Validation of Numerical Approaches for Simulating the Heat Transfer in Stator Ducts with Measurements

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***Abstract*— In this paper, two different methods are presented for simulating the heat transfer along the stator ducts of a hydro generator. The investigations are focused on the fluid flow in an air gap between the insulation of the winding bars and the stator iron. Traditionally, the air gap is modeled with the finite volume method, so the heat conduction and the heat convection are taken into account. As a novelty, it is shown that it is also possible to define a thermal resistance in the interface simulating only the heat conduction without modeling the air gap resulting in faster simulations. Measurements are available for a validation of the two different numerical approaches.**

***Index Terms*— Fluid Flow, Measurement, Stators, Thermal Analysis, Thermal resistance**

I. Introduction

The arising losses in a hydro generator have to be discharged to ensure the operating characteristics. In such electrical machines the heat has to be removed with a cooling circuit. For the design of the cooling scheme air flow and thermal networks are mostly used, as presented in [1]. For an improvement of those 0D methods, 3D numerical methods are often used [2], especially for calculating the heat transfer coefficient. The state of the art 3D methods are the finite element method (FEM) [3] and the finite volume method (FVM) as investigated [4] and [5]. The advantage of the finite volume method, used in conventional computational fluid dynamics (CFD) softwares, over FEM is the consideration of the actual wall heat transfer coefficient, but the losses in the solid domains cannot be considered, while FEM is capable of this. The conjugate heat transfer (CHT) method allows defining the sources in the solid domains and so the heat conduction and convection can be investigated [6].

This work presents the CHT method on simplified stator duct models, which differ from each other in the slot section components. The aim is a comparison of the solution of the defined thermal resistances at the contact interface with the numerical solution, which includes the meshed air gap in the model.

II. Numerical Model

A simplified model of a stator section has been under experimental investigation at the company ANDRITZ Hydro. These measurement data are used for a validation of the numerical model. Therefore, it is important to define the same conditions at the boundary as in the measurement and implement these in the CFD program.

A slot section model has been created with the winding assembly and the stator iron. The main difference between the three models is in the contact between insulation (f) of the winding bars and the stator iron (b) and (c).

Fig. 1 shows the numerical model with the fluid and solid domains in detail.

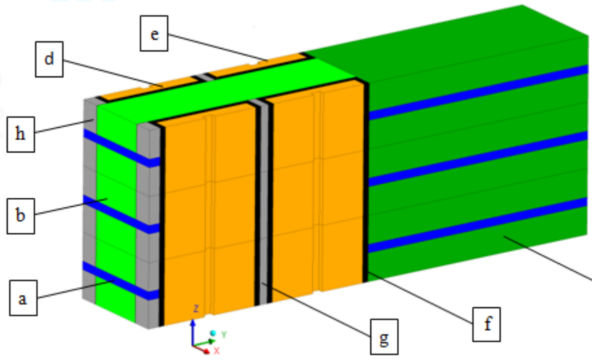

Fig. 1 Numerical stator model contains fluid and solid domains, which are (a) fluid in the stator duct, (b) iron teeth, (c) iron yoke, (d) top copper bar, (e) bottom copper bar, (f) insulation, (g) spacer between bars and (h) slot wedge

The three numerical models under investigation are:

A. *Model A*

This model has an air gap with a constant length between the insulation of the winding bars and the stator iron.

B. *Model B*

A ripple spring is positioned on one side between the insulation of the winding bars and the stator iron.

C. *Model C*

Model C has nearly the same assembly like model A with the air gap. The main difference is that the air gap is replaced with epoxy resin at some locations.

III. Simulation Method

Conventional CFD simulations are using the FVM to solve the transport equations:

$$\frac{\partial \rho}{\partial t}+\frac{\partial \rho u_j}{\partial x_j}=0 \tag{1}$$

$$\frac{\partial \rho u_i}{\partial t}+\frac{\partial \rho u_i u_j}{\partial x_j}=-\frac{\partial p}{\partial x_i}+\frac{\partial \rho \tau_{ij}}{\partial x_j}+\rho f_i \tag{2}$$

$$\frac{\partial \rho e_t}{\partial t}+\frac{\partial \rho u_i e_t}{\partial x_j}=-\frac{\partial u_i p}{\partial x_j}+\frac{\partial u_i \tau_{ij}}{\partial x_j}+\rho u_i f_i-\frac{\partial q_j}{\partial x_j}+Q \tag{3}$$

The boundary conditions on the walls are mostly defined with a constant temperature gained from measurements in these CFD set-ups. For the solution of the heat transfer only heat convection is taken into account. The conjugate heat transfer (CHT) method already considers the heat conduction in the energy equation [7]:

$$\frac{\partial \rho h_t}{\partial t} + \frac{\partial \rho u_s h_t}{\partial x_j} = \frac{\partial}{\partial x_j}\left(\lambda_j \frac{\partial T}{\partial x_j}\right) + Q \qquad (4)$$

Therefore, the solid domains are also modeled in this work. In the following research we have a look on the air gap between the insulation of the winding bars and the stator iron. This air gap is less than one millimeter wide and allows the air to stream in z-direction, see Fig. 1. The Reynolds number in this area is in the laminar zone and this requirement can cause problems to the numerical solution. A fine mesh must also be generated in the air-gap, which increases the numerical effort.

Another option is to omit the air gap and define a thermal resistance R (in Kelvin per Watt) on the interface between the insulation and stator iron:

$$R = \frac{d}{\lambda A} \quad , \qquad (5)$$

where d (in meters) is the path length of the air gap, A (in square meters) is the path area and λ (in Watt per meter and Kelvin) is the thermal conductivity of the material. This interface replaces the air gap and the consequence is that only the heat conduction has to be solved at the interface.

For the following cases the Shear Stress Transport (SST) turbulence model has been used with an automatic wall treatment.

IV. Preliminary Results and Conclusion

Fig. 2 shows the comparison of the results gained with the finite volume method (FVM) and the interface method (IFM). In this diagram the temperature difference has been calculated with an average value of the copper temperature in the top and bottom bars and has been normalized. The inlet temperature has been defined from the measurements, so the deviation is due to the copper temperature.

For model A the standard deviation is for the FVM model compared with the measurement about 2.08% and for the IFM model 2.03%. The same values for model C are 1.62% for FVM model and 1.67% for the IFM model. The results show that the error deviation is the same for both numerical methods, but the numerical effort is clearly higher by modeling the air gap like in the FVM model. In fact, the solution time is less than the half by defining a thermal resistance instead of modeling the gap.

The heat convection can be neglected for creeping flows and therefore the heat conduction is the main effect for the heat transfer at low Reynolds numbers.

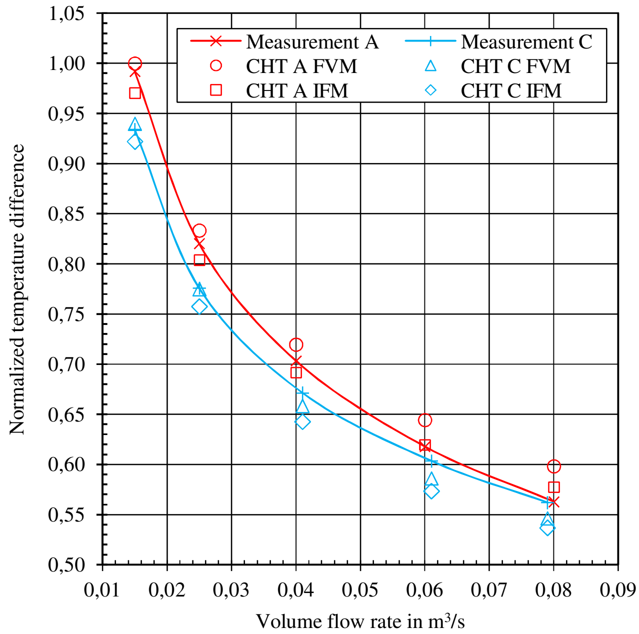

Fig. 2 Normalized temperature difference between copper and fluid inlet temperature for model A and model C

The paper describes two different approaches for simulating the heat flux in an air gap for small Reynolds numbers with the same standard deviation compared with the measurements. The numerical approach with defining the thermal resistance has a lower numerical effort and consequently a faster computing time.

V. Acknowledgment

This work has been supported by the *Christian Doppler Research Association (CDG)* and by the *ANDRITZ Hydro GmbH*.

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