Numerical simulation of the rotor winding temperature field with self-ventilation from the sub-slot duct

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Abstract. This paper describes the evaluation of rotor winding self-ventilation system with the radial ducts fed from sub-slot ducts for air and hydrogen cooled turbogenerators. The verification of the computational fluid dynamic (CFD) calculation has been carried out with the experiment. The conjugate heat transfer (CHT) is performed with the finite-volume method for rated excitation current. The analysis of the temperature field of the rotor winding with different configuration of the sub-slot duct was carried out.

1. Introduction

In modern heavy equipment industry, the main principal trend is the increasing the unit capacity of turbogenerators.

An important factor limiting the power of the turbogenerator is the rotor winding heating. Most of the world’s leading manufacturers accept gas cooling: air for generators up to 400-500 MW; hydrogen up to 1300-1500 MW (with a two-pole rotor).

The problem of efficient gas cooling of rotors remains one of the top priorities in the local and global turbogenerator industry as pointed out in (1) and (2).

The principle of rotor winding self-ventilation system with the radial ducts fed from sub-slot ducts has become widespread in world practice in the construction of air and hydrogen cooled turbogenerators (3). The goal of this type of gas cooling systems is to minimize the overall temperature level and its local (peak) values within the groove part of the rotor winding. This requires a detailed study of a number of factors that determine the nature of the temperature field in the rotor winding.

The decisive influence on the performance of this ventilation system is exerted by the size of the inlet section of the sub-slot ducts which imposes the main restriction on the active rotor length. If the issue of the total gas flow rate, which determines the level of the average winding temperature, is successfully solved, the problem arises of minimizing the temperature differences within the axial length of the slot part. In solving this problem, the designers focused on the root cause of the longitudinal non-uniformity of the winding temperature field, i.e. on the mechanism of gas velocity distribution in the radial ducts along the rotor length as pointed out in (4) and (5).

In up-to-date design, this problem is mitigated by the use of a variable-section sub-slot ducts, but the effectiveness of this design technique has not yet been adequately covered in the literature.
2. Verification of CFD calculation with experiment

To compare the numerical calculation with the experiment, the geometry was chosen in the study of the full-scale model of the turbogenerator rotor with self-ventilation from the sub-slot duct.

The numerical model (air) of solving CFD problem is presented in Figure 1.

![Figure 1. The numerical model (air) of solving CFD problem.](image)

Computational domain parameters: ½ rotor length 1960 mm; number of radial ducts 24; the ratio of the cross section of the sub-slot duct to the total area of the radial ducts 0.43; speeds of rotation 485, 750, 1000, 1200, 1500 rpm.

A reliable tool for studying the complex processes of CFD and CHT problems in electric machines are well-adapted practical application packages (ANSYS, Star-CCM).

A brief review of the application numerical calculation methods is described in article (6).

The stationary, isothermal, turbulent air flow with a constant density is considered. The physical properties of air at a temperature of 40 °C are taken into account.

The Reynolds-Averaged Navier-Stokes equations (RANS) are solved applying a finite volume approach in the ANSYS Workbench software package (7). In a RANS simulation the small statistical motion of the fluid (i.e. turbulence) has to be modelled within a turbulence model. In case of overall CFD calculations the well-known k-epsilon model has been selected (number of cells are 0.6M; 600 iterations). The multiple reference frame (MFR) was selected for rotating parts (8). The inlet to the sub-slot duct and the outlets from the radial ducts are connected by a connecting region.

Comparison of the numerical velocity distribution in the rotor radial ducts with the experiment is presented in the Figure 2. In general, there is a good correlation of the calculated data with experiment with minor differences in the nature of the flow, starting with radial duct No. 19.

The inequality is probably due to the presence of leakage at the end of the experimental model. To test this assumption, a model with a clearance from the end was made and the result is presented in the Figure 3. If there is a gap, there are obvious extremes in the distribution and the identity of the calculation with the experiment is observed.
3. CHT Result

The temperature field study was made of the rotor groove part for air and hydrogen-cooled turbogenerators using various configuration of the sub-slot ducts by solving the CHT problem in the ANSYS Workbench software package, e.g., (9) and (10). The calculation is performed for the rotor rated current.

The radial and sub-slot ducts geometric characteristics (duct pitch and shape, cross-sectional dimensions, radial duct length in the winding) correspond to those used in engineering practice.

For a hydrogen-cooled turbogenerator with a modeling 1200MW rated power, to assume the following dimensions: ½ rotor length 3840 mm; number of radial ducts 60; the ratio of the cross section of the sub-slot duct to the total area of the radial ducts 0.38; two-row radial duct arrangement; speed of rotation 1500 rpm.

For a air-cooled turbogenerator in operation with a 320 MW rated power: ½ rotor length 2450 mm; number of radial ducts 78; two-row radial duct arrangement; the ratio of the cross section of the sub-slot duct to the total area of the radial ducts 0.26; speed of rotation 3000 rpm.

To the existing CFD model with self-ventilation from the sub-slot duct (the model did not take into account the axial-radial distribution) were added:
- a rotor winding in the form of the solid bus. The copper section begins at the beginning of the radial duct;
- a rotor winding insulation;
- half of the rotor teeth on both sides of the sub-slot duct.

The CHT model is shown in the Figure 4.

Figure 4. The numerical model (air) of solving CHT problem.

The main gas thermophysical properties and the model parameters are presented in the Table 1.

Table 1. The main gas properties and model parameters.

| Turbogenerator rated power, MW | Gas environment | Rotor winding | Insulation |
|-------------------------------|-----------------|---------------|------------|
|                               | Density, kg/m³ | Dynamic viscosity, Pa·s·10⁻⁵ | Thermal conductivity, W/m·K | Current density, A/mm² | Volumetric heat flux, MW/m³ | Thermal conductivity, W/m·K |
| 320                           | 1.127           | 1.93          | 0.027      | 4.9       | 0.69                          |                           |
| 1200                          | 0.637           | 0.94          | 0.187      | 7.5       | 1.45                          | 0.26                      |

The stationary problem is considered. The CHT mesh has a much higher resolution in order to resolve the heat transport processes near the wall in comparison with CFD model. The solids can be represented by a much coarser mesh. In total region already has a mesh with about 3M cells. The number of iterations for solving process were 800.

Besides the flow rate through the ventilation duct system (taken from CFD) also the inlet temperature into the sub-slot duct has to be specified 40°C. The pressure is equalized with the surrounding gas at the radial ducts outlet.

The volumetric heat flux of the rotor winding is specified as a heat source. The rotor winding has anisotropic in thermal conductivity, i.e. limits heat transfer in the radial direction (bars consisting of single strands). The tooth thermal conductivity is 50 W/m·K. Symmetry conditions on the teeth boundaries are specified.

The initial temperature for gas and all solids (rotor winding, insulation, tooth) are 40 °C.

4. Results analysis

The main results of the calculation are presented in the Table 2.
Table 2. Comparison of basic ratios for air and hydrogen-cooled turbogenerators.

| Turbogenerator rated power, MW | Sub-slot duct | Maximum/Minimum radial ducts velocity ratio | Maximum/Average temperature rotor winding ratio | Maximum gas heat up in radial ducts |
|-------------------------------|--------------|---------------------------------------------|-----------------------------------------------|-------------------------------------|
| 320                           | constant     | 5.6                                         | 1.4                                          | 31.1                                |
|                               | variable     | 4.5                                         | 1.3                                          | 28.0                                |
| 1200                          | constant     | 3.0                                         | 1.4                                          | 27.8                                |
|                               | variable     | 2.1                                         | 1.3                                          | 25.1                                |

The most effective way to reach the uniform distribution of velocity in the radial ducts is by changing the inlet cross section and shape sub-slot duct. It becomes possible to reduce the local maximum temperature of the rotor winding when using the variable sub-slot duct. This is due to the equalizing of the velocity field in the radial ducts along the axial rotor length.

Zones with the intensive backward flow are observed in the initial part of radial ducts along the axial rotor length with a constant sub-slot duct. Backward flow is noticeably decreased with the presence of steps in the sub-slot duct (Figure 5).

![Figure 5](image.png)

**Figure 5.** The temperature distributions and velocity vector in the initial part of radial ducts along the axial rotor length with constant (a) and variable (b) sub-slot ducts of turbogenerator with a modeling 1200 MW rated power.

The non-uniform temperature distribution along the rotor winding length is lower when using a variable sub-slot duct.

According to the results of numerical simulation the velocity in the radial ducts and the average rotor winding temperature distributions along the rotor length are shown in the Figures 6–9.

The gas velocity in the radial duct is determined in the middle of the height of the radial duct and calculated through the mass flow rate.

In order to provide a brief validation of the shown method, some comparison with thermal tests of a 320 MW turbogenerator at Kashira Power Plant (Russia) is done. The rotor of turbogenerator has a variable sub-slot duct. The average experimental rotor winding temperature rise above the cold (inlet) air temperature on the rated rotor current is 57 °C.

This calculation data correlates well with the experiment.
Figure 6. The distribution of the rotor winding temperature rise above the cold (inlet) gas temperature along the axial rotor length with constant (a) and variable (b) sub-slot ducts of turbogenerator with a modeling 1200 MW rated power.

Figure 7. Gas heat up and velocity in the radial duct along the axial rotor length with constant (a) and variable (b) sub-slot ducts of turbogenerator with a modeling 1200 MW rated power.
5. Conclusions

1. The correctness of the numerical simulation of the turbogenerator rotor winding self-ventilation system with the radial ducts fed from sub-slot ducts is confirmed by a good agreement with the results of physical modeling of the system on a rotating the full-scale model both in terms of flow capacity (overall air flow rate)) and in relation to the velocity distribution along the rotor length.

2. The study of the thermal condition of the rotor groove part for air-cooled turbogenerator by means of numerical simulation is in good agreement with the scale of the average experimental rotor winding temperature with the results of thermal tests of a 320 MW turbogenerator under conditions of its operation at a power plant.

3. According to the results of numerical simulation, the temperature distribution in the groove part of the rotor winding of a gas-cooled turbogenerator (both air and hydrogen) has significantly less irregularity with a variable section of the sub-slot duct compared to the version of the sub-slot duct with a constant section, while the average temperature scale the winding remains almost unchanged.
6. References

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