A STUDY OF CONVECTIVE HEAT TRANSFER IN A ROTOR-STATOR SYSTEM OF DISK-TYPE ELECTRICAL MACHINES

Rasekh A.*[,1], Sergeant P.[2,3] and Vierendeels J.[1]
*Author for correspondence
[1]Department of Flow, Heat and Combustion Mechanics, Faculty of Engineering and Architecture, Ghent University, Ghent, Belgium.
[2]Department industrial Technology & Construction (EA20), Faculty of Engineering and Architecture, Ghent University, Ghent, Belgium.
3 Department of Electrical Energy, Systems and Automation (EA08), Faculty of Engineering and Architecture, Ghent University, Ghent, Belgium

E-mail: Alireza.Rasekh@UGent.be, peter.sergeant@ugent.be, Jan.Vierendeels@UGent.be

ABSTRACT

Accurate heat transfer analysis of the rotor-stator system in disk-type electrical machines is crucial for design purpose, especially to prevent overheating. If the internal temperature in the machine exceeds the critical value (T_c=$150°C$), it results in demagnetization in the rotor. A high temperature increases the resistivity of the copper windings, which negatively affects the efficiency of the machine. The study undertaken here aims to provide analytical modeling of the discoidal system which is applicable to various geometries and boundary conditions. The rotor-stator configuration is enclosed in a cylindrical chamber, and all the surfaces within the machine are considered isothermal, each on their own temperature. The average convective heat transfer coefficients are defined, taking into account the bulk flow temperature as the reference temperature instead of the ambient temperature. The latter is typically used in literature but has almost no influence on the heat transfer. Thereafter, in order to estimate the bulk fluid temperature, a linear correlation between the surface temperature of the rotor, the stator and the cover is made, based on CFD simulations. These are performed for different rotational Reynolds numbers and gap size ratios at different combinations of surface temperature of the rotor, the stator and the cover. Results have been compared with the available data in literature and good agreement has been found. It is showed that there is a gap size ratio for which the average convective heat transfer for the stator surface reaches a minimum. Additionally, it is found that the proposed correlations for the convective heat transfer in the rotor and the stator surface in the gap are totally applicable to the disk type electrical machine with different working conditions.

INTRODUCTION

Convective heat transfer in the discoidal configuration of the disk-type electrical machines is of paramount importance for the design purpose where the air gap limits the heat transfer [1]. Lack of precise knowledge in the prediction of temperature distributions in the machine can lead to fatal consequences. On the one hand, overheating may occur due to lack of sufficient cooling by designers, which in turn causes demagnetization as the temperature exceeds the critical value point (e.g. T_c=$150°C$ for SH type NdFeB magnets [2]). On the other hand, the designer should ensure the excessive cooling provision through the external ventilator, resulting in an extra energy consumption which in turn deteriorates the energy efficiency. As a result, a detailed understanding of convective heat transfer in the rotor–stator system of disk-type electrical machines seems to be vital.

Heat transfer and air flow analysis in the rotor-stator system have been approached by researchers [3-4]. Daily and Nece [5], to the knowledge of the author, pioneered the problem of fluid flow between rotor and the stator. Yuan et al. [6] studied the turbulent heat transfer on the stator and the flow characteristics in the gap between the disks. They showed that there is a rotor-stator distance where the heat transfer on the stator side reaches maximum. Howey et al. [7] investigated experimentally the stator heat transfer in a discoidal configuration and demonstrated that the local Nusselt number increases at the periphery due to ingress of air toward the stator.

CFD simulations have been widely applied to evaluate the thermal performance of disk type electrical machines [8-10]. Chong et al. [11] simulated the NGenTec Axial Flux Permanent Magnet (AFPM) prototype generator operating at 100rpm and built the experimental set up to validate their results. Moradnia et al. [12] studied the cooling of an electrical generator using the frozen-rotor concept. They showed that this numerical scheme is highly advantageous because it predicts the same velocity distribution as in the experiment.

The aforementioned studies indicate that the CFD approaches can be a powerful tool in the design of disk-type electrical machines. Nevertheless, they require significant amount of time in order to yield the results, inasmuch as the electrical machines consist of complicated geometries. As a consequence, having an empirical formula that estimates the convective heat transfer in the disk type electrical machines is highly useful. In this paper, an analytical formulation will be presented to calculate the average heat transfer in the air gap of the disk type electrical machine. The correlations, which are constructed based on the CFD simulation, are able to predict the mean Nusselt numbers for the stator and the rotor's surface in the gap for various gap size ratios, $G=s/R$, and the rotational Reynolds numbers, $Re=\omega R^2/\upsilon$, where s is the gap distance between disks, R is the disks' radius, ω is the rotor's angular velocity and υ is the air kinematic viscosity. The bulk fluid temperature is taken into account as the reference temperature instead of the ambient temperature which is typically used in the literature. Subsequently, a linear correlation between the surface temperature of the rotor, the stator and the cover is innovatively made in order to predict the bulk fluid temperature. Details of the proposed method and the results are discussed in the following sections.

PROBLEM SETUP

In disk type electrical machines, adverse temperature rises in the stator windings and the permanent magnet happen due to total losses (copper loss, iron loss, losses in magnets, windage loss…) which show themselves in the production of heat. Since most heat transfer phenomena occur in the air gap between the rotor and the stator, knowledge of the air gap convection is significant in proper design of the cooling purpose. Although the real geometry of a disk type electrical machine is very complex, having an understanding of a simplified discoidal configuration is highly beneficiary.

The schematic diagram of the problem under investigation has been depicted in Figure 1. The configuration consists of two rotors and one stator where only half of the geometry is taken into consideration. Thus, a symmetry plane has been defined in the middle of the geometry to halve the computational cost. The left disk and right disk correspond to the rotor and the stator, respectively.

It is worth mentioning that the surface temperatures of the rotor, the stator and the cover are considered to be isothermal, and the model is capable of predicting the convective heat transfer with different surface temperatures. Moreover, air is considered as an incompressible ideal gas, so its density varies with temperature and computes as below,

$$\rho = \frac{P}{RT} \tag{1}$$

Where P is the ambient pressure, T is the local fluid temperature, and R is the specific gas constant of air. The variation of thermal conductivity, viscosity, heat capacity with temperature found to be insignificant, and Prandtl number ($Pr=\mu c_p/k$) remains 0.7.

Considering $Re=1.26\times10^5$ and $G=0.01333$ as the reference point, the CFD simulations have been implemented with the rotational Reynolds number ranging from 4.19×10^4 to 4.19×10^5 and the gap size ratio ranging from 0.00333 to 0.08. We intend to consider the buoyancy effect to capture the natural convection due to air density gradient and the gravitational force. Therefore the computational domain should be 3D. The commercial CFD software Ansys FLUENT has been used to simulate the flow fields [13]. The turbulence was treated with a SST k-ω model and the boundary layers around the solid walls were designed to obtain a y^+ value below 1. SIMPLE algorithm was utilized to solve the pressure-velocity coupled problem. The second order upwind scheme was employed for discretization of the physical parameters namely energy, momentum, turbulent kinetic energy k and specific dissipation rate ω whereas standard scheme was used for the pressure corrective equation.

A mesh independency analysis has been implemented by comparing the results of the simulation for the grids with 753000, 2604000 and 6097000 nodes and it was found that the results are completely grid-independent.

PROPOSED FORMULATION

A fully predictive correlation to evaluate the convective heat transfer in an AFPM is of paramount importance. This formulation aims to estimate mean Nusselt numbers of the rotor and the stator surface in the air gap for different values of Reynolds number, the gap size ratio, and the surface temperature. Although it is possible to find the accurate results by doing the CFD simulations for each case, it would be definitely computationally demanding.

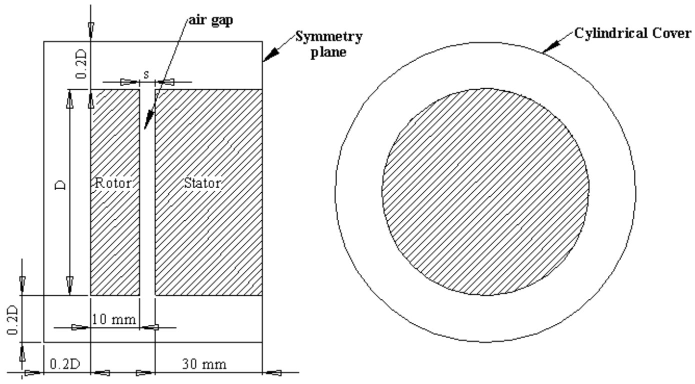

Figure 1 Problem under investigation

The mean convective heat transfer coefficient for the isothermal surface of "i" is defined by:

$$\bar{h}_i = \frac{\bar{q}_i}{\left(T_{surf,i} - \bar{T}_{ref,i}\right)} \tag{2}$$

Where $\bar{q}_i$ is the average heat flux of the surface i, $T_{surf,i}$ is the surface temperature and $\bar{T}_{ref,i}$ is the reference temperature. It should be mentioned that $\bar{T}_{ref,i}$ must strictly refer to the bulk fluid temperature adjacent to the surface where the measurement for $\bar{q}_i$ is being made, instead of the ambient temperature which is typically used in literature. Moreover, the average Nusselt number for each surface can be given through the following formulation,

$$\overline{Nu_i} = \frac{\bar{h}_i R}{k} \tag{3}$$

Where k and R are the thermal conductivity of air and the radius of the disks. Here, the following formulation is used to assess the average bulk fluid temperature in the air gap:

$$\bar{T}_{ref,i} = \frac{1}{V_i}\int_{V_i} T dV \tag{4}$$

where V_i is the volume in the air gap (the highlighted volume in Figure 2). In this way, the average convective heat transfer would become independent of the surface temperature as well as the ambient temperature.

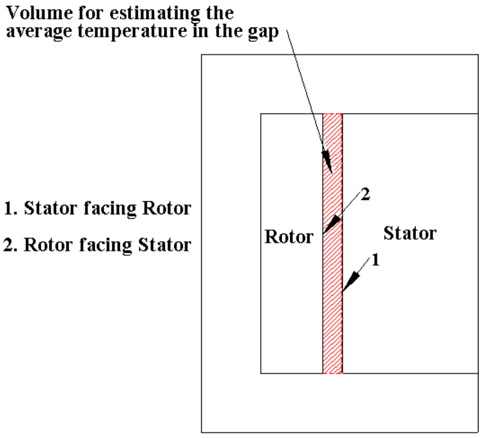

Figure 2 The highlighted volume is utilized to estimate the bulk fluid temperature

The heat transfer from the different surfaces can be dependent on the rotor temperature, the stator temperature and/or the cover temperature. Since there are three temperatures involved, we want to check which ones should be used in the calculation of the heat transfer coefficient. It turns out that it is best to use a combination of the three (instead of using e.g. ambient temperature when dealing with the heat transfer in the gap as has already been done in literature). Here, we presume that the bulk fluid temperature can be expressed as,

$$\bar{T}_{ref,i} = a_i T_R + b_i T_S + (1-(a_i+b_i))T_C \quad (5)$$

where the subscripts *R, S* and *C* correspond to the rotor, the stator, and the cover in the configuration, respectively. The coefficients a_i and b_i are dependent on *Re* and *G* which will be discussed later in this paper.

In order to find the appropriate correlations, CFD simulations are firstly performed at the reference point ($Re=1.26\times10^5$ and $G=0.01333$) for six different combinations of the surface temperature of the rotor, the stator and the cover. The details of the calculation are shown in following Table 1.

Table 1 Different surface temperature combinations in order to find the appropriate values of a, b and $\overline{Nu}$ for the surface "Stator facing rotor"

Combination	T_s(°C)	T_r(°C)	T_c(°C)	T_f(°C) (CFD)	a	b	$\overline{Nu}$
1	120	100	50	110	0.4829	0.5122	243.12
2	100	80	50	90.1			
3	120	80	50	100.3			
4	110	90	60	110.1			
5	130	110	60	120			
6	110	70	45	90.4			

In doing so, the average bulk fluid temperature and the mean convection heat transfer coefficients for different surface temperature in the gap can be given. Since the number of equations to find a_i and b_i is more than two, the least square method is employed to equation (5). Next, to calculate the proper values of the mean Nusselt number at the reference point, we average the values given by equation (3).

Furthermore, it is interesting to know the variation of a_i, b_i and $\overline{Nu_i}$ with *G*. The *CFD* simulations is, therefore, done for G={0.00333, 0.00666, 0.02, 0.02666, 0.04, 0.08} at $Re=1.26\times10^5$ for different surface temperature combinations of the rotor, the stator and the cover. Having the average bulk fluid temperature and the mean convection heat transfer coefficient in each surface for various temperature combinations, again the least square method is employed to equation (5) to estimate the values of a_i and b_i, and again the mean Nusselt number for each surface in the gap is found by averaging the results from equation (3). In a similar manner, the appropriate values for the a_i, b_i and $\overline{Nu_i}$ for a different range of Reynolds numbers Re={4.19×10^4, 8.38×10^4, 2.10×10^5, 3.35×10^5, 4.19×10^5} at fixed $G=0.01333$ can be calculated.

The objective of this analytical approach is to find the correlations that express the variation of a_i, b_i and $\overline{Nu_i}$ as function of *Re* and *G* :

$$\begin{aligned} a_i &= F_i(G, \mathrm{Re}) \\ b_i &= G_i(G, \mathrm{Re}) \\ \overline{Nu}_i &= Y_i(G, \mathrm{Re}) \end{aligned} \quad (6)$$

We assume that the above functions of two variables can be written as the product of two functions of one variable as,

$$\begin{aligned} a_i &= a_i^* f_{i,1}(G) \times f_{i,2}(\mathrm{Re}) \\ b_i &= b_i^* g_{i,1}(G) \times g_{i,2}(\mathrm{Re}) \\ \overline{Nu}_i &= Nu_i^* y_{i,1}(G) \times y_{i,2}(\mathrm{Re}) \end{aligned} \quad (7)$$

where the superscript $*$ refers to the values of a_i, b_i and $\overline{Nu_i}$ at the reference point. It should be noted that the output of these functions at the reference point is one. In order to find the formulations, the curve fittings have been done for both surfaces: "Rotor facing stator" and "Stator facing rotor". The details of the formulations can be found in Table 2.

Table 2 Proposed Correlations for the surfaces in the gap

Parameter	Surface: Rotor facing Stator (i=1)	Surface: Stator facing rotor (i=2)
a_i^*	*0.4829*	*0.4829*
b_i^*	*0.5122*	*0.5122*
Nu_i^*	*240.72*	*243.12*
$f_{i,1}$	$-7.447G+1.071$	$-7.447G+1.071$
$f_{i,2}$	$-8.524\times10^{-8}Re+1.018$	$-8.524\times10^{-8}Re+1.018$
$g_{i,1}$	$2.353G+0.9839$	$2.353G+0.9839$
$g_{i,2}$	$3.995\times10^{-8}Re+0.9912$	$3.995\times10^{-8}Re+0.9912$
$y_{i,1}$	$0.001366G^{-1.26}+0.675$	$0.0006897G^{-1.375}+0.7201$
$y_{i,2}$	$3.983\times10^{-6}Re+0.4822$	$3.765\times10^{-6}Re+0.5029$

RESULTS AND DISCUSSIONS

To give a clear insight about the convective heat transfer in the rotor-stator configuration, the temperature contour in the midplane of the geometry has been illustrated in Figure 3 when the surface temperature of the stator, the rotor and the cover are kept at 120 ºC, 100 ºC and 50 ºC, respectively. As seen, the coldest region is at the stagnation point at the bottom of the rotor, and air temperature increases at the higher radii. The magnified view in the gap demonstrates that temperature drops in the periphery due to inflow of the cold air inside the gap.

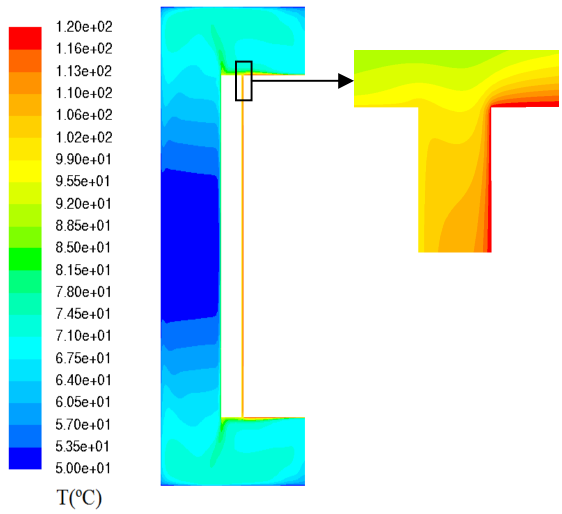

Figure 3 Temperature contour in the air gap for $Re=1.26\times10^5$ and $G=0.01333$

Figure 4 indicates the effect of the Reynolds number on the mean Nusselt number for the surfaces "*Stator facing rotor*" and "*Rotor facing stator*". It is seen that, for a fixed G=0.01333, the mean Nusselt numbers for both surfaces increase monotonically with the Reynolds number. As for the stator surface, increasing the Reynolds number results in more air recirculating in the gap which in turn improves the convective heat transfer.

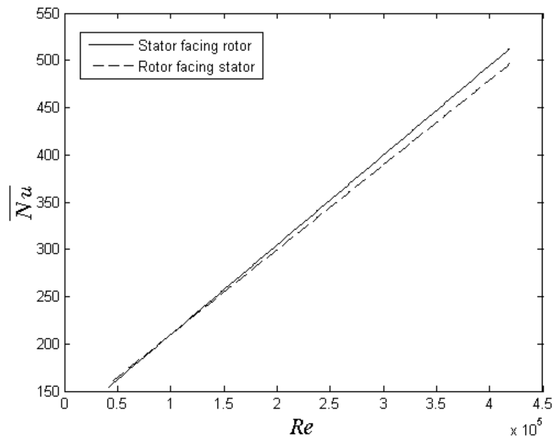

Figure 4 Effect of Re on the mean Nusselt number of the stator and the rotor's surface in the air-gap when $G=0.01333$

The effects of the gap size ratio on the mean Nusselt number in the stator's surface and the rotor's surface in the gap, when Reynolds number is kept at 1.26×10^5, have been illustrated in Figure 5. A sharp decrease in the mean Nusselt number can be observed as the G increases up to 0.03. At this point, the average Nusselt number remains almost unchanged when G goes up from 0.03 to 0.08. For the narrow gap, the flow structure is of a Couette-type flow so that an increase in the gap size results in a decrease in the temperature gradient which deteriorates the heat transfer. By contrast, for wide gaps, two separate boundary layers are constructed on both sides of the disks in the gap, known as Batchelor flow structure; that is why increasing the gap size has almost no influence.

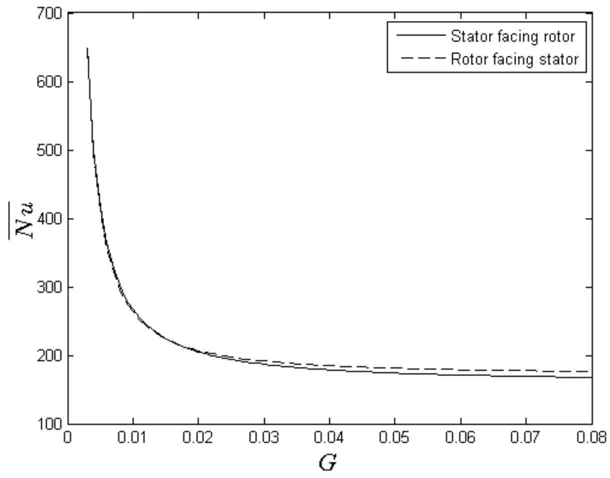

Figure 5. Effect of G on the mean Nusselt number of the stator and the rotor's surface in the air-gap when Re=1.26×10^5

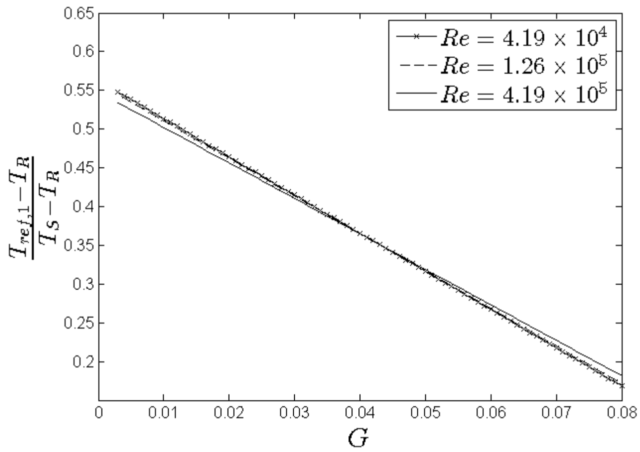

Figure 6 Non-dimensional mean fluid temperature in the gap versus G for different Re numbers

Figure 6 demonstrates the variations of non-dimensional bulk fluid temperature in the gap for different gap size ratios and Reynolds numbers. Note that the surface temperature of the stator, the rotor and the cover are kept at 120 ºC, 100 ºC and 50 ºC, respectively. The same trend reported by Howey et al. [7] can be seen here: the core fluid in the air gap cools down as the gap size increases. This is because the wider gap allows the cold air penetrating to the gap and it decreases the core fluid temperature accordingly. On the other hand, it seems that the core fluid temperature cools down with the Reynolds number in the narrow gap sizes, whereas the opposite trend can be seen for the wide gap sizes ($G > 0.04$).

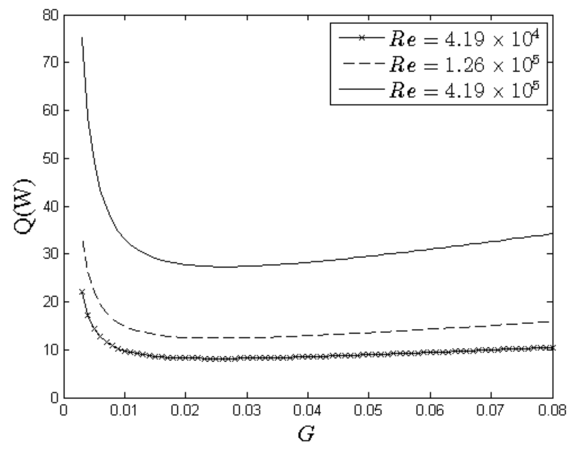

Figure 7 Average heat transfer of the stator surface in the gap versus G

Figures 7 demonstrates the variations of the average heat transfer of the stator surface in the gap versus G at different Re numbers when T_S=120ºC, T_R=100ºC, T_C=50ºC & D=150 mm. It is found that there is a sharp decline in the average heat transfer rate profiles as G rises until it reaches 0.02 (narrow gap size). When the gap size goes up further, there is a slight growth in the average convective heat transfer on the stator's surface. This implies that there exists an air gap size for which the convective heat transfer on the stator surface reaches its minimum that at first glance it seems to be incompatible with the results given by Yuan et al. [6]. To interpret this inconsistency, the attention should be given to the difference between the flow patterns in the air gap. In our case the flow structure is basically either a Couette-type flow or Batchelor flow structure, while in the study reported by Yuan et al. [6] the air gap size is wide enough and the flow pattern is a Stewartson-type, which means there is a boundary layer formed on the rotor wall but none on the stator wall.

A very interesting conclusion can be made out of figures 5, 6 and 7. By only considering the mean Nusselt number variations (see Figure 5), one may conclude that the convective heat transfer keeps slightly decreasing as the gaps size goes up from 0.02 to 0.08. In reality, however, the core fluid temperature becomes colder (Figure 6) and this results in an increase in the convective heat transfer profiles: Figure 7. Thus, the possible implication is that knowing the core fluid temperature variations in the air gap are vital in evaluating the convective heat transfer of the stator surface.

Validation of the Proposed Correlation

To check the validity of our analytical modeling, a comparison has been made with the experimental results of Yuan et al. [6] and Howey et al. [7], shown in Figure 8. It is worth mentioning that since the reference fluid temperature in our case is different than in their test runs, the absolute values of heat transfer rate from the stator surface in the gap has been taken into account. For the case of G=0.0976 and Re=3.33×10^5, an excellent agreement can be observed between our result and Yuan et al. [6], though the extrapolation has been done for this high value of G.

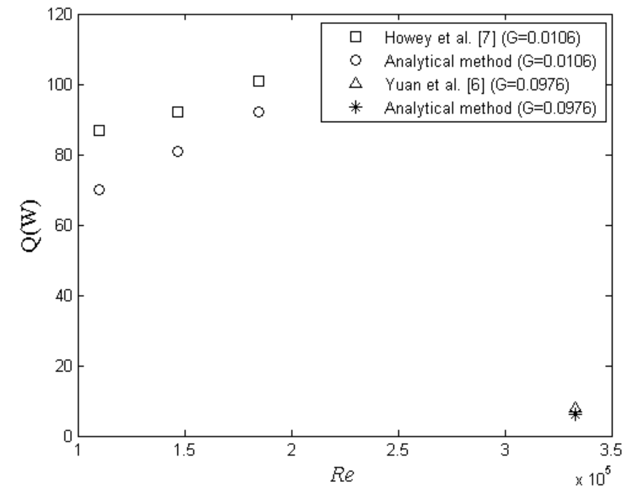

Figure 8 Comparison of the analytical results for the heat transfer rate from stator surface in the gap with the literature

The comparison between our results with Howey et al. [7] indicates that although the trends are similar, their results overestimate the heat transfer on the stator surface. The main reason for this discrepancy relates to the undertaken study zone for calculating the average Nusselt number, which in their experiment lies in the range of $0.6 \leq r/R \leq 1$, while the entire stator disk is considered in our analytical modeling. In other words, heat transfer from the stator surface mainly occurs in the higher radii due to ingress of cold air in the gap. On the other hand, there is a central admission of air without an imposing airflow at the stator in the measured values by Howey et al. [7], and it was found to be beneficiary for the stator heat transfer, whereas in our case the stator is without central opening. As a result, it is logical that their study overestimates the stator heat transfer prediction as compared to the analytical modeling presented here.

CONCLUSION

An empirical correlation has been successfully presented in order to estimate the convective heat transfer in the rotor-stator system of a disk type electrical machine. In the range of range of rotational Reynolds number $4.19\times10^4 \leq Re \leq 4.19\times10^5$ and the gap size ratio $0.00333 \leq G \leq 0.08$, CFD simulations were implemented. Having the numerical results, an analytical model was constructed with the help of the least square method and the curve fitting. Considering the rotational Reynolds number, the gap size ratio and the surface temperature of the stator, the rotor, and the cover as input, the analytical modeling is able to fully predict the average convective heat transfer for the stator and the rotor surface in the gap. The comparison between our results with the data available in the literature shows a good agreement.

The importance of considering the bulk fluid temperature in the gap as the reference temperature to calculate the mean Nusselt number was discussed. Although most studies in the literature, for simplicity, considered the ambient temperature as the reference temperature, here in order to evaluate the bulk fluid temperature, a linear correlation between the surface temperature of the rotor, the stator and the cover is made. In fact, the proposed correlations in the literature to estimate the heat transfer coefficient is only applicable for a given surface temperature of the rotor and the stator as well as the ambient temperature which is a serious drawback. On the contrary, the correlation presented here can be used for various surface temperature combinations. It was also shown that the variation of the mean Nusselt number and the bulk fluid temperature are equally important to assess the heat transfer rate on the stator surface.

The influence of the rotational Reynolds number and the gap size ratio on the convective heat transfer for the both surfaces in the gap, namely “rotor facing stator” and “stator facing rotor” was investigated. The results reveal that the mean Nusselt number increases with the Reynolds number, whereas there is a sharp decrease in the mean Nusselt number with the gap size ratio, for the narrow gaps. It can be also concluded that there is a gap size ratio for which the average convective heat transfer for the stator surface reaches a minimum.

REFERENCES

[1] Howey A., Childs R.N., and Holmes A.S., Air-Gap Convection in Rotating Electrical Machines, *IEEE Transaction on industrial electronics,* Vol. 59, 2012, pp. 1367-1375.

[2] Debruyne C., Sergeant P., Desmet J., Vandevelde L., Derammelaere S., Pyrhönen J., and Polikarpova M., Evaluation of the efficiency of Line Start Permanent Magnet Machines as function of the operating temperature, *IEEE Transactions on Industrial Electronics*, Vol. 61, No. 8, 2014, pp. 4443-4454

[3] Karabay H., Wilson M., and Owen J.M., Predictions of effect of swirl on flow and heat transfer in a rotating cavity, International Journal of heat and fluid flow, vol. 22, 2001, pp. 143-155.

[4] Jacques R., Le Quere P., and Daube O., Axisymmetric numerical simulations of turbulent flow in rotor stator enclosures, International Journal of heat and fluid flow, Vol. 23, 2002, pp. 381-397.

[5] Daily J., and Nece R., Chamber dimension effect on induced flow and frictional resistance of enclosed rotating disks. Journal of basic engineering, Vol. 82, 1960, pp. 217-232.

[6] Yuan Z.X., Saniei N., and Yan X.T., Turbulent heat transfer in the stationary disk in a rotor-stator system, *International journal of heat and mass transfer,* Vol. 46, 2003, pp. 2207-2218.

[7] Howey D.A., Holmes A.S, and Pullen K.R., Radially resolved measurement of stator heat transfer in a rotor–stator disc, *International Journal of Heat and Mass Transfe,* Vol. 53, 2010, pp. 491–501.

[8] Tindall C.E., Maynes B.D.J., Kee R.J., and Kenny R.G., Simulation of airflow and heat transfer in small alternators using CFD, IEE Proceedings on Electric Power Applications, Vol. 150, 2003, pp.146–152.

[9] Lim C.H., Airoldi G., Dominy R.G., Ingram G.I., Mahkamov K., Brown N.L., Mebarki A., and Shanel M., Experimental and CFD investigation of a lumped parameter thermal model of a single-sided, slotted axial flux generator, *International journal of thermal science*, Vol. 49, 2010. pp. 1732-1741.

[10] Pickering S.J., Lampard D., and Shanel M., Modelling ventilation and cooling of the rotors of salient pole machines, *IEEE International Electric Machines and Drives Conference,* 2001, pp. 806-808.

[11] Chong Y.C., Magahy D.A., Chick J., Mueller M.A., Staton D.A., and McDonald A.S., Numerical Modelling of an Axial Flux Permanent Magnet Machine for Convection Heat Transfer, *Renewable Power Generation (RPG 2011), IET Conference,* 2011, pp. 1-6.

[12] Moradnia P., Golubev M., Chernoray V., and Nilsson H., Flow of cooling air in an electric generator model-An experimental and numerical study, *Applied energy*, Vol. 114, 2014, pp. 644-653.

[13] ANSYS FLUENT code, http://www.ansys.com/Products/Simulation+Technology/Fluid+Dynamics/Fluid+Dynamics+Products/ANSYS+Fluent.