NUMERICAL AND EXSPERIMENTAL ASPECTS OF THERMALLY INDUCED VIBRATION IN REAL ROTORS

by

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Temperature fields in electric energy generators may lead to mechanical dissbalance of an already balanced rotor. The author collected information in a number of steam power plants and confirmed the existence of the problem. This paper presents the specific case of thermal deformation of the rotor, caused by an asymmetrical temperature field in scale of rotor. On the grounds of the relevant physical aspects, we proposed a mathematical model identifying fields in a turbo generator rotor and suggest the optimum control by which the unwanted effects are eliminated.

Key words: *turbogenerator, vibration, temperature, field, deformation, simulation*

Introduction

Electric energy is produced by multiple transformations of energy resources from one form to another by turbo generator. The energy generation is achieved in a turbo generator by complex electric, electromagnetic, mechanical, temperature, flow, and other processes.

There are various causes for thermal imbalance of a turbo generator rotor *i. e*. rotor winding shorted turns, negative sequence heating, blocked rotor-winding ventilation ducts, non-uniform friction forces acting around the circumference between rotor coils, rotor slot wedges, and rotor forging causing the rotor bent because the axial thermal expansion between the rotor winding copper and the rotor introduces axial bending forces, *etc*.

A specific case of thermal deformation of the rotor, caused by an asymmetrical temperature field in scale of rotor is presented in this paper. A rotor can also bend with a symmetrical temperature distribution if in the case that the friction forces between copper winding and the main body of the rotor are unevenly distributed. Several generator manufacturers perform so-called thermal sensitivity runs in the spin pit to check that rotors are delivered with acceptable thermal imbalance behaviour. All rotors show a certain thermal imbalance when the rotor field current is switched on at rated speed and the cooper winding expands more than the rotor steel of the rotor main body. During these spin pit tests there is no stator.

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During the process, due to different causes, the rotor temperature increases whilst the external surface cools, causing a thermal imbalance and an increase in the vibration level. Due to temperature imbalance and different isolation thickness, impurity, layers, *etc*., an asymmetric distribution of temperature field is formed, which is related to the temperature characteristics of different materials and causes thermo elastic deformation, inertial forces and kinetic pressure on the bearings, [1-4].

Thermally induced vibrations due to rub in real rotors are investigated in papers [5, 6]. The model commonly called ''spiral vibrations'' or ''thermally induced vibrations'' of full size rotors, has been introduced for calculating rub conditions and rotor-to-stator rub induced vibrational behavior. The capability of the proposed method to define the operation of real machines has been validated on the experimental case of 350 MW generator.

The authors of this paper presented an adjustment method which helps decrease temperature asymmetry and vibration. Through the process of analyzing the type of thermal cause of the vibration phenomena and diagnosing the rotor condition with vibration measuring equipment, it was possible to predict the cause of the thermal vibration and decrease its stroke. Using this method, a satisfactory vibration level was achieved for 600 MW to 700 MW class turbine generators.

The thermal effects of rotor-to-stator rub, as well as their influence on the vibrational response of the rotor are discussed in paper [7]. A specially developed transformation is applied to the system model, which contains discontinuities, and an averaging technique is then used to analyze the stability of the different rotor motion resonance regimes. These regimes are further used to calculate the heat generated during rotor-to-stator contact, as a function of thermal conditions and rotor thermal bow modal parameters. The calculated heat input is used as a boundary condition for the rotor heat transfer problem. The latter is treated as quasi-static, which allows the application of an asymptotic method to the problem. The solution (at its first approximation) is used to adjust the rotor thermal bow value. As a result of this calculation, an ordinary differential equation with complex variables is obtained for the thermal bow, and is derived for the thermal bow, and is investigated from the stability standpoint.

Thermally induced vibration in a rotor-active magnetic bearing (AMB) system was reported in the paper [8, 9]. It was clarified that the vibration was caused by thermal deformation coming from uneven iron loss distribution. The deformation and vibration mechanisms were demonstrated by varying imbalance control law. A model which included thermal dynamics was introduced and the stability condition was derived. It was revealed that the instability depends on the rotational speed, sensitivity of the thermal imbalance response, the phase relation between magnetic force and thermal deformation, *etc*. Finally, a method for preventing thermally induced vibration was discussed.

The authors discussed thermally induced vibration on a magnetic bearing rotor and identified mechanism that is related to the interaction between a bearing force and a thermal bending by the loss. A simple model which includes the interaction and derives an instability criterion was made. Additionally, it was discussed how to avoid this unstable thermal bending.

The discussion in paper [10] has been confirmed in the two-pole gas-turbine-driven generators. The thermal sensitivity problem due to vent holes blockage, which is took place on the two generators, shows that the risk of thermal sensitivity of the rotor can be controlled with appropriate monitoring and testing. This thermal sensitivity contributes to the reversible vibration; nevertheless, the generator is still able to operate if the vibration does not exceed the limit. Otherwise, the problem needs to be mitigated.

As previously stated, the behaviour is not stationary over time and space, and it is not possible to solve the problem through mechanical balance. Instead it is necessary to include some thermal intervention. It is possible to intervene based on a detailed analysis of the causes within the complex system of temperature increase and decrease in the turbo generator.

Investigation of vibrations caused by temperature

There are many causes of high vibration in a generator field. The most common are mechanical imbalance, thermal sensitivity, misalignment and bearing degradation. Other causes are rubbing, bent overhangs, rotor stiffness asymmetry, out-of-round journals and other design deviations caused by abnormal in-service operation. Each of these causes has dominated frequency and characteristic response. The cause of vibration can be diagnosed by thoroughly analyzing the vibration data.

A thermally sensitive rotor is characterized by a once-per-revolution frequency response signature due to a change in the rotor balance arising from the rotor bow. If the total vibration level of the field stays within acceptable limits, the field is not considered "thermally sensitive". Vibration performance is frequently plotted on a polar chart, because vibration is characterized by amplitude and phase angle [11].

If the frequency of vibration stays within 2 or 3 ms, or whatever is chosen as an acceptable vibration level, the vibration is not considered to be a problem. This is true even if the phase angle changes and the vibration move around the interior of this circle.

The change in vibration and phase angle within the polar plot from the starting operating point to the end operating point is called the thermal vector. In general (not always), the rotor vibration increases along with increases of the current of the rotor field, *e. g.* fig 1.

Mathematical model

As an initial relation, the basic function of the rotary turbo generator system, according to the notations of fig. 1 is:

$$T = f\ (r, \varphi, z, t) \tag{1}$$

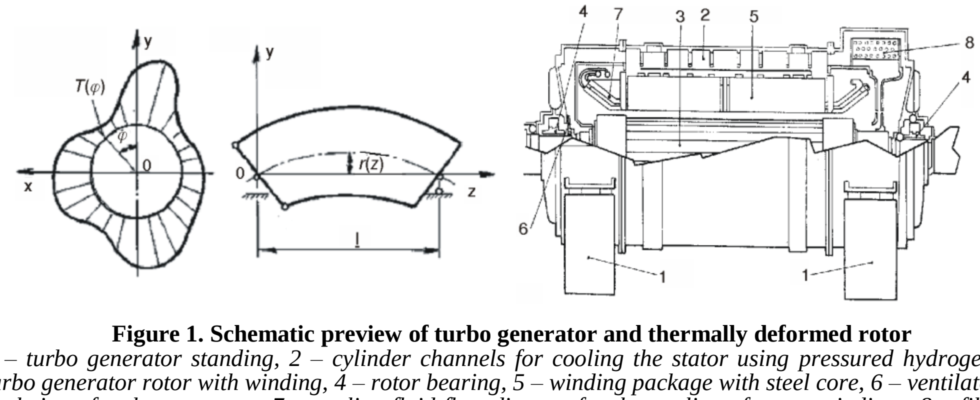

Figure 1. Schematic preview of turbo generator and thermally deformed rotor
1 – turbo generator standing, 2 – cylinder channels for cooling the stator using pressured hydrogen, 3 – turbo generator rotor with winding, 4 – rotor bearing, 5 – winding package with steel core, 6 – ventilator and seal ring of turbo generator, 7 – cooling fluid flow director for the cooling of stator windings, 8 – filter for purification of cooling fluid

For calculation of temperature values T [11-15] an original analytical model is formed, on the bases of mathematical physics. The model is modified as well as applied in the case of real turbo generator model.

A differential equation gives the temperature values at the rotor rim with synchrony speed ω during simultaneous heating and cooling, which effects the inertial forces.

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

$$F_{\mathrm{in}}=\int me(r\varphi,z,t,\omega,\Delta T)\,\omega^2 \mathrm{d}z \tag{3}$$

Original differential equations analytically defines thermal energy distribution on the rim of the rotor [13, 14]. The differential equations are solved and integrated by generating the functional variation that meets the definition of boundary conditions. Boundary conditions are of constant or variable character. For example, the specific elastic resistance of copper changes with temperature. The functional is generated based on a differential equation of thermal energy distribution. The heat propagation differential equation is numerically calculated and integrated using the finite element method. The real structure of the rotor is discretized with a finite element-type three-sided prism. By differentiating the functional at temperatures of the node points of the finite element, and matching these equations with zero, a system of algebraic linear equations is obtained. The number of these equations in the system is equal to the number of node points of the finite elements in the discretized structure of the rotor. The solution to the linear algebraic equations provides the values of temperature at the node points. The resulting asymmetric temperature distribution on the rotor causes the formation of thermo elastic deformation of the rotor in the form of bending. In addition, the mass centre of the rotor is performed outside the rotational axis of the rotor and of the bearings, which causes inertial centrifugal force. An increase in inertial forces leads to an increase in the kinetic pressure in the bearings. The increase in kinetic pressure directly increases the level of amplitude of the rotor vibration in the bearings.

The boundary conditions are defined at the edges of the finite elements where the heat is convected with the coefficients α_{i}. In fact, coefficients λ_{i} and α_{i} represent the boundary conditions themselves. Apart from that, boundary conditions are also the amount of heat generated in the finite elements and the finite element starting temperature of the rotor and the surrounding parts of the stator.

By original analysis the application of the Saint-Venant principle to the local loads and thermal stresses yields to the analytical relation for rotor deformation in the x- and y-directions [13, 14]:

$$\begin{aligned} a_{\mathrm{y}} &= \frac{\mathrm{d}^2 u_{\mathrm{y}}}{\mathrm{d}z^2}=-\frac{1}{E}\left(\frac{I_{\mathrm{y}}M_{\mathrm{Tx}}-I_{\mathrm{xx}}M_{\mathrm{Ty}}}{I_x I_y - I_{xy}^2}\right) \\ a_{\mathrm{x}} &= \frac{\mathrm{d}^2 u_{\mathrm{x}}}{\mathrm{d}z^2}=-\frac{1}{E}\left(\frac{I_{\mathrm{x}}M_{\mathrm{Ty}}-I_{\mathrm{xy}}M_{\mathrm{Tx}}}{I_{\mathrm{x}} I_{\mathrm{y}} - I_{\mathrm{xy}}^2}\right) \end{aligned} \tag{4}$$

and in the case of the principal axes $I_{\mathrm{xy}} = 0$ the simplified form is derived.

The total deformation of the rotor is defined by relation given in ethe quation:

$$e_{\mathrm{T}} = \frac{180L}{2\pi \mathrm{arctg}\dfrac{L\beta\Delta T}{D}}\left(1 - \frac{D}{\sqrt{D^2 + (L\beta\Delta T)^2}}\right) \tag{5}$$

In this analysis, the values of thermo elastic moments are in the form of equation:

$$\begin{aligned} M_{\mathrm{Tx}} &= E\int \beta T_{\mathrm{y}}\,\mathrm{d}A \\ M_{\mathrm{Ty}} &= E\int \beta T_{\mathrm{x}}\,\mathrm{d}A \end{aligned} \tag{6}$$

To obtain the numerical data it is necessary to start from the definition of boundary conditions. Boundary conditions include forms for total power and for total losses of turbo generator (cooling, friction, losses in iron due to hysteresis and Foucault current, losses in copper). On the basis of defined losses [15], an equation for heat generated in the copper conductor can be derived:

$$Q_{\mathrm{Cu}} = \int (\Delta P_{\mathrm{Cu0}} + I^2 R_0 \beta T_{\mathrm{Cu}})\mathrm{d}t \tag{7}$$

which causes the rotor to heat (partially transmitted to cooling fluid, and partially transmitted to the iron core through the insulation).

The definition of boundary conditions, the calculation of turbo generator power, calculation of losses in copper and iron, and their introduction into a numerical analysis allows for the calculation of temperature, thermo elastic deformations, inertial forces and kinetic pressure on the bearings of the rotor, with parallel thermo-mechanical causes. In a separate section of the paper [14], an analysis for kinetic-pressure in the case of a two-pole turbo generator and rotor with double stiffness is derived. A schematic representation of the model of dynamic inertial forces on the rotor is shown in fig. 2.

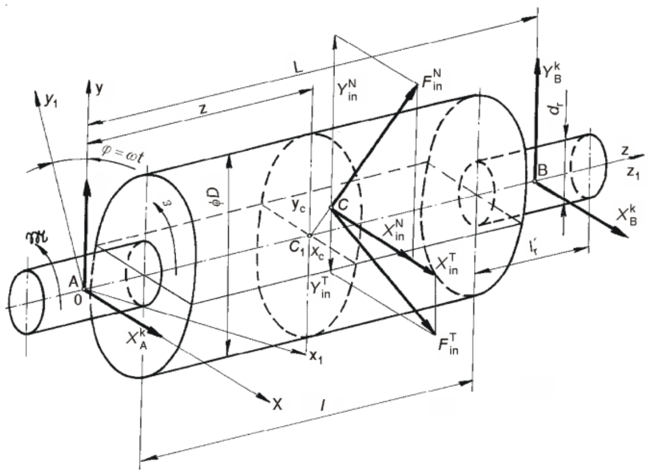

Figure 2. Dynamic model of turbo generator rotor

In this case, it is extremely important for the effect of the total displacement of the mass centre of the rotor from the rotating axis to contain the mechanical eccentricity e_{T} – error in mechanical treatment, dynamic deflection e_{d}, and temperature deflection $u = f(a_{\mathrm{x}}, a_{\mathrm{y}})$.

By performing the appropriate dynamic analysis, [14], the values for kinetic pressure on the bearings for the synchronous generator operation are derived (ω = const, and $\dot{\omega}$ = 0). The kinetic pressure on the bearings A and B is presented in the equations:

$$X_{\mathrm{A}}^{k} = \omega^2\left(\frac{Jxz}{L} - mx_{\mathrm{c}}\right), \quad X_{\mathrm{B}}^{k} = -\frac{1}{L}\omega^2 Jxz \tag{8}$$

Starting from the original analytical model (2) for the rotary temperature field, by applying the finite element method in the relevant expression for functional χ, it is possible to perform more precise calculation of the temperature field for the 350 MW turbo generator. This method enables numerical analysis of more complex turbo generator models with all complex structure and elements. For this, it is necessary to prepare data, which define the boundary conditions.

Equations (1) to (6) can also be used if the friction forces between copper winding and rotor are unevenly distributed. Thermal imbalance may also occur with a uniform distribution of generated heat, but with asymmetric cooling.

The complex structure of the rotor determines the method of discretization of its structure with the finite element type of three-sided prism. Two pole turbo generator rotors have two symmetrically placed systems each with axial channels 38 where tubular copper conductors with insulation are installed. Each electrical pole of the rotor contains 38 axial channels in which there are packages of five hollow copper insulated conductors. Cooling process is enabled using hydrogen at a pressure of 6 bar through every hollow copper conductor. The rotor structure modelling method of discretization of the rotor structure includes all its parts. The materials from which the rotor is made are: iron, copper, insulation, hydrogen as a cooling fluid and material for the simulation of thermal disruptions in the form of thin copper laminations in the conductors. The stated materials are: specific heat, density, coefficient of thermal conductivity by conduction, models of elasticity, coefficient of friction and coefficient of linear expansion and they represent input data. The generated heat in the copper wire is removed by cooling with the flow of hydrogen at a pressure of 6 bar. This structure of the rotor made its discretization and 14.600 finite-elements-type three-sided prisms with the 87600 node points were obtained.

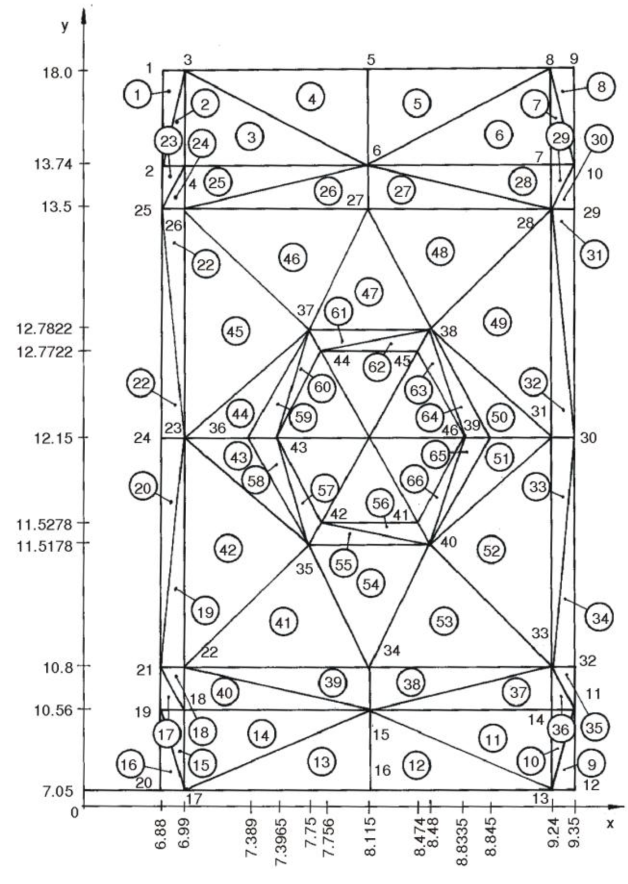

Figure 3. Part of the structure discretization with the insulation and contamination layer in the finite element triangle prismatic form

Figure 3 shows a solution for the discretization of the copper tube conductor with the insulation and a contamination simulation layer in the finite element in the form of three sided prism. This is achieved through the introduction of the z co-ordinate and the presentation of the flow diagram segment of originally developed programs for numerical calculation of temperature values within the corresponding node points and finite elements.

The contamination layer represent a thermal disturbance which disturbs uniform heat convection in simultaneous heating and cooling of the rotor. Difficulties arise from the increase of specific electrical resistance and total electrical resistance with a temperature increase in the copper conductor and an increase in the total generated temperature.

Preparing and solving the set of analytical models implies the formation an input data file and boundary conditions for all processes and elements of the rotation and simultaneous heating and cooling. This process is preceded by the detailed discretization of the rotor structure with finite elements. On the other hand, according to the analytical model of temperature distribution on the rotor rim and the established functional form, we developed a very complex system metric calculation for which a computer program that has a system of subprograms for calculation of temperature was developed. First, flow diagram was used for pending system for the integration of the differential equation, and then, a complex computer program with subprograms was developed. The programs were developed using Quick Basic computer language. The program has been tested and verified on IBM computer installation. After filling the file with input data and after performing numerical calculation of the temperature on the printer for each node point of the final element, real values of temperature-time relation are obtained. This selection of the results was performed using a subprogram in C professional computer language. Using the software package MATHCAD, numerical data are processed and the empirical time-temperature dependence is obtained, which is printed in the form of diagrams. The diagrams refer to the period from the start until the stationary temperature condition of turbo generator. The calculation was performed with 1100 time increments at 22.5 s and so after 6.875 hours, a stationary temperature regime was reached.

There are number of researchers who carried out similar numerical research: Bachschrid [5], Goldman [6], Watanabe [7], Takahashi [8], Thikhimporn [9], Bialik [11], Krok [15], Ota [16], Babic [17], and Manea [18]. The above mentioned numerical studies were performed according to their own mathematical models and with its own software programs. The results were different in final form, but all met the established requirements. However, no one of these studies, in their analytical models, include synchronous speed of the rotor. The results of these numerical researches define calculations of temperature on the rotor without its angular velocity.

Obtained results

Calculation of temperature values of the node points of the discretized rotor structure is carried out by introduction of 1200 increments each with 22.5 s during the simulation of thermal process from the starting point of rotor operation in order to reach stationary thermal regime, while the values at the end of the previous interval are used as starting values for the next interval.

With this research, by using powerful calculator, we can obtain the functions of temperature increase in rotor structure, which is shown in fig. 4.

Stated functions are presented in diagrams, tables, and through analytical forms, using the program MATHCAD. By simulation of different values of thermal disturbances asymmetrically placed on the rotor surface we can determine the values of stationary temperature, which are causing the thermo elastic stresses, rotor deformation, inertial forces and kinematics pressure in the bearings and vibrations.

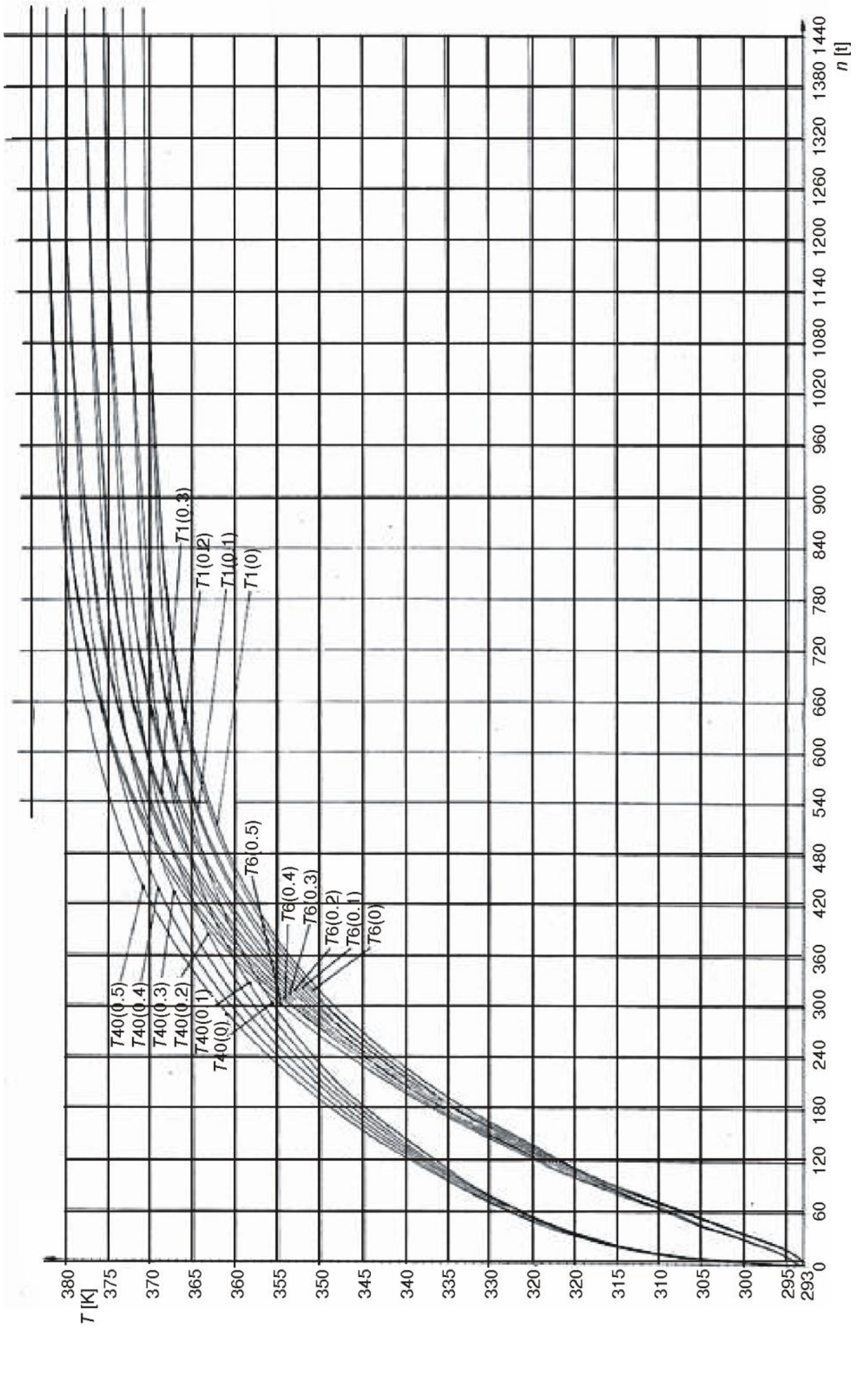

Figure 4. The temperature – time relation

Figures 5 and 6 present the results of obtained diagram dependence of deformation e_T and inertial forces as a function of thermal disturbances.

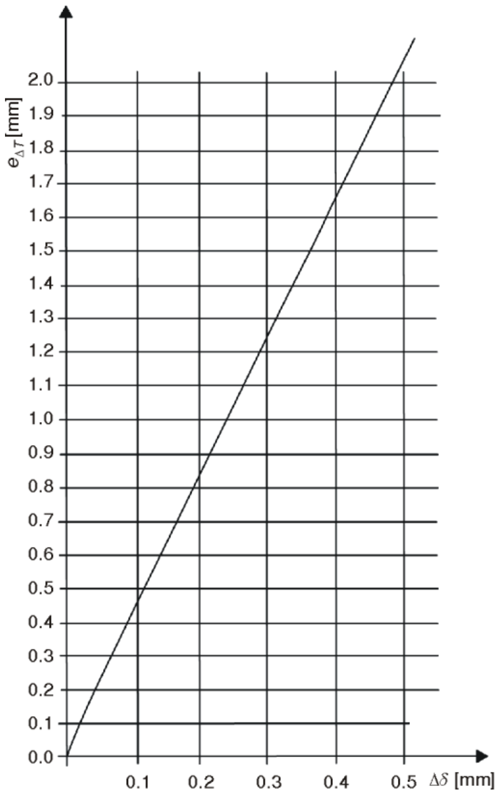

Figure 5. Thermo elastic deflection

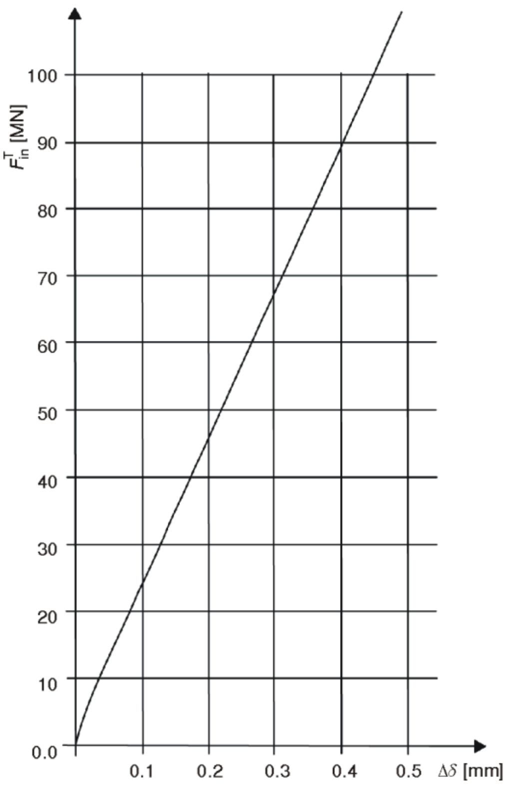

Figure 6. Change of inertial forces of thermal disturbances in the system for rotor cooling

Detailed data analysis and the original version of dynamic stability maps are generated (fig. 7.) and for the first time, quality criteria of thermo-dynamic operation of turbo generator rotor are introduced in literature. These maps represent the relation of vibrations for 350 MW turbo generator and thermal disturbances in the system of simultaneous heating and cooling, with precisely defined regions for fine, acceptable, boundary and restricted work..

The results obtained by numerical calculation are real and match the information on the behaviour of real turbo generator exploitation. Simulation period to achieve stationary temperature is 6.785 hours and in practice, 7 hours, which is a numerical error of 3.07%. Obtained stationary temperatures are 385 K, while in practice, the actual temperature is 380 K, which is a numerical error of 1.316%. When calculating the values of thermo elastic deformation, for thermal disturbance of 0.3 mm, deflection of the rotor of 1.2 mm and the inertial force of 67 MN was obtained. For critical cases of simulation of the thermal disturbance of 0.5 mm the deflection of 2 mm and the inertial force of the rotor of 110 MN were obtained. It is impossible to get the numerical value for the error because the calculation is carried out through simulating thermal disturbances, although detailed analysis of a hypothetical situation determined possible numerical error with low values. These data show

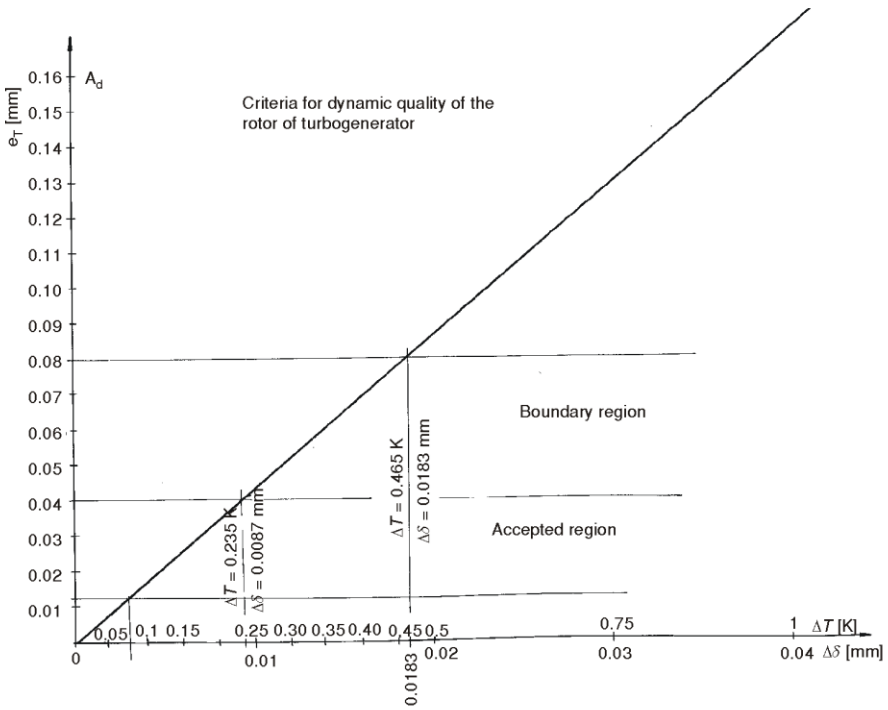

Figure 7. Original maps for dynamic thermal stability

the correctness of the numerical calculation carried out with the correct mathematical models, computer programs, included data and boundary conditions.

So far, in practice, ISO standard defined dynamic stillness of rotating machines caused by mechanical disturbances. This standard defined the amplitude of vibration on the bearings for different number of rotations of the rotor vibration for good, acceptable, boundary, and unacceptable operation. Based on the research, the authors designed the original map of thermo-dynamic stability of rotating machinery in the form of diagrams of dependency. Here, the abscissa is a double relation: the asymmetry of temperature field on the rotor rim ΔT caused by thermal disturbances $\Delta\delta$. There are values of vibration amplitudes at the bearings on the ordinate. Calculation which was carried out in this paper enabled us to obtain temperatures in node points of the finite element structure of the rotor for the defined thermal disturbances. These values define the temperature asymmetry on the rotor, which through thermal elastic deformation of the rotor and the inertial forces, causes a change in the amplitude of vibration of the rotor bearings. The dependency diagram was obtained by calculation of the values of vibration amplitude, depending on the temperature asymmetry or on the thermal disturbances. Thermal disturbance are presented by the thickness of impurities copper conductors, which prevents rapid heat dissipation during cooling. This map of thermo-dynamic stability consists of areas of good, acceptable, boundary, and unacceptable operation of rotary machines. According to the developed criteria for assessing the thermal dynamic stillness, thermal disorder of 0.08 K or 0,003 mm provides a good upper limit of the rotor operation. The thermal disorder of 0.235 K or 0.0087 mm defines the upper limit of acceptable operation. Also, thermal disruption of the ΔT = 0.0465 K, *i. e.*, $\Delta\delta$ = 0.0183 mm

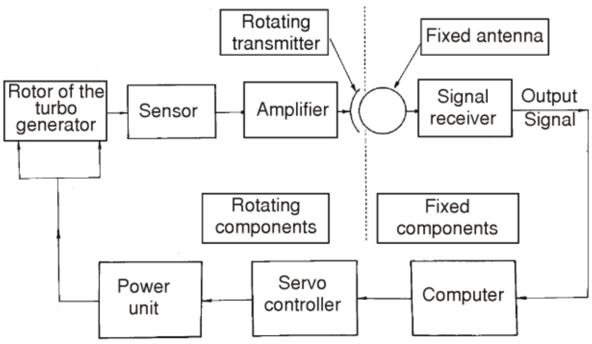

Figure 8. The system for automatic thermal balances of turbo generator

defines the upper limit of the boundary operation, when the machine must be stopped in order to prevent accidents.

Further research gives the original solution of the system for automatic temperature balance of an already dynamically balanced rotor which is shown in fig. 8.

The original system for the automatic thermal balancing of the rotor will be installed in new turbo generators. The system will identify thermal disorder and automatically execute its removal. This system will automatically adjust the asymmetric temperature field with the same temperatures on the rotor rim and thus make the thermal balance. Sensors are positioned on the rotor along with amplifiers and transmitters, sending radio signals about the value of temperatures of the rotor rim. Those signals are transmitted to the computer where information is then processed and sent to servo controls and power units. Servo sub-system performs correction of cooling the turbo generator by hydrogen under pressure. Correction is performed by changing the fluid flow of cooling fluid of the rotor heated places, which results in intensive cooling of more heated places, whereas, less fluid flow results cools down less heated places.

Experimental investigation of thermally originated vibrations

Author of this paper carried out experimental investigations of thermally originated vibrations in the power plant on a turbo generator Dolmel in Wroclaw Poland. In the operation of the above-mentioned turbo generator, it was noticed that the level of vibration increased at the vertical plain on the bearing No. 4 of turbo generator, which is placed close to the high pressure turbine. As the mechanical – dynamical balancing could not reduce the level of vibration, experimental investigations of regime and parameters were undertaken and their influence on the temperature field and the level of vibration was assessed. The reason for these investigations was that the increased level of vibrations was caused by temperature imbalance in the rotor system of the turbo generator. With such defined aims, the investigation of the turbo generator, schematically shown in fig. 1, was made. The amplitudes of vibrations were measured at the bearings in the horizontal, vertical and axial directions, with special attention to the bearing No. 4. The measurements of vibrations were made by a multichannel measurement installation. The changes in vibration amplitudes A4H, A4V, and A4A were recorded in a specified period and after changing the reactive power of the turbo generator. In this way, the change of reactive power influences on the temperature of formed asymmetric fields at synchronous speed of the rotor. The rotor with the asymmetric temperature fields is cooled by hydrogen. Cooling the stator with iron segments and copper conductors is also done by hydrogen under pressure. In the investigation, the critical amplitudes of vibrations were determined which, in the long run, would cause damage of the rotor due to significant increase in vibration over time.

Results and discussion

The results of experimental investigations are shown in fig. 9, with a decrease of the reactive power of turbo generator from 40 MVar to 0,1 MVar in a very short time, which at the same time caused a decrease in the temperature field asymmetry of the turbo generator rotor.

The active measured power was P = 43 MW. In the time interval of t = 35 minutes, the amplitude of vibrations in the horizontal direction A4H, changed from 40 μm to 61.5 μm, at the constant rotor speed of n = 3000 rpm. This effect is caused by the temperature imbalances which produces non-stationary asymmetric temperature filed at the rim of the turbo generator rotor. The change of parameters in the conductor with insulation and in the cooling system, during simultaneous process of heating and cooling, causes thermal asymmetry at the rotor rim, which is manifested as a change of dynamic behaviour and rapid increase in amplitude of turbo generator vibrations.

Then, a fast change of reactive power is made by its decrease from 40 MVar to 0,1 MVar. This change caused a decrease in vibration amplitude on the bearing A4H from 61.5 μm to 35 μm. Decrease in reactive power is followed by current (I) decrease from 1,37 kA to 0,82 kA. Also, the test showed the decrease of copper stator conductors T_{Cu} from 54 °C to 47.5 °C. The temperature of stator iron laminations T_{Fe} changed from 51.5 °C to 50 °C, indicating that the vibration level of turbo generator depends on the formed temperature fields and consequent thermal asymmetry of the rotor system.

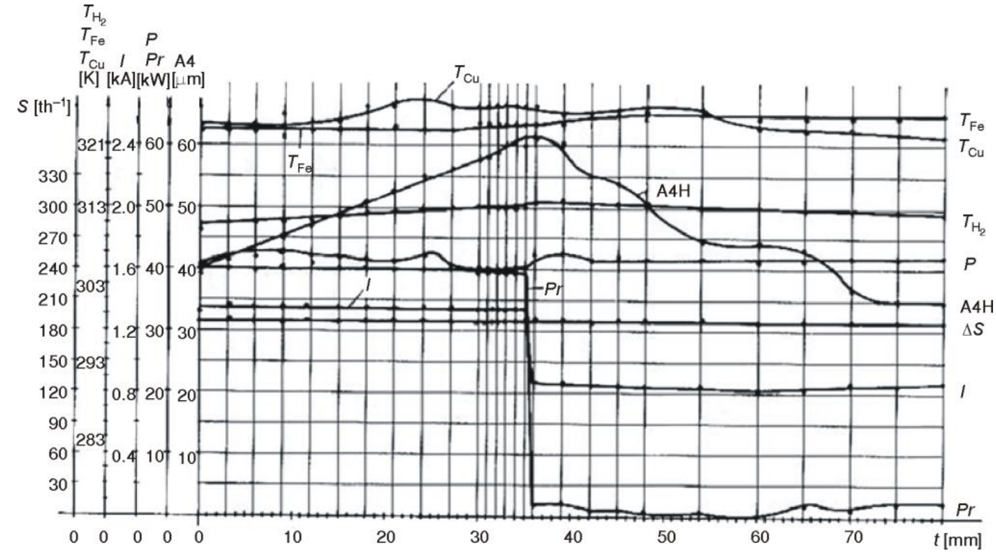

Figure 9. The testing operational parameters of the turbo generator

The tests showed that the rotor speed did not change n = 3000 rpm, and that the steam consumption was S = 190 to 200 t/h. Temperature was measured by thermo-elements located in the turbo generator system.

Based on the analysis of test results, it was concluded that for a constant rotor speed, the amplitude of vibrations decreases with the decrease of the reactive power Pr, which is followed by temperature decrease and decrease of thermal asymmetry on the rotor. This conclusion can be explained by the change of thermal asymmetry on the turbo generator rotor which causes the change of inertial forces influence on the dynamic behaviour and the level of amplitude vibrations. Thermal imbalance may also be caused by a non-uniform axial friction force distribution around the circumference.

The physicality of the conducted original numerical calculation and analysis of thermal asymmetry on the rotor was verified by experimental studies performed on a real turbo generator. Obtained numerical values correspond to real data of the behaviour of turbo generator and exploitation which defines the relation between numerical aspects and experimental research. The link is further confirmed by data on time-temperatures relation in reaching the stationary regime, which represent the result of the numerical calculation and experimental research. Also, during exploitation occurrence of thermal disturbance was verified during simultaneous heating and cooling of the turbo generator. Numerical investigations are associated with the practice and can protect the real turbo generators of the possible damages. Some twenty years ago, in the thermal power plant Obrenovac, Serbia, there was a turbo generator failure due to thermal disorder on the rotor rim. On the other hand, the experimental research conducted on a real turbo-generator has been confirmed by the occurrence of asymmetric temperature field, thus the connection of numerical and experimental investigations has been verified

Conclusions

Long-term, broad and deep research gives the results which indicate that very small thermal disturbances in the condition of simultaneous heating and cooling of rotor turbo generator structure cause the forming asymmetric temperature field and significant degradation of dynamic stability. These disturbances cannot be predicted in time and space, and they can cause rotor damage and prevent further exploitation of turbo generators. Model, which is presented in this paper, can be used for all cases of temperature distribution.

The author's patent will be introduced in the further development of new generations of turbo generators. The patent will include original system for automatic solving of thermal disturbances.

Nomenclature

A – cross section area, [m^2]
a_x – thermo elastic deformation of the rotor bearing in horizontal plane, [mm]
a_y – thermo elastic deformation of the rotor bearing in vertical plane, [mm]
C – total amount of generated heat, [J]
c – specific heat, [Jkg^{-1}K^{-1}]
D – diameter, [m]
E – module of elasticity, [Nm^{-2}]
e – eccentricity of centre of gravity from the rotary a_x and a_y (thermal deformation), [m]
F_{in} – inertial forces, [N]
I – electric current intensity, [A]
I_x, I_y, I_{xy} – moment of inertia, [m^4]
J_{xz} – moment of mass inertia, [kgm^2]
L – rotor length, [m]
M_{Tx}, M_{Ty} – thermal moment, [Nm]
m – mass, [kg]
ΔP_{cu} – energy loss in copper, [W]
Q_{Cu} – quantity of generate copper, [J]
q – additional amount of heat caused by friction and other impacts during exploitation of turbo generation, [J]
R – electrical resistance, [Ω]
T – temperature, [K]
t – time, [s]
r, φ, z – cylindrical co-ordinates, [m, °, m]
u_x, u_y – rotor deformation in x- and y-direction, [m]
X_A^k – horizontal component of kinetic force in the bearing A, [N]
X_B^k – horizontal component of kinetic force in the bearing B, [N]
x_c – horizontal deviation of the rotor mass center from the bearing axis, [mm]

Greek symbols

α – coefficient of heat convection, [$Jm^{-1}s^{-1}K^{-1}$]
β – linear displacement coefficient, [K^{-1}]
$\Delta\delta$ – thermal disturbances, [m]
λ – coefficient of temperature transfer, [$Jm^{-1}s^{-1}K^{-1}$]
ω – angular velocity, [rad s^{-1}]
ρ – density, [kgm^{-3}]

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