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A PARAMETRIC-CFD STUDY FOR HEAT TRANSFER AND FLUID FLOW IN A ROTOR-STATOR SYSTEM

ALIREZA RASEKH[1,*], PETER SERGEANT[2] AND JAN VIERENDEELS[1]

[1] Department of Flow, Heat and Combustion Mechanics, Faculty of Engineering and Architecture, Ghent University, Ghent, Belgium.
e-mail: Email: Alireza.Rasekh@UGent.be, Jan.Vierendeels@UGent.be

[2] Department of Industrial Technology & construction (EA20), Faculty of engineering and architecture, Ghent University, Ghent, Belgium.
e-mail: peter.sergeant@ugent.be

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Abstract. This study considers the turbulent heat transfer in the air-gap of an enclosed rotor-stator system subjected to the prescribed wall temperature. The aim is to improve the thermal performance of this discoidal system, and to do so a new cooling solution is investigated with holes introduced in the rotor. Multiple reference frame (MRF) and sliding mesh (SM) techniques are employed to model the rotor-stator interaction. The heat transfer rate and flow characteristics were calculated for the rotational Reynolds number $2.50\times10^4 \leq Re=\Omega R^2/\upsilon \leq 2.50\times10^5$ and the air-gap ratio $0.00667 \leq G = s/R \leq 0.02667$. The results reveal that the addition of the holes in the rotor is advantageous for the heat transfer and it can increase the heat transfer rate in the stator up to 25%.

1 INTRODUCTION

Over the past few decades, considerable attention has been paid to investigate the heat transfer in axial rotor-stator disk systems that are mostly crucial in disk type electrical machines, such as axial flux permanent magnet synchronous machines [1]. In an effort to avoid damage of components arising from local high temperature due to electromagnetic losses, the designers must take into account the factors of heat transfer including the shape, the rotational speed and the air-gap size between rotor and stator. Thus, it is very important to improve cooling in order to maintain both a long life time and high reliability of such machines. In particular, the magnets should be sufficiently below the critical temperature (T_c=150°C) to avoid demagnetization. Due to the lack of precise knowledge about the heat transfer, external cooling is supplied to the discoial system, resulting in an excessive energy consumption of the external ventilator and consequently a deteriorated energy efficiency. Thus, the acquisition of the local convection information in rotor-stator system is of paramount importance.

Numerous experiments have been conducted for the heat transfer and the fluid structure prediction in discoidal systems [2-5]. Nikitenko [6] studied an air-filled enclosed rotor–stator system experimentally where both disks were isothermal. Kapinos [7] estimated the averaged Nusselt number on an air-cooled rotor, over which the temperature profile was approximately parabolic. Pelle´ and Harmand [8] conducted an experiment to find out the influence of an axial jet flow on the convective heat transfers on a rotor surface in an open discoidal system. Bunker et al. [9] used the transient liquid crystal technique to measure the radial distribution of the convective heat transfer coefficient on both rotor and stator disks. Howey et al. [10] employed an electrical heater array method for measuring the stator heat transfer in a rotor–stator disc. They observed that the heat transfer rate increases at the periphery due to the rotor pumping effect.

On the other hand, numerical models of the rotor-stator configuration are also available in the literature [11-15]. Roy et al. [16] studied the fluid temperature distribution and the convective heat transfer coefficient distribution on the rotor disk in a rotor–stator cavity with both mainstream and secondary air flow present. Iacovides and Chew [17] used four different models of turbulence to study the convective heat transfer in three axisymmetric rotating disk cavities with throughflow. They also compared their results with those of available in the literature and concluded that the k–ε model with the one-equation near-wall treatment is preferred. The problem of heat transfer in the rotor-stator system subjected to a superimposed throughflow was investigated by Poncet and Schiestel [18]. Their numerical simulation was based one one-point statistical modeling using a low Reynolds number second order full stress transport closure. Poncet and Serre [19] examined the turbulent flow in an enclosed rotor–stator system using large eddy simulations (LES). Numerical simulation of turbulent flow in an open rotor-stator system was studied by Yuan et al. [20]. Their results revealed that there exists an optimum rotor–stator distance for a given Reynolds number, at which the average heat transfer on the stator reaches a maximum.

The objective of this paper is to numerically investigate the turbulent heat transfer in an enclosed discoidal system with prescribed surface temperature. The effects of the rotational Reynolds number, the disk-distance and presence of holes in the rotor on the fluid structure in the air-gap and the heat transfer will be addressed.

2 PROBLEM SETUP

The configuration of the problem under investigation has been illustrated in the fig. 1 where the left disk represents the rotor and the right one is stator. The case study consists of two rotors and one stator. Hence, a symmetry plane has been defined in the middle of the geometry to halve the computational costs. The flow is characterized by the rotational Reynolds number, $\mathrm{Re} = \Omega D^2 / 4\upsilon$, and the air-gap ratio, *G*=*s/R*, where *s* and *R* are defined in fig. 1, Ω is the angular velocity of the rotor and ν is the kinematic viscosity of air. The surface temperature of the rotor, the stator and the cover are kept at 100 ºC, 120 ºC and 50 ºC, respectively. In the range of $2.50\times10^4 \leq \mathrm{Re} \leq 2.50\times10^5$ and $0.00667 \leq \mathrm{G} \leq 0.02667$ the heat transfer rate and the flow characteristics in the gap between the disks were calculated. Moreover, air was considered as an incompressible ideal gas, so the density variation due to temperature differences was considered.

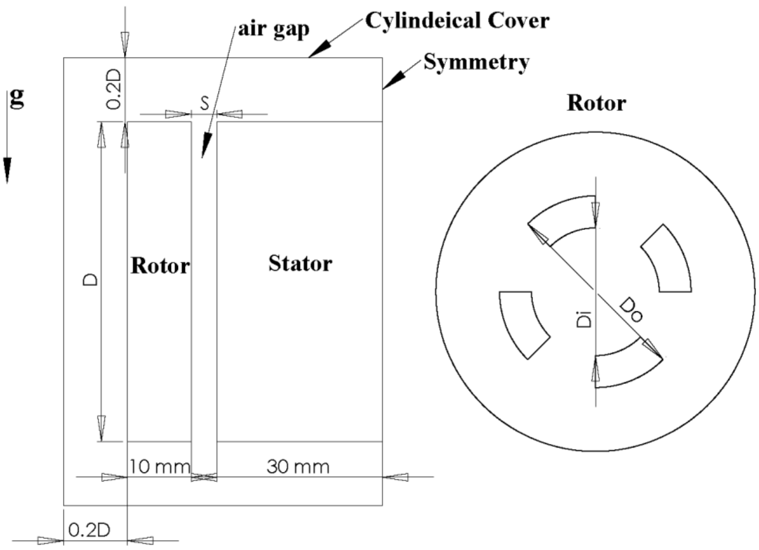

Figure 1. Problem configuration.

2 CFD SIMULATIONS

The commercial CFD software Ansys FLUENT has been used to simulate the 3D flow fields. 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 5. 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.

Rotor-stator interaction has been modeled using multiple reference frame (MRF) and sliding mesh (SM) techniques. The former case involves steady state computations and produces time averaged flow field, whereas the latter one involves transient computations to produce time accurate flow field. It should be noted that in the MRF technique the mesh remains fixed for the computation and the flow in each moving cell zone is solved using the moving reference frame equations. This is analogous to freeze the motion of the moving part in a specific position and observing the instantaneous flow field with the rotor in that position. Contrary to the MRF approach, the SM method cannot neglect unsteady rotor-stator interaction and it accounts for the relative motion of stationary and rotating components. Although the SM method is computationally demanding, it takes into account the effect of the changing position of the holes during rotation. We utilized the flow field and the heat transfer results of the MRF model as an initial condition for a transient sliding mesh calculation. To check the independency of the grid size and time step, several exploratory simulations were carried out. Once the length of the first cell in the vicinity of the wall is 100 μm and each time step corresponds to the rotation of the rotor equal to 0.5º, the results are independent of the grid and the time step.

Several CFD runs were implemented for different combinations of geometrical variables including the rotor & stator diameter, the air-gap size and the velocity of rotor sidewall. The base case accounts for $D = 150mm$, $s = 1mm$ and $V = 30m/s$ that is equivalent to Re=2.50×10^4 & G=0.0133. The results of the SM method are basically periodic with the value of any flow variable at a particular point in solution domain being dependent on the relative location of the holes in the rotor.

3 RESULTS AND DISCUSSION

The streamlines around the rotor-stator system have been depicted in the fig. 2 as the Reynolds number varies from 2.50×10^4 to 2.50×10^5 while the air-gap ratio is constant G=0.0133. Due to the buoyancy effect, the flow patterns are not presented symmetrically. Another interesting phenomenon is that the vortical eyes move to the rotor side as the Reynolds number goes up. The velocity vectors in the gap between the rotor and the stator has been illustrated in the fig. 3. It can be seen that a re-circulating region exist in the gap.

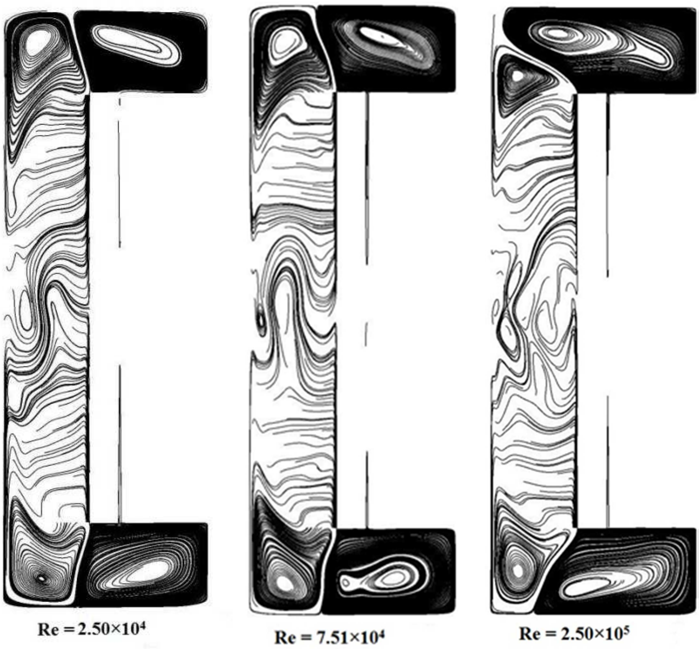

Figure 2. streamlines in the rotor-stator system at different Re number with fixed G =0.0133.

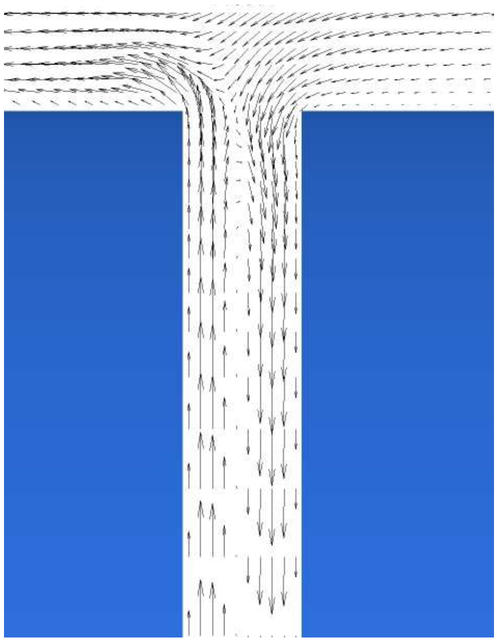

Figure 3. velocity vectors inside the gap between the rotor and the stator for Re=2.50×10^4 and G=0.0133.

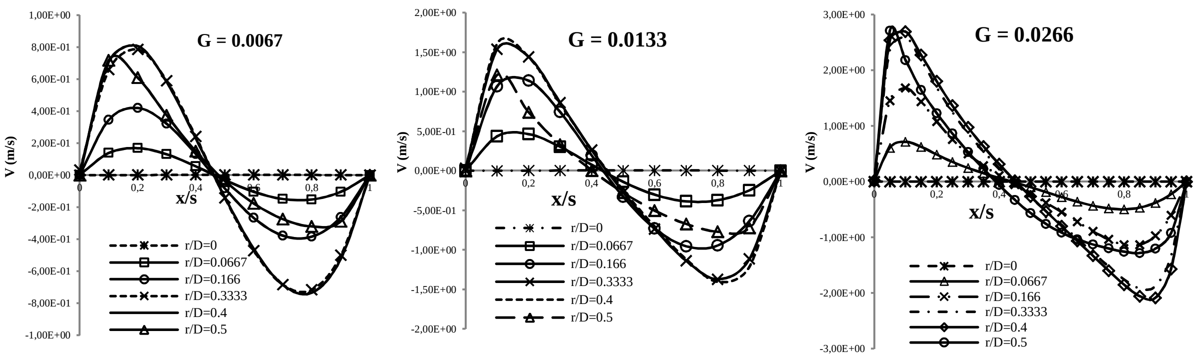

Figure 4. Radial velocity profiles inside the gap when Re=7.51×10^4.

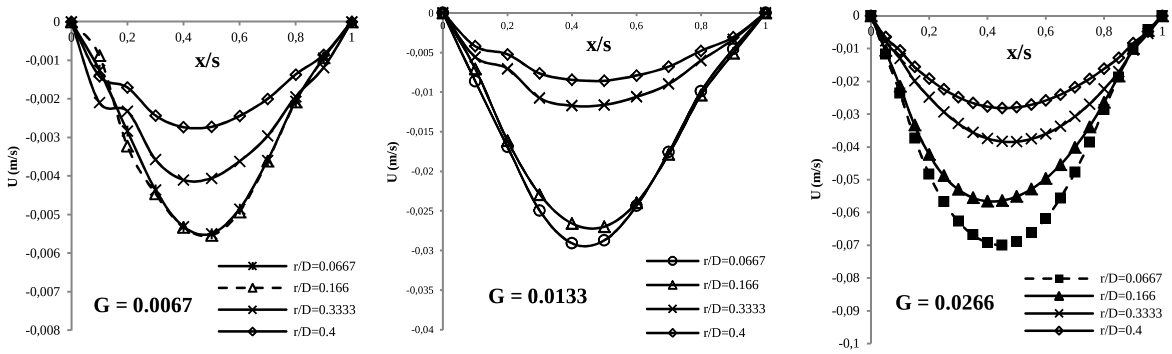

Figure 5. Axial velocity profiles inside the gap when Re=7.51×10^4.

Fig. 4 shows the variation of the radial velocity of air flow inside the gap along the rotational axis for different air-gap ratios at six radii. It should be mentioned that the negative values stand for inward flow. As seen, the air flow changes its radial direction inwards after crossing the point *x/s= 0.4* for all air-gap ratios. Fig. 5 gives the distribution of the axial velocity component U. Comparing to the radial velocity profiles, the absolute values of the axial velocity is much smaller. Note that the negative values of the axial velocity correspond to the flow direction toward rotor from stator. The reason for negative values of the axial velocity is that the fluid is drawn in the axial direction to the rotor to replace the fluid which has been pumped out.

Tab. 1 shows the influence of the Reynolds number together with the air-gap ratio on the heat transfer rate in our rotor-stator configuration. It is seen that there is an increase in the heat transfer as the Reynolds number rises from 2.50×10^4 to 2.50×10^5. In the similar manner, when the air-gap ratio increases from 0.00667 to 0.02667, the heat transfer rate enhances in this discoidal system.

Table 1. Effect of Reynolds number and air-gap ratio on heat transfer rate (W)

Re	G	Rotor wall	Stator wall	Cover wall
2.50×10^4	0.0133	10.0	23.3	-33.3
7.51×10^4		27.5	42.3	-69.8
2.50×10^5		72.8	90.7	-163.5
7.51×10^4	0.00667	23.4	40.0	-63.4
	0.0133	27.5	42.3	-69.8
	0.02667	36.2	50.9	-87.1

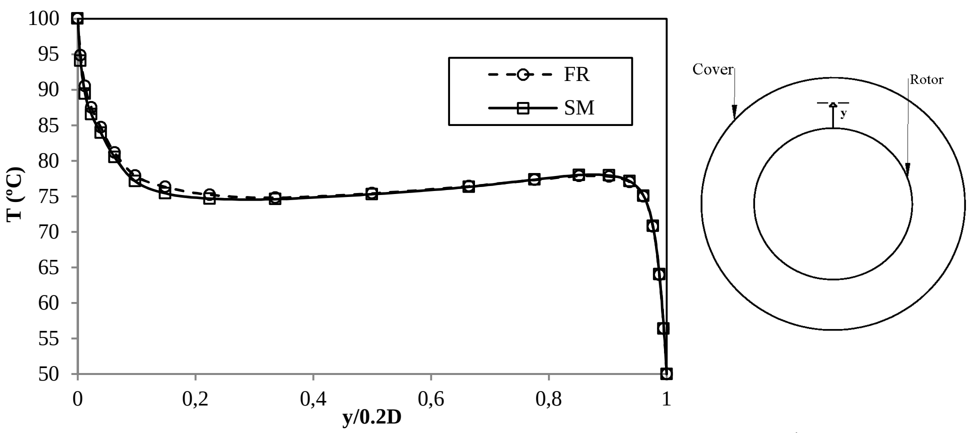

Figure 6. Temperature distribution along the line "y" for Re=7.51×10^4 and G=0.0133.

Here the presence of holes in the rotor is discussed. The simulation has been carried out through MRF and SM methods. A substantial physical simulation time of *t=1s* is required to get the fully periodic results of a SM simulation. Fig. 6 shows a comparison between the results of SM and MRF approaches for the temperature variation alongside the line “y”. It can be seen that there is a good agreement between these results. As expected, a thermal boundary layer effect is observed near the walls.

It is interesting to see how much air passes into holes and penetrates into the gap. To do so, the detailed information of the mass flow rate has been demonstrated in the tab. 2. It is observed that there is an infinitesimal discrepancy between the amounts of mass flow entering and leaving the holes which satisfies with the conservation of mass requirement. On the other hand, an excellent agreement can be seen between the results of SM method with those of obtained by MRF method.

Table 2. Mass flow rate (kg/s) passing through the holes.

	MRF method	SM method
Net inlet flow	0.0002034	0.0002101
Net outlet flow	0.0001992	0.0002158

To give more insight about the flow pattern inside the hole, the velocity vector for both the MRF and SM methods in a meridional plane of the rotor has been shown in the fig. 7. The left hand side of the plot is where the airflow penetrates to the gap and the right side illustrates how airflow enters into the gap. As seen, there is a re-circulating region in the middle of the hole.

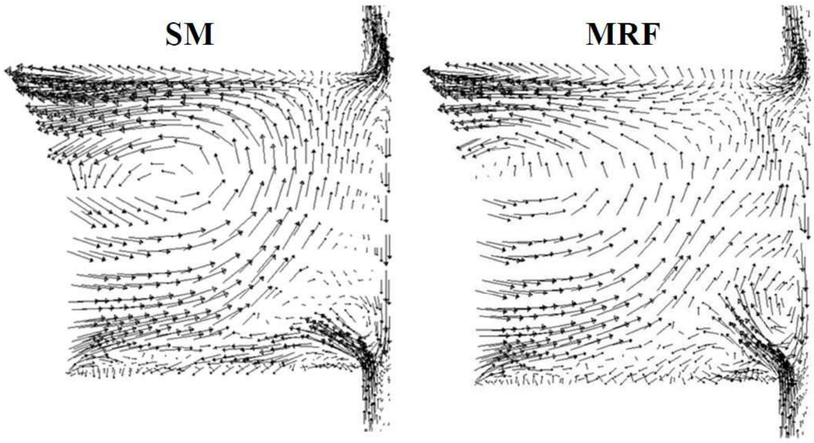

Figure 7. Velocity vectors inside the hole for Re = $7.16{\times}10^4$ and G = 0.02667.

Table 3. heat transfer rate (W) in the discoidal system when holes exist in the rotor.

		Rotor wall		Stator wall		Cover wall	
Re	G	MRF	SM	MRF	SM	MRF	SM
5.01×10^4	0.0200	21.6	21.1	31.3	31.0	-52.9	-52.1
7.51×10^4	0.0133	33.9	33.5	56.2	53.1	-90.1	-86.6
1.00×10^5	0.0100	51.0	49.7	85.1	79.9	-136.1	129.6

Tab. 3 represents the heat transfer rate in the rotor-stator system for different combinations of the Reynolds number air-gap ratio with the presence of holes in the rotor. Again, there is a good agreement between the results of MRF and SM methods. However, it seems that the results of MRF approach would be overrated in all cases. Furthermore, with a comparison between the results of tables 1&3, it can be realized that the presence of holes would be advantageous for the heat transfer in the discoidal system as the recirculating air in the gap would enhance the convective heat transfer.

4 CONCLUSION

In this paper, the thermal performance of a rotor-stator system has been assessed numerically. Based on the information obtained from CFD simulations, an insight has been provided into the turbulent heat transfer in the air-gap of the discoidal system. MRF and SM methods have been used for rotor-stator interaction. Afterwards, their results were compared and a remarkable agreement was observed. It can be concluded that the heat transfer rate has been improved with an increase in the rotational Reynolds number as well as the air-gap ratio. More importantly, the presence of the holes in the rotor was found to be beneficiary for the 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.

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