CONFERENCE 2015 – MANCHESTER
Abstract Transactions, SMiRT-23
Manchester, United Kingdom – August 10–14, 2015
Paper ID 308

Simulation and experimental study of turbogenerator stator vibrations

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ABSTRACT

The objective of this study is to give an understanding of how natural frequencies and mode shapes of the end winding components in high-power turbogenerators are determined. This study represents results of the analysis performed for various calculation methods and highlights general problems that hamper the computation of natural frequencies within the accuracy required for practical purposes. The results of numerical simulation performed to determine natural frequencies of outputs and part of the stator bus bar ring in 3D simulation were obtained using a finite element method. Also, this study represents a dependence of design characteristics on both the applied computation algorithms, and proper simulation of boundary conditions. The improvement of numerical simulation authenticity requires the use of experimental data obtained from the bench tests and full-scale experiments in calculation procedures, thus a transition from numerical computations and simulation to calculation and experimental methods to determine the vibration condition of high-power turbogenerators should be provided.

INTRODACTION

The most common damages to stators in high-power turbogenerators are loosening and damages of fastening components in end windings, wearing and mechanical damages of frame insulation, failures of bar connection seals. Actually, all these damages are associated with high vibrations of the stator winding.

The core is the vibration source of the end windings in turbogenerator stators. Magnetic forces radially acting in the air gap between the rotor and the stator set up vibrations of the stator core. The main harmonic induction constituent in air gaps of two-pole turbogenerators gives rise to vibrations, which frequency is two times higher than the rotor frequency. In other words, when the rotor frequency is 3000 rpm, the vibration frequency is 100 Hz [5].

Experimental studies of turbogenerators determined that if a fastening system for the stator winding is designed improperly, its vibration displacement can reach unacceptable high levels under normal operating conditions [2,4,7].

The established practices of calculation and experimental vibration balancing of turbogenerator stators are generally based on the experimental efforts performed on both manufacturers' test benches and power plants during the installation of equipment.

Recently, the intense development of computer engineering and analysis calculation methods has provided better simulation solutions to this problem using advanced capabilities of versatile calculation systems.

So, major advantages the calculation approaches are, as follows:

- possibilities for creating a geometrically accurate object model (3D simulation), without any assumption leading to the occurrence of cumulative errors.
- possibilities for reviewing any operational mode with no harm to the generator, including abnormal, emergency and pre-emergency situations, and modes not recommended for continuous operation,
- no need to stop the generator, reschedule its operation or carry out preparatory works for tests;
- relatively low time expenditures for numerical experiments,
- low (compared to full-scale experiments) cost of works,
- possibilities for multivariant calculations and selecting the most effective design version,
- possibilities for obtaining comprehensive information about all calculation values concerned,
- easy visualization of obtained results; representation of data in graphic or tabular form.

The development of methods intended to calculate the vibration and stress condition of the stator end winding has a hundred year history. Initially, the bar end winding was assumed as a straight multispan beam with swing supports simulating space blocks. Such model provided the analysis of winding deformations depending on the forces acting on it, as well as the assessment of the effect of the distance between the space blocks on natural frequencies of the end windings.

However, further analyzing the nature of the end winding of the stator winding bars showed that their fastening components are quite flexible and, in addition to bending, twisting plays a major role in deformations of the bars.

End winding baskets were considered as cyclically closed systems where each end winding arc was represented by a single weight system with some equivalent weight and rigidity [6].

3D bar calculation models of the end windings on elastic supports were developed based on the actual end winding arc geometry as well as the theory of thin bar vibrations [1]. Such type of calculation model has the following structure: the stator end windings in large turbogenerators represent a system of bars connected between each other and with the external support structure. Each end winding arc consists of two bars connected by their heads and starting at the outlet of the stator slot. The end winding bar is divided into a straight section being a continuation of the slot part, evolvent section or an evolvent on a tapered surface and a head.

Other calculation models [3] represent the stator end winding as a complex system of connected curved bars. All system bars are assumed to be monolithic and have rigidly sealed ends. Every bar connects to two adjacent bars in several points along the length, as well as to the bar of other layer so they forma connected system called a 'basket'. A cyclic symmetry method that uses an assumption about identical bars, connections and structural symmetry of the system is applied to solve the problem.

One of major disadvantages of the applied calculation models is insufficient reliable data on mechanical parameters of the winding and uncertain elastic properties of the fastening structure. These are initial values for calculations that finally determine poor accuracy of the results.

Another disadvantage is a variety of assumptions in the calculation methods. A need to idealize the calculation model relates to both problems with the geometry of the end winding, nature of their connection with each other and support structures, as well as technology features, applied materials and unreliability of much initial data.

Thus, major disadvantages of calculations performed for natural vibration frequencies of the end winding components in turbogenerator stators are:

- Uncertain physical and mechanical properties of materials;
- Uncertain fastening conditions;
- Uncertainties occurring at the stage of manufacture or installation;
- Idealization and assumptions of calculation models.

COMPUTER SIMULATION

Let's consider principal designs of the turbogenerator (figure 1) and the turbogenerator stator (figure 2).

The stator consists of a frame, core, tension ribs, shield, pressure ring, winding basket with banding rings, output bus bar ring, connecting bars, and end winding fasteners.

The output of the stator winding is connected to the output bus bars by means of a flexible jumper and fixed on the stator frame. A rubber gasket is installed between the output end and the frame. Bus bars in the bus bar ring are fixed using bus holders. Fastening is made using a set of fiber-glass plastic pads; bus bars are installed between these pads and steel studs threaded into the holes of the pressure ring.

Let's consider a calculation of natural frequencies and mode shapes of the outputs and a corresponding part of the bus bar ring with bus holders.

All calculation studies were performed using the ANSYS multifunctional finite element calculation system. Subject to the practices applied for measuring dynamic characteristics of the stator end winding components, natural frequencies of bus bars should not fall within the balanced range of 90–120 Hz.

To find a solution to the problem, a 3D finite element model was developed based on the 3D geometric model (figure 3).Table 1 represents basic physical and mechanical properties of applied materials.

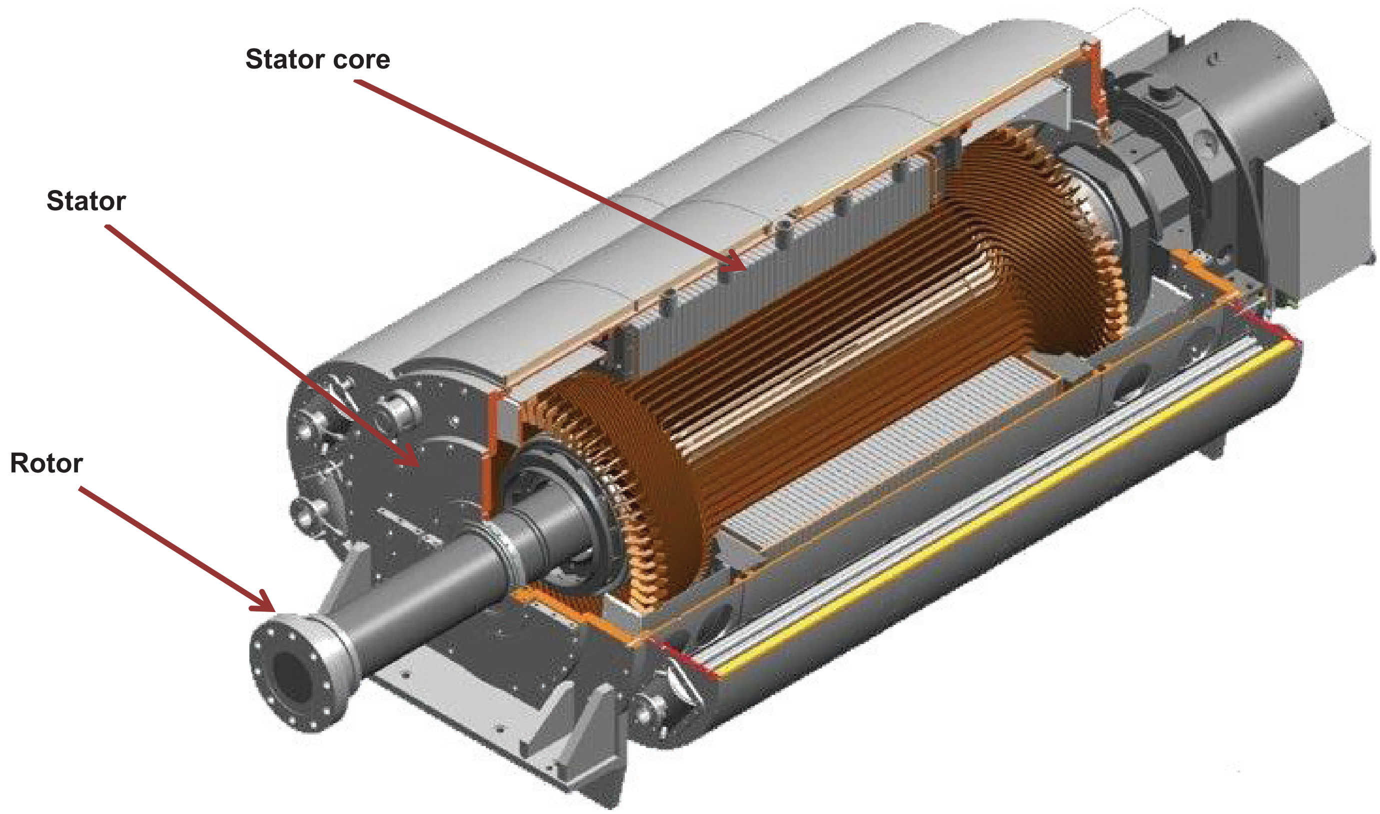

Figure 1. Design of turbogenerator

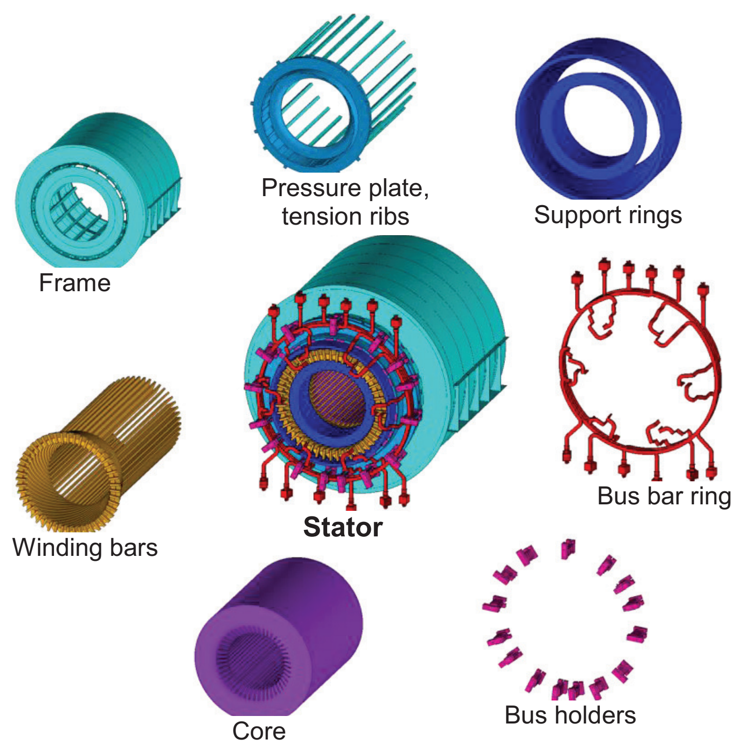

Figure 2. Design of turbogenerator stator

Table 1. Physical and mechanical properties of materials

Material	E-modulus, Pa	Poisson ratio, ν	Density, kg/m^3
Copper	$1.15*10^{11}$	0.3	8900
Rubber	$1*10^{7}$	0.49	2500
Fiber-glass plastics	$2*10^{10}$	0.3	1600

As Table 1 shows, the properties of applied materials essentially differ from each other and that factor should be taken into account when creating a calculation model. The composite structure of Fiber-glass plastics introduces an additional uncertainty and requires a series of experimental works to determine effective physical and mechanical characteristics. Connecting bars of the model have a three-layer structure: rectangular conductor (copper), fiber-glass plastic insulation and cooling water inside, thus providing adequate simulations of rigidity and weight characteristics of bus bars.

The accuracy of calculations performed for natural vibration frequencies of output bus bars is largely determined by the boundary conditions.

The analysis of the results of calculations performed for natural frequency spectra provided the determination of basic components, which rigidity variations have the major effect on the natural vibration frequencies of the bus bar output ends.

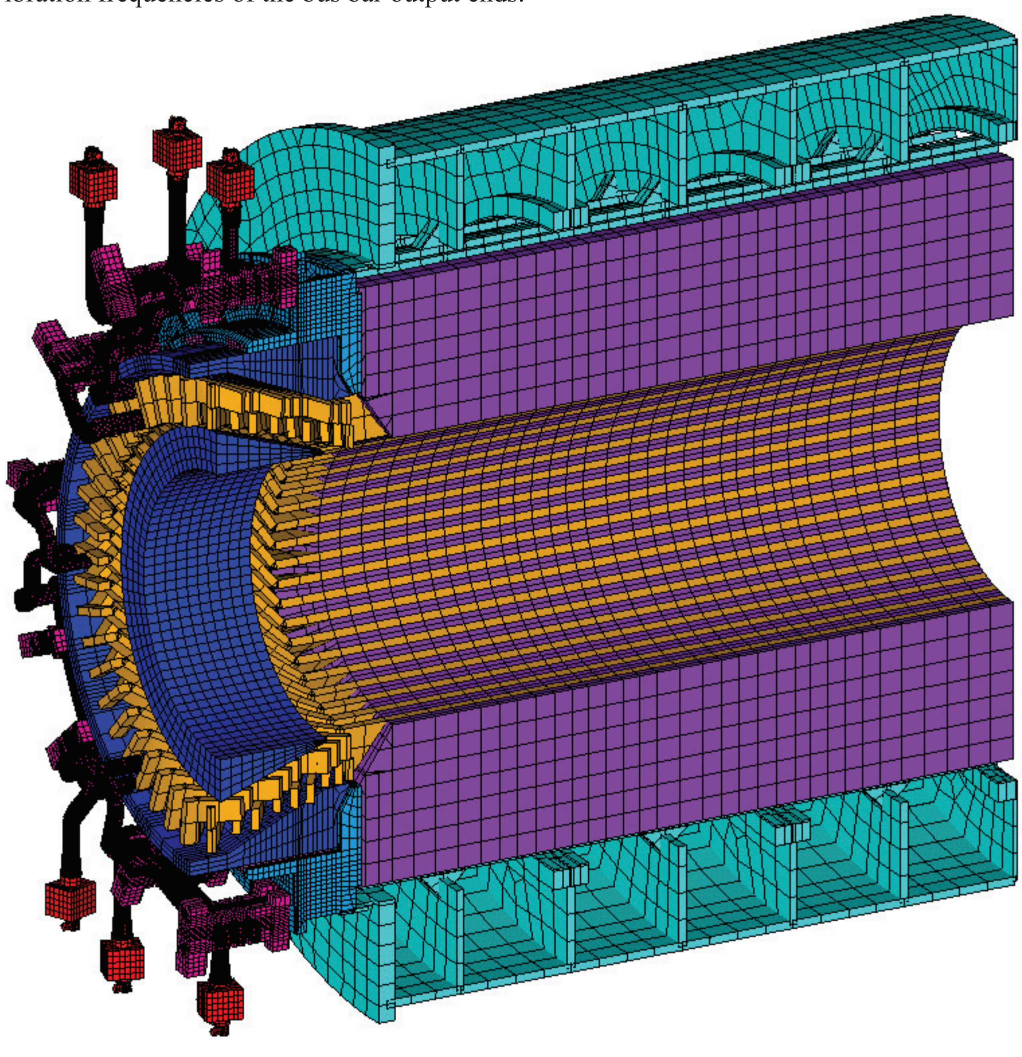

Figure 3. Finite element model

Effect of the connection between the bus bar output and the end terminal

Let's consider four connections types for the bus output end and the end terminal:

- Free end – no connection between the tip and the end terminal is available;
- Flexible connection - the tip and the end terminal are connected using an insulating box through rubber gaskets;
- Flexible jumper – the tip and the end terminal are connected using токоведущей «плетенки», connecting the bus bar with end terminal;
- Rigid connection - the tip and the end terminal are connected using an insulating box through rigid fiber-glass plastic gaskets.

The results of variation calculations performed for natural vibration frequencies of the output bus bars within the corresponding range are given in Table 2.

Table 2. Variations of natural frequencies.

	Variations of natural frequencies, percent					
	1		**2**		**3**	
Calculation method	left	right	left	right	left	right
Free end	1	1	1	1	1	1
Flexible connection	3.5	0.6	5.3	0.01	0.01	2.9
Flexible jumper	1.2	4.9	2.9	0.01	0.01	1
Rigid connection	40.5	21.5			21	39.5

Thus, if we take the natural frequencies of the output bus bars as the reference values when no connection is available between the tip and the end terminal, differences between the natural vibration frequency values will be 40 percent in different the boundary conditions! In other words, variations in the natural frequencies from the connection conditions for the bus output ends and the end terminals, at a vibration frequency of nearly 100 Hz, can reach 40 Hz that exceeds the balanced range value, i. e. 30 Hz.

A substantial effect of the connection conditions for the bus output ends and the end terminals on the calculation results of natural frequencies and, consequently, vibration balancing of turbogenerator stators shows that the use of only calculation methods cannot give a solution to the problem within the required accuracy. Calculation models should be adjusted based on both the experimental data obtained from the bench tests carried out by manufacturers and using a series of special experimental studies conducted for rigidity and weight characteristics of major components affecting the design weight characteristics. Here, the defining factors are flexibility and the connection nature of the end winding fasteners between each other.

Effect of the holder rigidity

Bus bars in bus bar ring are fixed using bus holders (figure 4). Bus bars in the bus holder are pressed to each other through fiber-glass plastic pads and screwed to the pressure ring by studs. As the first approximation, let’s consider the entire bus holder as a single whole, isotropic material.

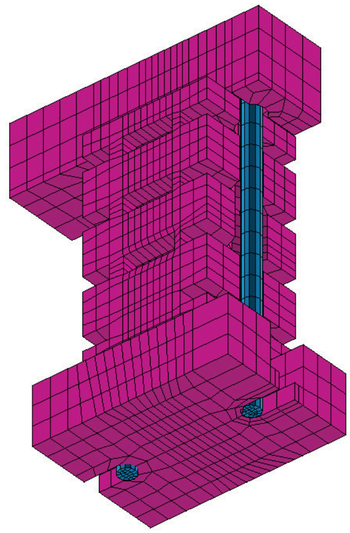

Figure 4. Fastening of bus bars to the pressure ring

To assess the effect of rigidity and weight characteristics of bus holders on natural frequency values, calculations were performed for both types:

- bus holder material: fiber-glass plastic, $E = 2*10^{10}$ Pa (refer to Table 2),
- bus holder material: steel, $E = 2*10^{11}$ Pa.

These two types enable the consideration of the entire range of variations in the natural frequencies relating to rigidity of the bus holder itself. The fiber-glass plastic bus holder is initially more flexible; rigidity of a steel bus holder is higher than that of the composite structure. Table 3 shows relative variations in output bus frequencies when materials of the bus holder are changed.

Table 3. Variations of natural frequencies.

	Variations of natural frequencies, percent					
	1		**2**		**3**	
Calculation method	left	right	left	right	left	right
Free end	2.1 4.7	2	2.6	2.3	11.5 1.9	1.7 3.8
Rigid connection	1.3	0.1	-	-	6.1	1.3

Thus, when the bus holder rigidity changes 10 times, the natural vibration frequency value increases by up to 11.5 percent depending on the mode shape. These figures were obtained after the simulation of a bus holder was simulated as a single whole, without any regard to flexibility of the bus bars fixed in it.
To adjust the simulation model, a series of experimental works should be performed to determine actual axial, radial and tangential rigidity of bus holders.

Effect of the pressure plate

To evaluate the effect of the pressure plate on frequency characteristics of the system, a 3D finite element model (figure 5) was developed. This model comprises:

- Output buses,
- Bus holder,
- Pressure plate.

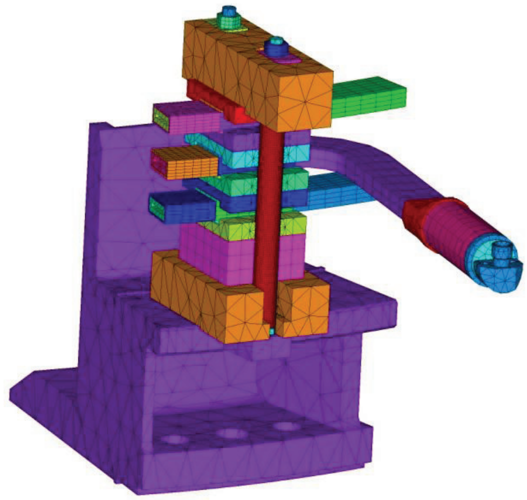

Figure 5. Finite element model for the bus holder with pressure plate

Conditions of rigid fastening on the contacting surface of the plate and the stator components were accepted as the boundary conditions.
The results of variation calculations performed for natural vibration frequencies of the output bus bars are given in Table 4.

Table 4. Variations of natural frequencies

Design type	Variations of natural frequencies, percent					
	1		**1**		**3**	
	left	right	left	right	left	right
With pressure plate	**-3.7**	**-2.1**	**-1.4**	**-0.3**	**-5.1**	**-9.2**

The obtained results show that the consideration of the pressure plate flexibility reduces the natural vibration frequencies of the output bus bars to 9 percent.
It should be noted the pressure plate is not an equally rigid structure in the areas where bus holders are mounted; in this regard, fastening rigidity for every bus holder is different.
The calculation results of natural frequencies of the bus output ends confirm the need to consider the pressure plate during calculations. The pressure plate structure with bus holders tightened by studs is adequately determined by the calculation studies.

CONCLUSION

- Effect of boundary conditions on simulation results is extremely important so as calculation results for natural frequencies are strongly depend on boundary conditions.
- Experimental data should be taken into consideration to correct simulation models what enables to improve prediction validity of simulation for actual vibrational state of turbogenerator.

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