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# Experimental investigation on two-dimensional heat transfer and secondary flow in a rotating smooth channel

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ABSTRACT

In the current study, we experimentally investigated the two-dimensional heat transfer with different thermal boundary conditions and secondary flow in a rotating smooth channel. The Reynolds number, based on the channel hydraulic diameter ($D = 80$ mm) and the bulk mean velocity ($U_m = 1.82$ m/s) is 10000, and the rotation number ranges from 0 to 0.52. The mean density ratio (d.r. $= (T_w - T_b)/T_w$) is about 0.1 in the current work using transparent heater glass (Indium Tin Oxide) to provide uniform heat flux. Three different thermal boundary conditions (Case A: four walls are heated, Case B: leading and trailing walls are heated, and Case C: only leading or trailing wall is heated) are taken into consideration to investigate the heat transfer distribution on the leading and trailing side. The heat transfer results show that thermal boundary condition has a significant effect on the details of the distribution of heat transfer on the leading and trailing side, especially on both end of the Z/D directions. The ratio $Nu/Nu_0$ of case B is about 10% larger than that of case A on leading side, and the difference between case B and case C is not obvious. To explain the phenomena, secondary flow in the rotating channel without wall heated was measured by PIV. The vortex-pair is found on the cross-section. Two small vortex-pair appear with the increase of rotation numbers, and then the intensity of the small vortex-pair is weakened near the leading side when the rotation number equals 0.52 in the current work. More details of the two-dimensional distribution of heat transfer and secondary flow in the rotating channel are presented in this paper.



## 1. Introduction

The increasing demand on the high efficiency of gas turbine engines requires the engine designers to develop effective cooling technologies because the gas temperature at the inlet of turbine is far beyond the working temperature even melting point of material. To resolve this conflict, lots of cooling techniques are applied to protect the turbine blade. Internal forced convection is one of the most classical and popular methods used to keep an appropriate temperature in the blade material. As one of the effective types for the internal forced convection, serpentine passage in the middle section of a turbine blade has been investigated, improved, and applied in the turbine blades more than thirty years. Over the past several decades, a vast amount of studies dealing with internal cooling of turbine blades have been reviewed by Han [1,2].

The flow and heat transfer of the coolant within the internal channel of the turbine blade is typical heat transfer in rotating channels. The history of studies on the heat transfer in rotating channels can stretch back to 1989. Wagner and Johnson [3,4] reported detailed measurements in a rotating channel. They concluded that rotating increases the heat transfer up to 3.5 times on the trailing surfaces and decreases to 40% on the leading surfaces compared to non-rotating results in the inlet passage. They also found that four parameters influence the heat transfer in a rotating passage: coolant-to-wall temperature ratio, Rossby number, Reynolds number and radius-to-passage hydraulic ratio. Then many researchers started to investigate the heat transfer in a rotating channel. Kukreja et al. [5] used the Naphthalene sublimation technology to investigate the heat transfer in a rotating two-pass square channel. They also found that the Coriolis induced by rotation increases the mass transfer on the trailing wall, and reduces the mass transfer on the leading wall. Such trend was also observed by other researchers [6–10]. However, all the researches focus on the averaged heat transfer distribution along the channel based on the copper plate technique. In a rotating internal cooling channel, rotation induced-second flow and centripetal buoyancy force makes the flow and heat transfer more complex. Therefore,

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**Nomenclature**

| | |
|---|---|
| A | heat transfer surface area ($m^2$) |
| *Buo* | rotational buoyancy parameter (see Eq. (10)) |
| $C_p$ | specific heat capacity |
| d. r. | density ratio |
| D | hydraulic diameter (mm) |
| h | heater transfer coefficient ($W/(m^2\ K)$) |
| HSV | Hue, Saturation, Value |
| ITO | Indium Tin Oxide |
| LS | leading side |
| TLC | Thermography Liquid Crystal |
| *Nu* | local Nusselt number (see Eq. (7)) |
| $\overline{Nu}$ | surface averaged Nusselt number |
| *Pr* | Prandtl number |
| q | heat flux rate ($W/m^2$) |
| r | rotation radius (m) |
| RGB | Pixel Red, Green and Blue values |
| *Re* | Reynolds number (see Eq. (8)) |
| *Ro* | rotation number (see Eq. (9)) |
| T | temperature (K) |
| $\Delta$ T | temperature between walls and bulk flow (K) |
| TS | trailing side |
| U | mean average velocity of coolant (m/s) |
| X | stream-wise direction |
| Z | span-wise direction |

*Greek symbols*

| | |
|---|---|
| $\mu$ | viscosity of the coolant (Pa$*$s) |
| $\lambda$ | thermal conductivity of the coolant ($W/(m^2K)$) |
| $\Omega$ | rotate speed (rpm) |
| $\rho$ | density of the coolant ($kg/m^3$) |
| $\delta$ | thickness of Plexiglas |

*Subscripts*

| | |
|---|---|
| b | bulk |
| e | environment |
| f | fluid |
| in | inlet of the heated channel for heated section |
| joule | joule power |
| loss | loss |
| net | net |
| plexi | Plexiglas |
| s | stationary |
| w | wall |
| 0 | fully-developed turbulent flow in non-rotating smooth round pipe |

more detailed information about two dimensional heat transfer data along the whole surface in the rotation channel is needed for turbine blade designers.

Bons and Kerrebrock [11,12] presented a detailed measurements by PIV and an infrared detector (IR) to show the flow filed and heat transfer distribution in a rotating square smooth channel with Reynolds number equaling 8100 and the max rotation number equaling 0.2. According to the velocity data, they found there is a strongly distorted stream-wise profile indicative of a buoyancy effect on the leading side, which is helpful for investigators to understand the phenomena of the heat transfer distribution in the rotating channel. However, they did not obtain the velocity data within the boundary layer, which is very important to explain the phenomenon of heat transfer. What' s more, due to the different resistance of the four heater on the four walls, the thermal boundary conditions of their investigation are not symmetrical, resulting in the max difference equaling 20% between the four walls. As we all know, heat transfer and flow are coupled to each other; therefore, thermal boundary condition is very important for the investigation in a rotating channel, which has been verified by Han's investigation [13]. In 1992, Han investigated the influence of the different thermal boundary conditions in a rotating channel. Three cases of thermal boundary conditions were studied: (A) four walls with uniform temperature, (B) four walls with uniform heat flux and (C) leading and trailing walls being hot and two side being cold. The investigation found that the relationship of the value of $Nu/Nu0$ on the leading side for three cases is case B > case C > case A, and on the trailing side is case A > case C > case B. However, the difference of $Nu/Nu0$ among the three cases on the trailing side decreases compared with that on the leading side. Parsons et al. [14] continue their experiment on wall heating condition in a rotating two-pass square channel with rib turbulators in 1995. In their investigation, two thermal boundary conditions cased were studied: (A) all four walls with the dame temperature, and (B) all four walls with same heat flux. They found that, Nusselt number ratio for Case B vary from 20% below to 50% above Case A with untwisted orientation 60°and 90° ribs. But for the twisted orientation 60° and 90° ribs on both leading and trailing side, Nusselt number ratio of Case B varys from 25% below to 150% above that for Case A.

Liou et al. [15] firstly investigated the heat transfer in a rotating internal channel by TLCs with Reynolds number equaling 10,000 and the rotation number ranging from 0 to 0.2. The paper obtained the detailed local Nusselt number distributions on the leading and trailing walls. From the detailed data, the work found that there is a critical rotation number Roc. Below Roc, rotation has no prominent effect on the regional averaged Nusselt number ratios of the region after a 180° sharp turn in a rotating two-pass cooling passage, which is very useful for practical tests of computational models. However, in their experiments, the air is hot and the channel wall is cool, which causes the opposite direction of the buoyancy force between their works with the actual conditions in the channel. Therefore, the heat transfer data under the effect of buoyancy force is not reliable. Then, they [16] used IR to measure the wall temperature in a radially rotating twin-pass smooth parallelogram channel with the Reynolds number and rotation number ranging from 5000 to 20,000 and 0 to 0.3, respectively. In the experiment, they used two stainless steel heating foils to heat the leading and trailing side, and the other two sides of the channel are cold. They generated a heat transfer correlation determining the area-averaged Nusselt numbers over the inlet and outlet legs and over the turning region.

Recently, Mayo et al. [17,18] from VKI, investigated the details of the characterization of the flow and heat transfer distribution in a rotating ribbed channel with TRPIV and TLC. The wall with ribs is leading side when the channel rotates clockwise and trailing side when it rotates counter-clockwise. Due to the visualizing test technology, they obtained the details of the distribution of flow and heat transfer between the 6th and 7th ribs, which was really helpful for other researchers to understand the effect of rotation in the rotating channel. However, in their experiment, limited by the visualizing test technology, only the ribbed wall was heated, the other three walls were cold, which causes the distribution of heat transfer on the edge of the measured wall in this case is not