International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:11, No:1, 2017

CFD-Parametric Study in Stator Heat Transfer of an Axial Flux Permanent Magnet Machine

Alireza Rasekh, Peter Sergeant, Jan Vierendeels

***Abstract*—**This paper copes with the numerical simulation for convective heat transfer in the stator disk of an axial flux permanent magnet (AFPM) electrical machine. Overheating is one of the main issues in the design of AFMPs, which mainly occurs in the stator disk, so that it needs to be prevented. A rotor-stator configuration with 16 magnets at the periphery of the rotor is considered. Air is allowed to flow through openings in the rotor disk and channels being formed between the magnets and in the gap region between the magnets and the stator surface. The rotating channels between the magnets act as a driving force for the air flow. The significant non-dimensional parameters are the rotational Reynolds number, the gap size ratio, the magnet thickness ratio, and the magnet angle ratio. The goal is to find correlations for the Nusselt number on the stator disk according to these non-dimensional numbers. Therefore, CFD simulations have been performed with the multiple reference frame (MRF) technique to model the rotary motion of the rotor and the flow around and inside the machine. A minimization method is introduced by a pattern-search algorithm to find the appropriate values of the reference temperature. It is found that the correlations are fast, robust and is capable of predicting the stator heat transfer with a good accuracy. The results reveal that the magnet angle ratio diminishes the stator heat transfer, whereas the rotational Reynolds number and the magnet thickness ratio improve the convective heat transfer. On the other hand, there a certain gap size ratio at which the stator heat transfer reaches a maximum.

***Keywords*—**Axial flux permanent magnet, CFD, magnet parameters, stator heat transfer.

I. Introduction

THE AFPM machines are highly efficient devices that have been widely used in the applications where the axial length of the machine is a limiting factor [1]. Due to their compact construction, overheating, which is one of the main issue in the design of electrical machines, becomes much more severe and this needs to be prevented. In fact, demagnetization occurs in the permanent magnet materials once the temperature exceeds 150 ºC (for SH type NdFeB magnets) [2]. The resistivity of the copper winding in the stator disk increases with temperature and this deteriorates the efficiency of the machine. Moreover, the lack of insight about the thermal performance of the machine entails the designer to provide excessive cooling by external pump or ventilator to avoid overheating. For these reasons, it is vital to have a deep knowledge of the cooling in the AFPM machines.

Thermal study of the AFPM machines has gained much attention over the past few years [3], [4]. Moradnia et al. [5] implemented the CFD modeling of an electrical machine by MRF method. It was reported that this approach is roughly useful as it makes very accurate prediction of the airflow characteristics in the machine. Pyrhönen et al. [6] improved the cooling of the AFPM machines by using copper bars as extra heat carriers in the construction. Chong et al. [7] performed the numerical modeling of the direct-drive AFPM generator operating at 100 rpm for power level of 25 kW and built the experimental set up to validate their results. They showed that some ventilation openings at the side of the machine could enhance the convection heat transfer in the machine. Lim et al. [8] conducted an experiment to assess the convective heat transfer coefficient in an AFPM machine. They indicated that the effects of natural convection are significant in thermal evaluation of the machine.

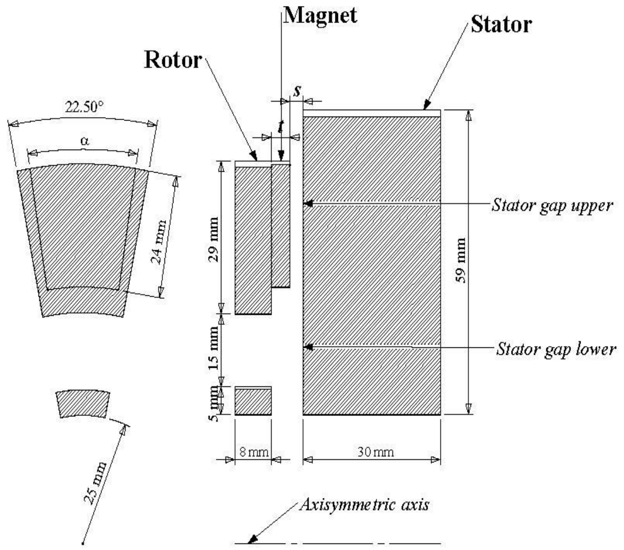
In an AFPM machine, most heat losses occur in the stator winding and the main heat transfer mechanism to expel the heat from this region is by convection. Therefore, the convective heat transfer in the stator disk of the AFPM machines has been investigated in the literature. In this regard, Yuan et al. [9] investigated the turbulent heat transfer on the stator surface and the flow characteristics in the gap between the disks. They showed that there is an optimum rotor-stator distance where the heat transfer on the stator side reaches a maximum. Howey et al. [10] measured the stator heat transfer in a discoidal configuration using an electrical heater array. They found that the local Nusselt number increases at the periphery due to ingress of air toward the stator. Rasekh et al. [11] studied the effects of holes in the rotor side on convective heat transfer on the stator facing rotor in the gap. It was concluded that the presence of holes is beneficiary for the stator heat transfer as air is allowed to enter into the air-gap through the holes, resulting in a net radial flow in the gap region between the rotor and stator.

Recently, Rasekh et al. [12] presented correlations for the convective heat transfer in the simplified discoidal system of an AFPM machine. The model is able to predict the convective heat transfer for different values of the rotational Reynolds number and the gap size ratio. The average bulk fluid temperature adjacent of each surface was utilized as the reference temperature instead of the ambient temperature to calculate the convective heat transfer coefficient. By doing so, the correlations for the surfaces in the gap became

Alireza Rasekh is with the Department of Flow, Heat and Combustion Mechanics, Faculty of Engineering and Architecture, Ghent University, Ghent, Belgium (phone: +32-489-305832, fax: +32-926-43590; e-mail: alireza.rasekh@ugent.be,).
Peter Sergeant is with the Department of Electrical Energy, Systems and Automation, Faculty of Engineering and Architecture, Ghent University, Ghent, Belgium (e-mail: peter.sergeant@ugent.be).
Jan Vierendeels is with the Department of Flow, Heat and Combustion Mechanics, Faculty of Engineering and Architecture, Ghent University, Ghent, Belgium (e-mail: jan.vierendeels@ugent.be).

independent of the surface temperature as well as the ambient temperature.

In this paper, the presence of the magnets on the rotor disk of the AFPM machine is taken into account. The objective is to find the correlations for the convective heat transfer coefficients of the stator surfaces according to the geometrical parameters of the magnets. A minimization method is used to calculate the appropriate value of the reference temperature, which makes these correlations independent of the surface temperatures. Details of the proposed method and the results are discussed in the following sections.



(a)

(b)

Fig. 1 (a) Schematic diagram of the rotor-stator system in the machine, (b) the real rotor disk with magnets

II. Design Properties of the AFPM Machine

The geometrical details of the rotor-stator configuration together with the actual rotor disk have been illustrated in Fig. 1. The system is composed of one stator disk encompassed by two rotor disks in which only half of the whole system is taken into consideration. The radii of the rotor and the stator are 74 mm and 84 mm, respectively. There are 16 magnets on the rotor disk which are evenly distributed. The space between the magnets creates the radial air-channel, which would be beneficial to the cooling of the machine. This can be further improved by the annular opening at the entire rotor disk by allowing the cold air to pass the space between the rotor and the stator. Due to the cyclic flow, only one slice of the discoidal system containing one complete magnet is studied.

The flow and heat transfer characteristics in this system can be defined through four non-dimensional numbers including the air-gap ratio, $G=s/R$, the rotational Reynolds number, $Re=\omega R^2/\upsilon$, the magnet angle ratio, $\alpha_m=\alpha\times 16/360$, and the magnet thickness ratio, $L=t/R$, where s is the air-gap thickness, R denotes the radius of the rotor, ω is the angular velocity of the rotor, υ is the kinematic viscosity of air, α is the magnet angle, and t denotes the magnet thickness. For the AFPM machine under study, these parameters are ranged at $0.0068 \leq G \leq 0.0811$, $3.5\times10^4 \leq Re \leq 3.5\times10^5$, $0.7 \leq \alpha_m \leq 0.9$ and $0.027 \leq L \leq 0.0811$.

The thermal conductivities of the rotor and the stator surfaces are rather high so that the surfaces are considered isothermal. The variations of air properties such as viscosity and density with temperature are taken into account by the Sutherland's viscosity law and the incompressible ideal gas formulation, respectively. Note that the Reynolds number is evaluated at the average temperature of the rotor, the stator and ambient, that is, $(T_r+T_s+T_{amb})/3$.

III. CFD Simulation

The commercial CFD software ANSYS FLUENT is used to solve the flow field and heat transfer characteristics. The turbulence is modeled through the *k-ω SST* model. The MRF which yields the steady state solution, is applied to model the motion of the rotor. It should be mentioned that the computational mesh is fixed in this approach, and the flow in each domain is solved using the moving reference frame equations. The 3D computational domain is made of the hexahedral mesh type with 2,944,000 nodes. It was found that the numerical modeling is independent of the mesh size, by comparing the heat flux on the stator surface in the gap with the coarse and the fine mesh sizes including 478,800 and 10,605,250 nodes, respectively.

IV. Flow and Temperature Profiles in the Gap

To have a better understanding of the heat transfer in this system, it is necessary to find out the various flow structures being formed in the air-gap between the magnets and also the stator surface. Therefore, Fig. 2 depicts the radial velocity contour in the orthogonal plane through the middle of the magnets and in the meridional plane through the center of the air-channel for G=0.0135, Re=1.06×10^5, α_m=0.8, and L= 0.0540. As seen, the magnets act as a centrifugal fan, accelerating the airflow outward through the air-channel. On the other hand, it is interesting to see the temperature profiles between the magnet and stator surface in the gap. Thus, the variations of the non-dimensional temperature profiles along the air-gap between the magnet and the stator surface at different radii and gap size ratios have been illustrated in Fig. 3.

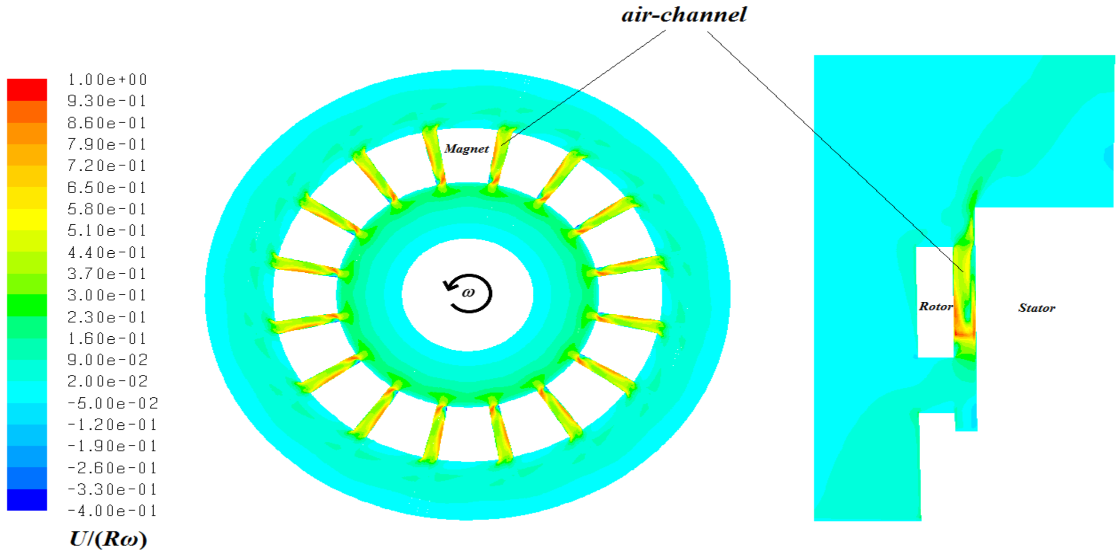


Fig. 2 Non-dimensional radial velocity contour in the orthogonal plane through the middle of the magnets and in the meridional plane through the center of the air-channel at G=0.0135, Re=1.06×10^6, α_m=0.8, and L=0.0540

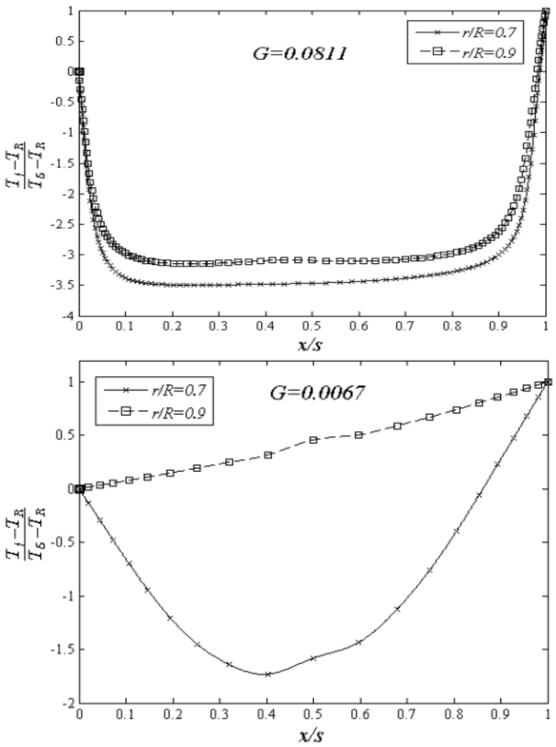


Fig. 3 Non-dimensional temperature variations across the air-gap for different G at Re=1.06×10^6, α_m=0.8 and L=0.0540

Note that the right side of the horizontal axis represents the area adjacent to the stator surface in the gap, and the left side belongs to the magnet surface. It is clear that the thermal boundary layers next to the magnet and the stator surface are merged, whereas for the wide gap thickness, two distinguishable unmerged thermal boundary layers can be noticed. In addition to that, the slope of the graphs in the vicinity of the stator surface (right side of the horizontal axis) shows that the heat transfer increases at the outer periphery of the gap region.

V. Correlations for Convective Heat Transfer Coefficient

The objective of commencing the CFD simulations is to acquire data to construct correlations for the convective heat transfer coefficient at the stator surfaces; namely, "*Stator gap upper*" and "*Stator gap lower*". The proposed correlations are able to assess the average heat flux for the mentioned surfaces with respect to the significant non-dimensional parameters including the gap size ratio, the rotational Reynolds Number, the magnet angle ratio, the magnet thickness ratio at different surface temperatures of the rotor and the stator.

For an isothermal surface, the convective heat transfer coefficient can be given by,

$$h = \frac{\dot{q}}{(T_{sur} - T_{ref})} \tag{1}$$

where $\dot{q}$ is the heat flux, T_{sur} is the surface temperature and T_{ref} denotes the reference temperature. The Nusselt number can be obtained as,

$$Nu = \frac{hR}{k} \tag{2}$$

R represents the radius of the stator and k is thermal conductivity of air. In order to make the convective heat transfer coefficient independent of the surface temperature, the suitable value of the reference temperature should be considered. Here, it is assumed that the reference temperature can be correlated with the surface temperatures and the ambient temperature through the following linear formulation,

$$T_{ref} = aT_r + bT_s + [1-(a+b)]T_{amb} \tag{3}$$

where the subscripts *r, s,* and *amb* denote the rotor, the stator and the ambient. The unknown coefficients of *a* and *b* are dependent on *G, Re,* α_m and *L*. By substituting (3) to (1), then the result to (4), the Nusselt number is rewritten as,

$$Nu = \frac{\dot{q}}{(T_{sur} - aT_r - bT_s - [1+(a+b)]T_{amb})k} \tag{4}$$

CFD simulations are carried out with the non-dimensional parameters kept at the reference point, i.e. G=0.0135, Re= 1.06×10^5, α_m=0.8, and L=0.0540 at four combinations of the surfaces temperature, shown in Table I. Given the heat flux from the numerical results, the unknowns in (3) will be *a*, *b*, and *Nu*. The idea is to find certain values of *a* and *b* to minimize the standard deviation σ of the Nusselt number in (3) as,

$$\sigma = [\frac{1}{n-1}\sum_{i=1}^{n}(Nu_i - \overline{Nu})^2]^{1/2} \tag{5}$$

According to Table I, i=4 indicates the number of surface temperature combinations. A Pattern-Search algorithm from the MATLAB Optimization Toolbox is employed to find the global minimum of the standard deviation of *Nu* in (4). The initial points are the results of the solution by setting *a* and *b* to 0.001 in the first place. Once the minimization algorithm yields the solution, the coefficients *a* and *b* will be defined. The mean value of the Nusselt number in (4) will be considered as the desired value for the Nusselt number. In this case, so long as the standard deviation resulting from (3) is small, the Nusselt number becomes nearly independent of the surface temperatures.

TABLE I
MINIMIZATION METHOD TO FIND A AND B FOR THE SURFACE "HOLE LOWER"

case	T_s (ºC)	T_r (ºC)	$\dot{Q}(w)$	a	b	T_{ref}(ºC)	Nu_i	$\overline{Nu}$	σ
1	120	100	123.4	0.415	0.001	56.2	469.4	466.9	9.1
2	120	80	137.3			47.9	462.0		
3	130	110	133.2			60.4	464.0		
4	100	70	109.5			43.8	472.4		

Having known *a, b*, and *Nu* at the reference point according to Table I, the goal is to find these as *G, Re,* α_m and *L* as well as the surface temperature of the rotor and the stator vary. To achieve this, a set of CFD simulations are carried out for *G*= {0.0068, 0.0101, 0.0270, 0.0405, 0.0541, 0.0676 & 0.0811} at four temperature combinations of the rotor and the stator, while the other parameters are kept at the reference point, i.e. Re=1.24×10^5, α_m=0.8, and L=0.0540. Thus, the heat fluxes for stator surfaces at all cases are computed. Once more, the minimization algorithm is used to assess the appropriate values of *a* and *b* for each case separately.

As a consequence, variations of the convective heat transfer with *G* when the other parameters are fixed at the reference point are given. In the same way, the cases where *Re,* α_m, are *L* are individually subjected to change will be investigated. Finally, we intend to formulate the variations of *a, b,* and *Nu* as follows,

$$\begin{aligned} a &= F(G, \mathrm{Re}, \alpha_m, L) \\ b &= G(G, \mathrm{Re}, \alpha_m, L) \\ Nu &= Y(G, \mathrm{Re}, \alpha_m, L) \end{aligned} \tag{6}$$

It is assumed that the above functions of four variables can be written as the product of four functions of one variable as,

$$\begin{aligned} a &= a^* f_1(G)\times f_2(\mathrm{Re})\times f_3(\alpha_m)\times f_4(L) \\ b &= b^* g_1(G)\times g_2(\mathrm{Re})\times g_3(\alpha_m)\times g_4(L) \\ Nu &= Nu^* y_1(G)\times y_2(\mathrm{Re})\times y_3(\alpha_m)\times y_4(L) \end{aligned} \tag{7}$$

where the superscript $*$ represents the values *a, b,* and *Nu* at the reference point. To find all of the above functions with one variable, curve fitting for or each surface should be carried out one after the other. Note that output from each function at the reference point is almost equal to unity. The details of the formulations for stator surfaces in the gap are presented in Table II.

TABLE II
CORRELATIONS TO ESTIMATE THE CONVECTIVE HEAT TRANSFER

Surface:	Stator gap upper	Stator gap lower
a^*	0.4153	0.2317
b^*	0.0010	0.0010
Nu^*	374.53	243.90
f_1	$0.2824G^{-0.3381}-0.2002$	$0.2033G^{-0.373}$
f_2	$74.78Re^{-0.3734}$	$2.096\times10^6Re^{-1.422}+0.9092$
f_3	$2.803\alpha_m^{1.518}-0.9881$	$26.23\alpha_m^2-40.02\alpha_m+16.23$
f_4	$0.4205L^{-0.2972}$	$0.1982L^{-0.5497}$
g_1	$3.386\times10^{-16}G^{-7.289}+0.9843$	$0.009G+1$
g_2	$7.556\times10^{13}Re^{-3.218}+0.9953$	$1.824\times10^{-9}Re+1$
g_3	$1.731\times10^{-15}\alpha_m+1$	$1.731\times10^{-15}\alpha_m+1$
g_4	$-7.017\times10^{-11}L+0.9999$	$2.506\times10^{-10}L+1$
y_1	$-2.967G^{0.6937}+1.142$	$-0.6211G+1.005$
y_2	$1.12\times10^{-4}Re^{0.7824}+0.04018$	$3.289\times10^{-4}Re^{0.679}+0.1569$
y_3	$-0.7732\alpha_m^{5.583}+1.224$	$-0.8183\alpha_m^{6.34}+1.21$
y_4	$13.86L^{1.709}+0.9064$	$7.906L^{1.575}+0.9243$

VI. CORRECTNESS OF THE CORRELATIONS

The correlations presented in this paper have been obtained by varying one non-dimensional parameter while keeping the rest fixed at the reference point and repeating this for all the other variables. Hence, by comparing the results of the correlations with those obtained by CFD simulation for the cases in which neither *G, Re*, α_m, nor *L* is at the reference point, the efficacy of the proposed correlations can be ensured. As a result, the comparison has been made for four different cases, as shown in Table III.

TABLE III
COMPARISON BETWEEN THE CFD AND THE PROPOSED CORRELATIONS RESULTS FOR HEAT TRANSFER RATE (*W*).

Surface	Re= 8.9×10^4, G=0.0203, L=0.0338, α_m=0.77, T_s=115ºC, T_r=95ºC		Re= 1.21×10^5, G=0.0108, L=0.0642, α_m=0.82, T_s=125ºC, T_r=105ºC	
	Correlation	CFD	Correlation	CFD
Stator gap upper	97.7	106.1	137.8	143.4
Stator gap lower	33.0	36.4	45.0	45.3

It is seen that the correlations presented in this paper are able to assess accurately the convective heat transfer for the stator surfaces in the gap in the practical ranges of the important parameters in AFPM machines with a good accuracy.

VII. CONCLUSION

The convective heat transfer in the stator disk of an AFPM machine was studied in this paper. CFD simulations have been performed considering different ranges of the predominant non-dimensional parameters that govern the heat transfer and fluid flow; namely, the rotational Reynolds number, the gap size ratio, the magnet angle ratio and magnet thickness ratio. It was shown that the radial air-channel is constructed between the magnets and also in the gap region. This acts as a centrifugal fan to expel the heat from the system.

Based on the numerical results, the objective was to construct correlations to estimate the convective heat transfer coefficient, applicable to the different non-dimensional parameters as well as the surface temperature of the rotor and the stator. To do so, a minimization method by means of Pattern-Search algorithm has been employed to find the proper value of the reference temperature that makes the estimated convective heat transfer coefficient independent of the surface temperature. Afterwards, the correlations for calculating the convective heat transfer for the stator surfaces in the gap were given by using curve fittings. It was concluded that the correlations are scalable and quite versatile tools in the thermal modeling of AFPM machines.

ACKNOWLEDGMENT

The financial support for this study by FWO project G.0110.13 is gratefully acknowledged.

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