, etc.) asymmetric and nonstationary temperature fields occur in different locations and has its change in time.

To these losses it should be added another 25% of losses in iron and 20% of losses due to friction in bearings and seals in relation to the losses in Fig. 1a. As most of the lost energy is converted into heat which must be removed from the generator by cooling, another 25% of energy is needed (according to diagram 1a) for cooling the generator [5]. Table 1 shows the type of cooling and the influence of the type of cooling on the efficiency factor K .

Fig. 2 shows a schematic cross-section of a synchronous generator with a cooling system. Cooling of rotor and stator grooves with windings is performed with coolant according to Table 1 and the appearance of rotor grooves and possible cooling methods is shown in Fig. 3.

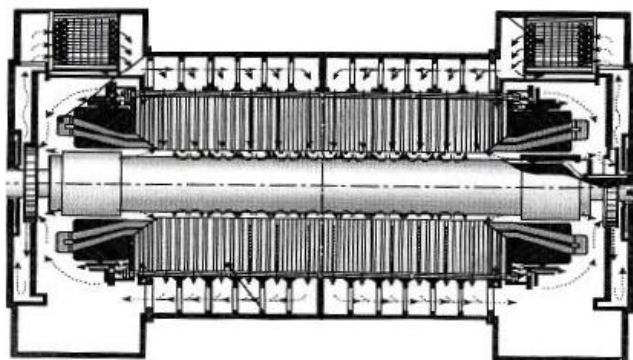


Fig. 2. Schematic display of turbogenerator with cooling system

Table 1.

Cooling		Refrigerant			Power [MW]	Efficiency factor K [kVA min/m ³]
stator	rotor	For stator windings	For rotor windings	For stator iron		
indirectly	indirectly	air	air	air	100	-
indirectly	indirectly	hydrogen	hydrogen	hydrogen	320	11
indirectly	directly	hydrogen	hydrogen	hydrogen	500	17,5
directly	directly	hydrogen	hydrogen	hydrogen	800	28
directly	directly	water	hydrogen	hydrogen	800	28
directly	directly	water	water	water, air	1200	42

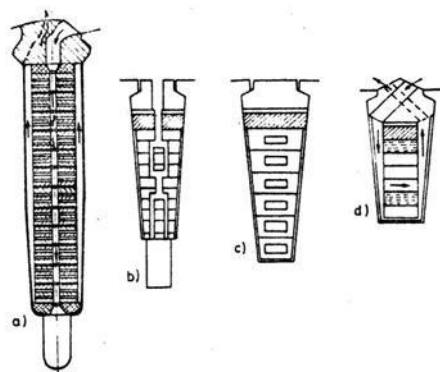


Fig.3. Schematic display of grooves shape and cooling type

Due to such, even very small temperature differences due to the high elasticity of the rotor and the large gap between the bearings, a deflection of the neutral line of the rotor is formed which has the direct consequence of smaller or larger vibrations on the generator bearings. It should be noted that a temperature difference of only 1.5°C can cause a deflection of the elastic rotor line of 0.1 mm. In companies where generators are produced, in such cases the empirical procedure of correcting the flow of coolant in certain zones of the rotor is performed.

3. MATHEMATICAL MODEL OF THE ROTOR IN THE EXISTENCE OF THERMAL DEFLECTION

The mathematical model is oriented towards the detection of this defect by means of the analysis of the sensor response given by the sensors placed in the bearings of the turbogenerator. The model is shown as an elastic rotor with concentric masses. Additional imbalance due to thermal deflection introduces additional inertial force into the model as an external force into the system. Consider a model of an elastic rotor with an asymmetric temperature field, Fig. 4. [9..12]

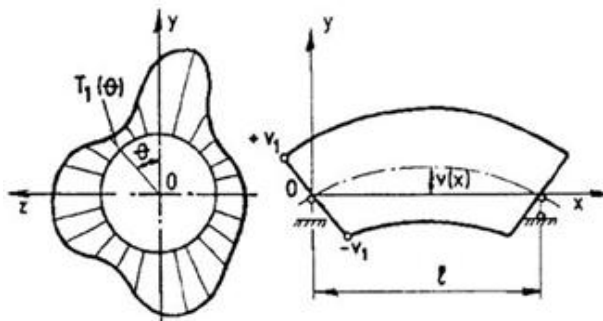


Fig. 4. Schematic representation of asymmetric temperature fields and deformable shape of the rotor

Based on the equations of mathematical physics, for the case of a real model of a rotors, the analytical model of a rotating temperature field can be described by differential equation (1):

$$\frac{1}{r} \frac{\partial}{\partial r} \left(\lambda_r \frac{\partial T}{\partial r} \right) + \frac{\lambda}{r^2} \frac{\partial^2 T}{\partial \theta^2} + \frac{\partial T}{\partial \theta} \omega \lambda + \frac{\partial^2 T}{\partial x^2} = \rho c \frac{\partial T}{\partial t} + q(r, \theta, x, t) \quad (1)$$

with boundary condition (Fig. 4):

$$T = T_1(\theta) \text{ for } r = R \text{ and } T(r, \theta) = \sum_{n=0}^{\infty} \left(\frac{r}{R} \right)^n (a_n \cos(n\theta) + b_n \sin(n\theta))$$

where temperature field in case of system defined by cylindrical coordinates. In order to define the deflection of the neutral line of the bent rotor, the Bernulli-Euler hypothesis which defines the thermoelastic moments and the deflection of the rotor was used.

In the case of stationary mode, the temperature deviations can be described by differential equations:

$$\frac{d^2V}{dx^2} = -\frac{\alpha a_1}{R}, \quad \frac{d^2V}{dx^2} = -\frac{\alpha b_1}{R} \quad (2)$$

where a_1 and b_1 are coefficients.

By including the boundary conditions that the deflections on the supports are equal to zero (Fig. 3), and by integrating the differential equations (3), we obtain the expression for the deflection of the neutral line due to thermal deflection:

$$e = \frac{\alpha l x}{d} \left(1 - \frac{x}{l} \right) \sqrt{a_1^2 + b_1^2} \quad (3)$$

It is clearly seen that the existence of thermal deflection in the mathematical sense creates an additional imbalance that differs in phase from mechanical, with amplitude and phase change depending on the temperature difference in the rotor structure. Logically, the conclusion follows that the image of rotor balance will change in the vibrational response (change of 1X component amplitude and phase).

The maximum deflection of the neutral rotor line is in the middle of the rotor and has the form:

$$e_{\max} = \frac{l(1 + \alpha \Delta T)}{2d \cdot \arcsin \frac{l\alpha \Delta T}{2}} (d - \sqrt{d^2 - (l\alpha \Delta T)^2}) \quad (4)$$

Now on the basis of known dependence it is possible to obtain a value for the inertial forces due to thermal imbalance.

By solving differential equation (1), values of temperatures when rotating the rotor are obtained, which affect the intensity of inertial forces. Thus, the inertial force due to thermal imbalance can be analytically expressed as:

$$F_{in} = \int m e(r, \theta, x, t, \omega, \Delta T) \omega^2 dx \quad (5)$$

The overall mathematical model of a multimodal rotor in a fluid environment can be presented and described as follows (Fig. 5):

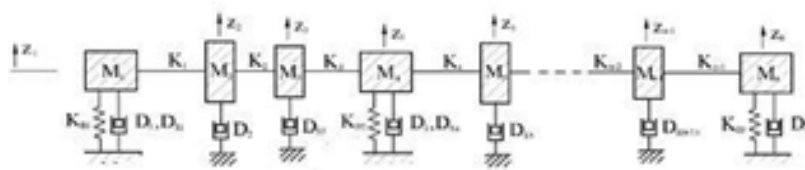


Fig. 5. Model of multimodal rotor with sliding bearings

Differential equations of motion of a multimodal rotor with sliding bearings, where in addition to the mechanical unbalance there is also a thermal unbalance can be written in the form:

$$M_i \ddot{z}_i + D_{si} \dot{z}_i + K_i (z_i - z_{i+1}) + K_{i-1} (z_i - z_{i-1}) + F_{\phi i} = m_i r_{mi} \omega^2 e^{j(\omega t + \epsilon_i)} + F_{in}^T \quad (6)$$

$$i = 1, \dots, n; z_1 - z_0 = 0; z_n - z_{n+1} = 0$$

4. DIAGNOSISING THE EXISTENCE OF THERMAL DEFLECTION ON THE TURBOGENERATOR ROTOR

The procedure of diagnosing and determining the influence of the thermal field on the dynamic behavior of the rotor will be presented on the example of monitoring and testing of turbogenerators in TPP Gacko. Real-time monitoring of turbogenerator condition showed that the 1X vibration component (amplitude and phase) of rotor vibrations measured near the generator bearings (Fig. 6) changed, which significantly affects the dynamic behavior of the turbogenerator as a whole.

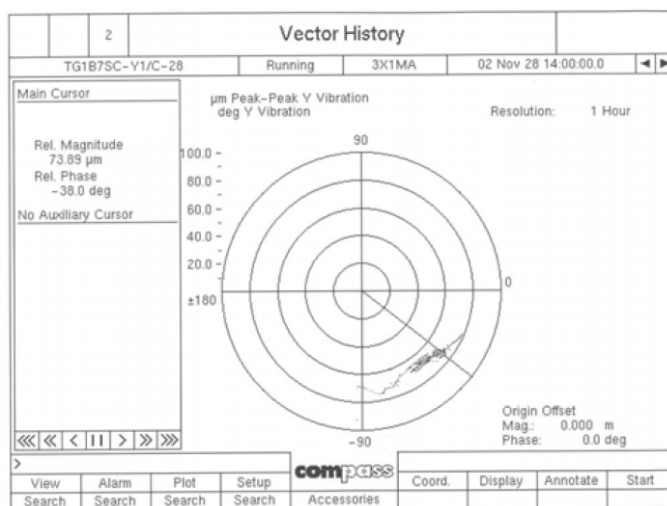


Fig. 6. Trend 1X vibration components of rotor vibrations near bearing L7 In this sense the asymmetry of the magnetic field and the existence of thermal deflection on the rotor part of the turbogenerator

The following test program was performed:

- At idle speed, at operating speed, without excitation current, perform vibration analysis of generator bearings and, if possible, rotor shafts on contact rings. Stabilization time regime should be not less than 10-15 min.
- Give the excitation current to the windings at idle rotor generator and record the vibration state of the bearings generator (determine the effect of magnetic asymmetry fields).
- Synchronize and connect the generator to the network. Raise the load gradually by staying at least at 3 stable operating modes up to full load. The stabilization time of each regime should be minimum 45 min. Record for each mode vibration characteristics of bearings.
- Switch off the drive and record at operating speed vibration characteristics of bearings

It should be noted that the installed online diagnostic system "COMPASS" version 6, which is used to monitor the operation of the unit, was used for testing purposes.

Table 2.

Analysis of rotor vibrations measured in the bearing area 6 and 7 $\left[\mu m\right]$																
	Time (19.11.02.)	Number of revolutio ns	Active power [MW]	Reactive power [Mvar]	Bearing 6						Bearing 7					
					V			H			V			H		
					1X	phase 1X	2X	1X	phase 1X	2X	1X	phase 1X	2X	1X	phase 1X	2X
1	17:03	3000	-	-	49,6	-95	22, 4	32,9	153	14	28,9	132	29,6	75,9	-84	37,9
2	17:47	3000	given initiative		55,2	-103	22, 5	37,8	144	13,9	37,5	116	30,9	75,7	-94	39,6
3	18:42	3000	73	17,7	57,8	-108	26, 2	37,8	148	18,3	28,8	113	30	73,1	-94	39,6
4	20:00	3000	274	86	49	-99	25, 4	34,8	158	21,2	29,3	118	30,4	72,7	-86,6	36,1
5	21:10	3000	282	91	56	-71	32	32	198	25	29	222	36	89	-10	41
6	21:30	3000	Initiative off		59	-82	31	35	179	25	21	191	34	75	-40	41

The test results are shown in Table 2. In Fig. 7 and 8 polar representations of the trend of 1X rotor vibration vectors measured near generator bearings over time are given. The analysis of the results from the polar diagrams clearly shows the existence of thermal imbalance, as well as the influence of the magnetic field asymmetry on the dynamic behavior of the aggregate as a whole. Vectors of 1X rotor vibrations in different operating modes of the unit are shown: A1-vector of 1X rotor vibrations by going to operating speed, A2-vector 1X with given excitation, A3- vector 1X after synchronization, A4-vector 1X after increase of excitation current, ie reactive power from 17.7 to 91 MW and system stabilization (45 min), A5-vector 1X after disconnection.

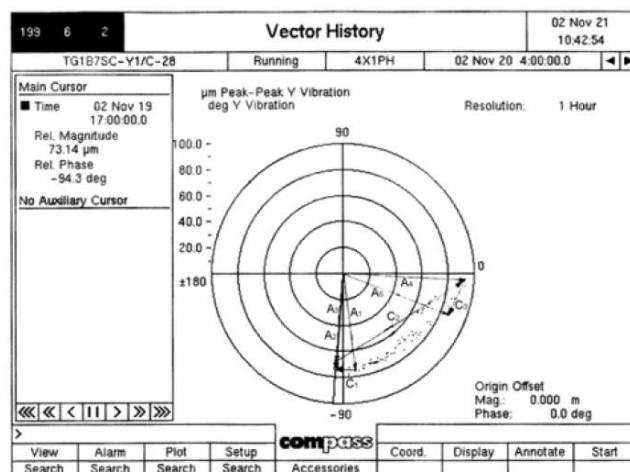


Fig. 7. Polar diagram of 1X vibration component in bearing 6

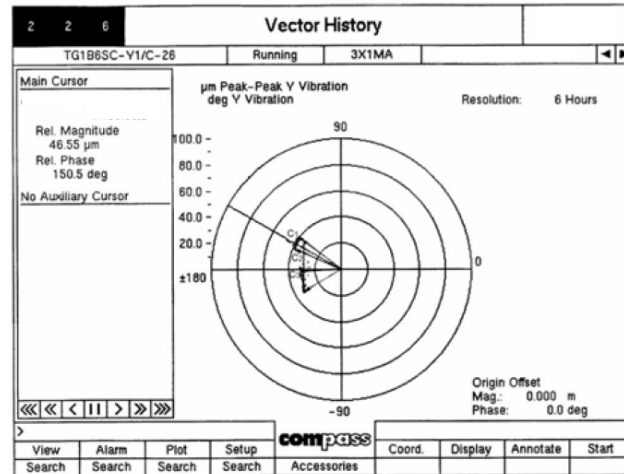


Fig. 8. Polar diagram of 1X vibration component in bearing 7

The influence of thermal imbalance and magnetic field asymmetry on the overall dynamic behavior of the rotor can now be determined. The determined test values are: C1-vector of asymmetry of the magnetic field in the cold state of the rotor, C2-vector of thermal imbalance with 17.7 to 91 MVar, C3-vector of thermal imbalance in the heated rotor. It can be seen that the thermal imbalance at a reactive power of 91 MVar is the same in intensity as the residual mechanical imbalance on the rotor, but that it differs in phase by 45°. Following the trend of further changes in the thermal imbalance vector, it can be concluded that by further increasing the reactive power, the total unbalance vector (vector 1X of rotor vibrations) would be inadmissible for further operation of turbogenerators (especially on bearing 7). So, in the process of further exploitation of the block the reactive power should be limited.

5. CONCLUSION

This paper deals with the problem of thermal unbalance of synchronous generators. Emphasis is placed on the timely diagnosis of this defect by analyzing vibrations and determining its impact on the dynamic failure of the system as a whole. It has been shown that the best indicator of the existence of thermal deflection is the monitoring of 1X vibration vectors in the production process. The existence of thermal deflection causes a change in the amplitude and phase of this vector. However, such a change in the 1X vibration vector can be a consequence of other defects in the system (looseness of rotating parts, contact of the rotor on the stator due to increased mechanical deflection, propagation of cracks in the rotor or foundations, etc.). Thus, to determine the existence of this defect, a specific test program should be performed (Chapter 3). After determining the existence of thermal deflection, vibrodiagnostic analysis can determine its effect on the overall dynamic behavior of the system and clearly predict side effects. If it is determined that in the production process the process parameters will be in the zone where they will cause asymmetry of the temperature field of the rotor that causes inadmissible deflection, it must be resorted to its elimination. Note that the elimination of this problem cannot be solved by mechanical correction of the rotating masses of the rotor, but the intervention of thermal balancing must be resorted to.

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