Eddy resolving simulation of mixed convection in a rotating annular cavity with one heated disk and axial throughflow: the effect of the surface macro-relief

A G Abramov, D K Zaitsev, E M Smirnov and E E Kitanina

Peter the Great St.Petersburg Polytechnic University,
Polytechnicheskaya str. 29, St.Petersburg, 195251 Russia

E-mail: smirnov_em@sbptu.ru

Abstract. We present the results of hybrid RANS/LES computations of non-isothermal buoyant flow in a rapidly revolving enclosure with paraxial transit stream of the cooling air. Foil heat flux meters mounted on the disk surface in the base experiment are mimicked by means of the grid resolved macro-relief. The results obtained using the relief and smooth disk models are collated with available measurements. According to the simulation, the addition of the relief has resulted in switching from two to three pairs of cyclonic/anti-cyclonic global circulations, and the overall heat transfer rate has increased by 20%. It has been found also that the sensor readings can be up to 25% higher than the heat flux averaged over the circumference at the same radius. Despite this distinct effect of the surface relief, the local heat transfer rate is still underestimated considerably as compared to measurements.

1. Introduction
Annular enclosures with heated disks and paraxial transit stream are widely used in axial compressors of gas turbines to cool the rotor body [1] so an adequate prediction of the wall heat exchange in such systems under various conditions is of major importance. Due to interaction between the buoyancy and Coriolis forces, the flow in rapidly revolving enclosure is usually very complex and unsteady whereas the dedicated measurements are typically limited to the time-averaged local wall heat flux or/and the overall heat flow data. Systematic measurements (including flow visualization) in rotating annular chambers with heated sidewalls (disks) were reported in [2, 3]. These configurations were investigated numerically in [4–6] using different 3D CFD codes but all the computations resulted in considerable (up to several times) underestimation of the heat transfer rate.

The poor accuracy of the CFD-based heat transfer prediction in rapidly spinning annular enclosures with heated disks seems to be a common tendency rather than the failure of a particular test. It is believed that the main reasons for the heat exchange underestimation are the insufficient grid resolution and the usage of RANS turbulence models that seem to be unsuitable for reproducing the flow and heat transfer in quasi-laminar or transitional Ekman layers close to the disks. However, recent computations using LES-based vortex-resolving methods and rather fine grids [7–9] didn’t lead to cardinal improvement of the heat transfer prediction accuracy for particular experimental configurations considered.

The present paper covers the results of CFD study of the air flow in a rapidly revolving enclosure investigated experimentally in [10] and numerically in [9]. The study is aimed at getting an estimation
of possible effect of measurement sensors (heat flux meters mounted on the disk surface) on the flow and, consequently, on the local and integral heat transfer in the enclosure. The computations are performed by means of ANSYS Fluent CFD code using same hybrid RANS/LES approach and same-resolution grids as in our previous study [9]; the sensors on the disk surface are modelled as macro-relief explicitly resolved by the grid. The present relief-disk simulation results are compared with the smooth-disk computations [9] and measurements [10].

2. Base experiment and computational model

Identically to the previous study [9], the enclosure geometry and the flow conditions are defined in accordance with the description given in [10] for experiment No.6. Figure 1a shows the sketch of the experimental rig [10]. The enclosure dimensions are: \( b = 371 \), \( a = 150 \), \( s = 75.2 \) (mm). One of the enclosure sidewalls (disks) is exposed to radiation heating from a stationary electric heater; the other one is made of translucent polycarbonate. The revolving enclosure and the heater are placed into a stationary shell; a window in the shell is arranged for LDA measurements. The cooling air flows axially along the inner cylinder rotating together with the enclosure. The radial clearance between the cylinder and the disks is 4.8 mm, the air inlet velocity is 4.3 m/s, the inlet temperature \( T_{in} = 300 \) K, the enclosure rotation speed, \( \Omega \), is 165 rad/s. Figure 1b shows the heated wall excess temperature, \( \Delta T_w = T_w - T_{in} \), as measured in [10].

![Figure 1. Sketch of the spinning enclosure experimental rig (a) and the radial distribution of the heated disk temperature (b).](image)

To provide the wall heat flux and temperature measurements, 10 RdF sensors (foil heat flux meters combined with thermocouples) were arranged on the surface of the heated disk, but no information on the sensors’ position or/dimensions was given in [10] and only 7 measurement points were shown in the plots (like in figure 1b). For lack of other information, in the present study we considered 7 sensors placed along single radius at the position of the measurement points shown in figure 1b. Moreover, based on the modern RdF micro-foil heat flux sensors catalogue, we assumed the sensor size of 1×1 cm and the thickness of 0.35 mm as a reasonable guess. Note that the adopted sensor thickness is of the same order as the Ekman layer characteristic thickness, \( (\nu \Omega)^{1/2} = 0.3 \) mm, i.e. the sensors are actually not so thin as it might appear and they are able to affect the near-wall flow and heat transfer.

The computational grid for the present study was created on the base of the finest one used in [9]. That grid consisted of 17.3 million hexahedral cells and it was uniform in the circumferential direction (800 points). In the meridional section, the maximum cell size was about 1.5×1.5 mm; the grid was
gradually refined toward the walls. At the disk surface, the first near-wall grid step was 0.05 mm that provided the normalized wall distance, \( y^+ \), being below unity throughout the disks.

In the present study, the grid was not precisely adjusted to the sensors. Instead, the cells whose centroid lay inside a sensor were removed from the initial grid, so that the resulting size of the grid-represented sensors differed slightly from the nominal one (1×1 cm). As well, to provide better resolution of the flow around sensors, the grid was additionally refined close to the heated disk (0.6 mm from the wall) via the mesh adaption tool of ANSYS Fluent. The adapted grid consisted of 22.4 million cells, with 12 cells per the sensor thickness and at least 8×8 cells over the sensor surface (figure 2b).

![Figure 2](image)

**Figure 2.** General view of the computational domain (a), and the computational grid near the sensors (b) and in the meridional section (c).

Most of numerical options used in [9] were maintained in the present study. In particular, the Embedded LES approach was used for turbulence modelling. Namely, the \( k-\omega \) SST turbulence model was used in the RANS zone around the inner cylinder (see figure 2c) but the WALE model was activated in the principal part of the enclosure (LES zone in figure 2c). The measured wall temperature (figure 1b) was applied all over the heated disk surface (including sensors); zero-heat-flux boundary condition was used for the other walls. A more detailed description of the flow conditions and the numerical options used is given in [9].

Taking into account the former experience [9], the physical time step of 0.4 ms was used in the present computations that provided appropriate values (mostly below unity) of the Courant number in the LES zone. Computation of one second of the flow took about three days using 60 CPU cores. The computations started from a fully developed flow obtained formerly for the smooth disk model (Case 5 in [9]). After two seconds (50 revolutions), the data sampling for time statistics option was activated and eight seconds more were simulated.

3. Simulation results

Shown in figure 3, the time evolution of the heat flux (averaged over the disk surface) evidences that the statistically steady-state flow was really achieved within the first two seconds, and the eight seconds sample was long enough to yield reliable statistics.
Figure 3. Time evolution of the computed heat flux over the whole disk.

Figure 4 displays instant distributions of the mid plane flow velocity and temperature ($\Delta T = T - T_w$) along with contours of the wall Nusselt number ($Nu = q_w K (\Delta T_w)^{-1}$). For comparison, corresponding flow fields obtained in [9] for smooth disk (without sensors) are given in figure 5.

Although the presented velocity fields are rather intricate, several global vortices with alternating cyclonic and anti-cyclonic circulation can be recognized (for clarity, these vortices are highlighted by “streamlines”). Altogether, six global vortices are present in figure 4a, whereas only four vortices were detected in [9] for the smooth disk (figure 5a). So, the addition of a few relatively small and rather thin sensors on the disk surface has resulted in considerable change of the global flow structure. It’s worth emphasizing here that, the flow pattern shown in figure 4 is not attached to the sensors angular position; actually, it displays slow counter-rotating (clockwise in figure 4) precession relative to the enclosure.

Figure 4. Instant distributions of the mid plane velocity vectors (a) and air temperature (b) and the wall Nusselt number (c) in the revolving enclosure with relief disk.

Figure 5. Same obtained in [9] for the smooth disk case.
In the temperature maps (figure 4b and 5b) one can catch sight of radial “arms” of cold air that conform to the outward flow regions between the cyclonic and anti-cyclonic vortices (consequently, there are three “arms” in figure 4b but two ones in figure 5b). Imprints of these “arms” are well observable in the wall Nusselt number distributions (figure 4c and 5c). In all, the mean temperature level in figure 4b is about two degrees higher than in figure 5b (note that the two color scales are different) that is consistent with higher typical values of the Nusselt number on the relief disk (figure 4c) as compared to the smooth one (figure 5c).

Figure 6 shows the time-averaged distributions of the Nusselt number and the flow rotation speed (in the stationary reference frame); the results obtained here for the relief disk are compared with the smooth disk computations [9] and experimental data [10].

Figure 6a evidences that the addition of the sensors has resulted in an increase of the disk local heat transfer rate by 20% approximately (the mean heat flux has increased from 565 to 671 W/m²). At that, the Nusselt number “measured” by sensors is up to 25% higher than the value averaged over the disk circumference at the same radius. So, the predicted sensor readings can be up to 50% higher than the heat flux obtained from the smooth disk simulation. Despite this distinct effect of the surface relief, the local heat transfer rate is still underestimated considerably (up to 2.3 times) as compared with measurements [10].

Given in figure 6b, the computed mid-plane velocity profiles exhibit rather weak effect of the sensors on the core rotation: in the outer part of the enclosure the velocities are nearly the same whereas in the inner part the difference doesn’t exceed 15% of the velocity lag. The distinction between the predicted velocity profiles is negligible as compared to the observed disagreement with experimental data [10], so the latter cannot be referred to the disk surface relief.

![Figure 6. Computed and measured time-averaged distributions of the wall Nusselt number (a) and the flow rotation speed (b).](image)

4. Conclusions

The non-isothermal buoyant flow in a rapidly revolving annular enclosure with paraxial transit stream of the cooling air has been computed using the hybrid RANS/LES approach (Embedded LES). To estimate possible influence of heat flux sensors mounted on the disk surface in the base experiment, a row of thin plates of reasonable dimensions was introduced into the computational model. The addition of the surface relief has resulted in switching from two to three pairs of cyclonic/anti-cyclonic global circulations, and the overall heat transfer rate has increased by 20%. It has been revealed also that the sensor readings can be up to 25% higher than the local heat flux values averaged over the circumference at the same radius. Unlike the heat transfer results, the computed mid-plane profiles of rotational velocity are almost insensitive to the presence of sensors. As compared to the
measurements, both the heat transfer rate and the rotational velocity lag are still underestimated considerably in the computations. In all, the surface relief effect on the heat transfer is found to be significant, but it cannot be the only reason for the observed systematic disagreement with the experimental data.

Acknowledgments
This study was supported by the Russian Foundation for Basic Research (grant no. 20-08-01090).

References
[1] Harmand S, Pelle J, Poncet S and Shevchuk I V 2013 *Int. J. Thermal Sciences* **67** 1–30
[2] Bohn D, Deuker E, Emunds R and Gorzelitz V 1995 *J. Turbomachinery* **117** 175–83
[3] Bohn D, Deutsch G, Simon B and Burkhardt C 2000 Proc. *ASME Turbo Expo* (Munich) GT2000-280
[4] Ivanov N G, Ris V V, Smirnov E M and Smirnovskii A A 2005 Proc. 3rd Int. Conf. Advanced Computational Methods in Engineering (Ghent) p 9
[5] Bohn D, Bouzidi F, Kitanina E, Ris V V, Smirnov E M, Burkhardt C and Wolff M W 2002 9th Int. Symp. Transport Phenomena and Dynamics of Rotating Machinery (Honolulu) p 8
[6] King M P and Wilson M 2005 *Numer. Heat Transf. A* **48** 529–45
[7] Smirnov P E, Kapetanovic S, Braaten M E, Egorov Y and Menter F R 2009 *ASME Paper* GT2009-59621.
[8] Atkins N R and Kanjirakkad V 2014 Proc. *ASME Turbo Expo* (Dusseldorf) GT2014-27174
[9] Abramov A G, Zaitsev D K, Smirnov E M and Smirnovsky A A 2020 *J. Phys.: Conf. Ser.* **1683** 022089 p 8
[10] Owen J M and Powell J 2006 *J. Eng. Gas Turbines Power* **128**(1) 128–34