Heat transfer investigation in a rotating U-turn smooth channel with irregular cross-section

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A R T I C L E   I N F O

Article history:
Received 24 July 2015
Received in revised form 29 December 2015
Accepted 30 December 2015
Available online 4 February 2016

Keywords:
Heat transfer
Rotating channel
Engine-similar
Smooth walls
Irregular

A B S T R A C T

In the current study, heat transfer performance in a rotating U-turn smooth channel with engine-similar irregular cross-section was investigated experimentally. The channel consists of an inlet pass with irregular cross section \( (d_1 = 24.5 \text{ mm}) \), a sharp 180-degree turn, and an outlet pass with nearly rectangular cross section \( (d_2 = 19.6 \text{ mm}) \). In the experiments, the Reynolds number ranges from 25,000 to 50,000 for the inlet pass and 31,000 to 62,000 for the outlet pass, respectively. In addition, the highest rotation number is 0.72 for the inlet pass and 0.37 for the outlet pass. The mean density ratio maintains around 0.138 in all working conditions. The results indicate that the cross-section affects the heat transfer in both stationary and rotating conditions; but the strengths of effect are different. In the stationary case, the \( \frac{Nu}{Nu_0} \) ratio on the trailing side is up to 1.5 times of that on the leading side in the inlet pass. In the rotating case, the \( \frac{Nu}{Nu_0} \) ratio is up to 4.3 on the trailing side in the inlet pass, but not changing remarkable on the leading side and all sides in the outlet pass. Due to the shrinking hydraulic diameter of the outlet pass, the \( \frac{Nu}{Nu_0} \) ratio in the outlet pass is approximately 1.5 times of that in the inlet pass. In the outlet pass, the \( \frac{Nu}{Nu_0} \) ratio shows different tendency with \( \text{Re} \) at different Reynolds numbers, which means that the \( \frac{Nu}{Nu_0}-\text{Ro} \) graph cannot eliminate the influence of \( \text{Re} \) for the current work.

1. Introduction

The increasing demands about the higher efficiencies of gas turbine engines need the engine designers to develop effective cooling technologies because that the turbine inlet temperatures of the gas are far beyond the working temperature even melting point of material. The internal cooling technique, which is classical and effective cooling technique, has been investigated, improved, and applied in the aero engine 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 [1–3].

In a rotating internal cooling channel, the flow and heat transfer characteristics are controlled by the Coriolis, buoyancy forces, and centrifugal forces. At the same time, these forces are influenced by some geometrical factors of the channel and working conditions, such as channel orientation, shapes of the cross-section, flow parameters (such as \( \text{Re} \) and \( \text{Ro} \)), rotation orientation (\( \beta \)), and channel aspect ratio (AR) in regular cross-section channels.

1.1. Effect of the rotation

In a rotating internal cooling channel, Coriolis, buoyancy forces and centrifugal forces are the three main forces that significantly alter the flow and heat transfer inside the channel. Wagner et al. [4,5] reported detail 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. Kukreja et al. [6] used the Naphthalene sublimation technology to investigate a rotating two-pass square channel. They also found that the rotation-induced Coriolis increases the mass transfer on the trailing wall reduces the mass transfer on the leading wall. Such trend was also observed by other researchers [7–13].

1.2. Effect of the cross-section shape

Cross-section shape is a critical factor in the mechanism of the second flow formation and strength, which will affect the heat transfer in the rotating channel. Early researchers paid more
attention on tubes with a circular cross-section [7,8]. Later, researchers began to focus on square and rectangular smooth cross-section ducts [9–12]. Kuo and Hwang [9] and Soong et al. [11] conducted a series of tests on the rotating rectangular ducts with aspect ratio of (AR) = 0.5, 1, 2 and AR = 0.2, 0.5, 1, 2, 5, respectively. They found that the heat transfer performances are different for the cross-section ducts with different aspect ratio; they also concluded that the Coriolis-induced secondary flow provides a positive effect on the heat transfer enhancement, while the buoyancy forces have the negative effect. Moreover, the channel with AR = 1 shows the best heat transfer enhancement for all the experimental cases. Murata and Mochizuki [10] numerically investigated the effect of cross-section aspect ratio (0.25–4) in a rotating smooth channel via Large Eddy Simulation. They draw the conclusion that the Coriolis-induced secondary flow and the fluctuating components of Coriolis influence the turbulent flow indirectly and directly, respectively. In addition, they also found the same conclusion with Kuo and Hwang [9] and Soong et al. [11]. Zhou et al. [12] investigated a smooth two-pass coolant passage with AR = 4, and found that the heat transfer of the inlet-leading surface for the AR = 4 enhancing with Re at low Re (Re < 20,000) and degrading with Re at high Re (Re > 20,000).

Furthermore, Dutta et al. [13] experimentally and numerically studied the local heat transfer in a rotating two-pass triangular duct with smooth walls. Unlike the square channel, the difference between the trailing and leading sides is greater than that in the square duct. From the numerical results, they concluded that the triangular duct has one dominant vortex structure at the broad section and one smaller vortex at the narrow corner in the cross flow plan, which is different from the two symmetric counter rotating vortices and is the main reason for the different heat transfer performance between triangular and square channel.

1.3. Effect of the orientation

Channel orientation is another parameter that will influence the heat transfer in a rotating channel. Park and Lau [14] used the naphthalene sublimation technology to investigate the effect of channel orientation to the heat transfer in a smooth rotating channel. The results showed that the orientation of the rotating two-pass channel significantly affect the local mass transfer distribution on the first pass of the channel, due to the rotational Coriolis in the diagonally oriented channel induces secondary flow and shifts the high stream wise velocity diagonally. The same results were obtained in the investigation of Dutta and Han [15]. Then, Huh et al. [16] performed a study on a rectangular channel with smooth and ribbed wall. They draw the conclusion that in the smooth case, the channel orientation is important and beneficial to the enhancement of heat transfer on the leading surface in the first pass. Li et al. [17] investigated the effect of channel orientation on the heat transfer in rotating smooth square channel at high rotating number. The results showed that the heat transfer in angled channels decrease at a relatively lower rotation number (Ro < 0.35) for both inlet and outlet passes. While at high rotation number (Ro > 0.35), heat transfer decrease on both leading and trailing walls in the first pass, and increase in the second pass.

As mentioned above, heat transfer studies in rotating channels have been investigated for decades of years. However, most of these investigations were performed at regular cross-section (such as circle, square, rectangular, et al.) and at the specified orientations. There are not many on the engine-similar geometries of the cross-section. In actual turbine blades, the cross-section is always irregular and the orientation is not always perpendicular to the rotating direction. Under this condition, the flow and heat transfer in the channel will be more complex and sophisticated. The geometrical shape and orientation will generate the different structures of Coriolis-induced secondary flows. In addition, heat transfer is the boundary effect of flow field, which means that the cross-section will affect the performance of heat transfer severely. These may generate some especial phenomena of flow and heat transfer. Therefore, it is interesting to find out the characteristics in an engine-similar irregular cross-section, with particular orientation, which can be widely applied for engine designers.
The objectives of current work can be summarized as follow:

1. Investigate the effect of cross-section shape on the heat transfer in the U-turn smooth channel for both stationary and rotating cases. In the current study, the rotation number and Reynolds number varies from 0 to 0.72 and 25,000 to 50,000 (referred to the hydraulic diameter of inlet pass, \( d_1 = 24.5 \text{ mm} \)), and 0 to 0.37 and 31,000 to 62,000 (referred to the hydraulic diameter of outlet pass, \( d_2 = 19.6 \text{ mm} \)), respectively. Which covers the work conditions of the actual aero-engine \( \text{Ro} \) ranges from 0.1 to 0.3 with the \( \text{Re} \) of 25,000 [18].

2. Find out the different phenomenon of heat transfer on leading and trailing sides for both inlet pass and outlet pass under stationary condition and rotating condition with current cross-section shape.

3. Develop the surface averaged heat transfer correlations to predict the heat transfer in the current engine-similar cross-section channel.

2. Experiment setup

2.1. Rotating facilities

A schematic of the rotating facility is given in Fig. 1. The facility is similar with the previous work which has been introduced in Deng et al. [17,19]. In the facility, the coolant air, provided by a compressor, passes through a mass flow meter, and then transports to the rotating facility via a two-pass rotary union (air in port). After that, the air traverses along the horizontal hollow shaft, turns 90 degree into the two-pass test section. Before expelled to the room via the 2-pass rotary union (air out port), the air passes through a back pressure control valve which is used to ensure the safety of whole test section at high pressure. The test section is placed at the top of the rotating arm with the maximum rotating diameter of 1.3 m, and the balance weight is on the opposite side to ensure the balance of the rotating system. Thermocouples are used to measure the temperature of the test section. The analog signals of the thermocouples are converted into digital ones before transmitted to the non-rotating facilities by slip rings. All the cold ends of thermocouples are placed in a rotating cavity, which is isolated from hot parts such as bearings and shafts. Therefore, the temperatures in this cavity can be relatively constant and even. The absolute temperature in this cavity is measured by several DS18B20 chips.

2.2. Test section

The dimension of current two-pass channel is shown in Fig. 2 and the experimental variables are shown in Table 1. The channel consists of an irregular inlet duct, a sharp U-turn bend, and an almost rectangular outlet duct. The hydraulic diameter of the inlet pass is \( d_1 = 24.5 \text{ mm} \), and that of the outlet pass is \( d_2 = 19.6 \text{ mm} \), respectively. The entire channel (\( L = 13d_1 \)) includes a heated section and an unheated section. The heated section contains a straight channel with \( L = 8.0d_1 \) and a sharp turn with \( L = 0.5d_1 \). The regional averaged heat transfer coefficients of the heated section are measured. In addition, the unheated section with 4.5\(d_1 \) long unheated straight channel before/after the heated section is just used to provide better inlet/outlet boundary conditions. The inlet and outlet are located at 9.4\(d_1 \) long from the rotational axis.

In order to identify the location of the channel, the heated walls are divided into 19 test positions along the stream wise. 43 copper plates are used to heat the channel walls of inlet pass, outlet pass, and sharp turn. In the inlet pass, there are 8 copper plates placed on the trailing side, leading side, and outer side, respectively. Same conditions are added in outlet pass. Moreover, in the bend turn, only 3 copper plates are placed on the outside. On other sides (inlet pass inner side; outlet pass inner and outer sides; leading, trailing and inner sides of turn), due to the narrow space, there are not any copper plates to heat the wall. Inversely, thermal insulation material Teflon is used to keep the adiabatic condition of the side wall. All the copper heaters, connected in series, have the same resistances, so that they can provide the uniform heat flux. Due to the high conductivity copper plates (398 W/(m K)), the regional average temperatures can be measured by 43 T-type thermocouples placed in the blind hole on the back of each copper plate. There are also 2 T-type thermocouples for inlet and outlet air temperature, and 1 for the environment. More details about the heat plates were introduced in the previous works [19].

3. Data reduction

Copper plate technique [4] is used to obtain temperatures in each heated plate segment. The regionally averaged heat transfer
coefficient \((h)\) is calculated by measuring the net heat transferred to the flow from the heated copper plate segment \((Q_{net})\), the regionally averaged wall temperature \((T_w)\), the local bulk mean temperature \((T_b)\), and the heat transfer surface area \((A)\)

\[
h_i = \frac{Q_{net,i}}{A_i (T_{w,i} - T_{b,i})}
\]

\[
Q_{net,i} = V_i \cdot I_i - Q_{loss,i} = V_i \cdot I_i - x_i (T_{w,i} - T_e)
\]

The net heat flux can be obtained by measuring the total energy generated by each film heater \((V \cdot I)\) and the energy dissipated into the environment \((Q_{loss})\). Before and after the experiments, heat loss calibration tests have been performed to determine the external heat losses for each given copper plate segment. The channel is heated with fulfilled insulation and operated under the rotational speed of 0–900 rpm. Thus, the heat loss coefficient \((x)\) is the function of rotational speed. In current study, the percentage of heat losses vary from 3.8\% \((Re_1 = 50,000, n = 0 \text{ rpm})\) to 10.1\% \((Re_1 = 25,000, n = 900 \text{ rpm})\).

The local bulk mean temperature \((T_b)\) is calculated by local energy balance method as follow:

\[
T_{b,i} = \frac{T_{in,i} + T_{out,i}}{2} + \frac{1}{2} \frac{\sum_{\text{span-wise}} Q_{net,i}}{m C_p}
\]

\[
Nu_i = \frac{h_d}{x}
\]

The subscript \(i\) represents the local parameter. \(T_{in,i}\) is the inlet temperature at local position \(i\). \(T_{out,i}\) is the average outlet temperature at local position \(i\), which can be calculated by the net heat transferred to the copper plate \((Q_{net,i})\) at local position \(i\), the mass flow rate \((m)\) of coolant and the heat capacity \((C_p)\) of coolant. The local Nusselt number \((Nu_i)\) can be calculated by local heater transfer coefficient \((h_i)\), hydraulic diameter \((d)\) and thermal conductivity of the coolant \((x)\).

Due to the different hydraulic diameters between inlet pass and outlet pass, subscript 1 and subscript 2 are used to represent the characteristic value of inlet pass and outlet pass, respectively. All the experimental data are recast in term of dimensionless form, introducing Reynolds number \((Re)\) and rotation number \((Ro)\), and overall buoyancy parameter \((Buo)\) as followed:

\[
Re = \frac{\rho Ud}{\mu} \quad \text{and} \quad Ro = \frac{\Omega d}{\nu}
\]

\[
Buo = \frac{\rho g (\gamma - 1) d^3}{\mu^2}
\]
\[ R_0 = \frac{\alpha d}{U} \]  
\[ B_{\text{tu}} = \frac{T_w - T_{\infty}}{T_w - T_{\text{ave}}} \]  

where \( \rho \) is the density of the coolant, \( U \) is the mean average velocity of coolant, \( d \) is the hydraulic diameter, \( \mu \) is the viscosity of coolant, \( A \) is the heat transfer surface area, \( \Omega \) is the rotational speed, \( T_w \) is the average temperature of the wall, \( T_{\infty} \) is the inlet temperature of coolant, and \( T_{\text{ave}} \) is the average rotating radius of rotation channel.

The Dittus–Boelter/McAdams correlation is used to calculate the Nusselt number (\( Nu_0 \)) for fully developed turbulent flow through a smooth stationary pipe. Thus, the heat transfer enhancement (\( Nu/Nu_0 \) ratio) is given as (Prandtl number (\( Pr \)) = 0.71):

\[ \frac{Nu}{Nu_0} = 0.023 Re^{0.8} Pr^{0.4} \]  

According to error transfer theory, the data uncertainty is estimated by the method offered by Kline and McClintock [20]. The uncertainty of \( Re \), \( Ro \) and \( Nu \) comes from the uncertainty of some measurement quantities, such as temperature (\( T \)), rotation speed (\( \Omega \)), mass flow rate (\( m \)) and heat flux (\( Q \)). The measurement accuracy of the thermocouples is ±0.5 °C, optical tachometer is ±1 rpm, mass flow meter is ±1%, and digital multimeter is ±0.0001 A. It is noteworthy that the heat loss does not mean the uncertainty of net heat flux, because the heat loss can be deducted from the total energy. However, the uncertainty of the net heat flux comes from the uncertainty of heat loss. In the current work, we estimating that the maximum uncertainty of heat loss is 40% conservatively. The maximum uncertainty of \( Re \) and \( Ro \) are 2.6% and 3.4%, respectively. The calculated uncertainty of \( Nu \) varies from 3.6% to 10.1% of the presented data.

4. Results and discussion

4.1. Data validation

As mentioned above, most of the previous investigations concentrate on the rectangular cross-section channel, the research on the engine-similar duct with irregular cross section is lacking. Therefore, in order to validate the experiments, the current study was compared firstly with the square or rectangular cross-section channel. Fig. 3 shows the stationary results of the averaged heat transfer for inlet pass from Ref. [21] (smooth parallelogram channel, \( d = 32.17 \text{ mm} \)), Ref. [22] (smooth 2:1 aspect ratio channel, \( d = 16.93 \text{ mm} \)), Ref. [19] (smooth square channel, \( d = 24 \text{ mm} \)) and current study (engine-similar channel, \( d = 24.5 \text{ mm} \)). All the previous work of Refs. [19,21,22] and the current work showed the same results that the trend of \( Nu/Nu_0 \) ratio decreases with the increase of Reynolds number. (\( Nu \) is the surface averaged Nusselt number on the whole leading or trailing side).

4.2. Station \( Nu/Nu_0 \) ratio

Fig. 4 shows the stationary \( Nu/Nu_0 \) ratio distribution in the stream wise direction for both leading and trailing wall at \( Re_1 = 30,000, 40,000, 50,000 \) and \( Re_2 = 37,000, 50,000, 62,000 \), respectively. The trend of \( Nu/Nu_0 \) ratio along the stream wise is typical. Due to the effect of inlet condition, the \( Nu/Nu_0 \) ratio decreases monotonously in the stream wise direction of the inlet pass, then elevates prominently nearing the 180-degree turn, and finally drops along the outlet passage. However, it is interesting to find out that in the inlet pass, the \( Nu/Nu_0 \) ratio of trailing side is up to 1.5 times of that on the leading side; but the difference between the trailing side and the leading side in the outlet passes decreases monotonously along the flow direction. This phenomenon is different from other investigations with square or rectangular cross-section [4,19,23,24]. However, the work of Dutta et al. [13] also found a slight difference of heat transfer between the leading and trailing side in a triangular duct without rotation at low Reynolds number. In the current study, due to the corner vortex generated by the small angle between trailing side and inter side of the inlet pass the flow around the trailing side is more disordered, which is the main reason that the \( Nu/Nu_0 \) ratio of the trailing side is higher than leading side for the inlet pass. However, in the outlet pass, under the effect of the 180-degree turn, the flow near the trailing wall and leading wall is well mixed, so the difference of heat transfer between the leading side and trailing side decreases.

4.3. The effect of rotation

In a rotating channel, the rotation-induced Coriolis is always an important force influencing the distribution of the heat transfer. The Coriolis pushes the main flow towards the trailing side of the inlet pass and leading side of the outlet pass, which results in a pair of cross-stream vortices that impinging directly onto trailing surface in the inlet pass and leading surface in the outlet pass, respectively. Then, the heat transfer on these surfaces is expected to be higher than other surfaces. However, in current work, as shown in Fig. 5, due to the irregular shapes of cross-section and the unique direction of the rotating axis, the Coriolis is not perpendicular to the leading surface or the trailing surface, which weakens the effect of the Coriolis.

Fig. 6 shows the distribution of \( Nu/Nu_0 \) ratio on leading and trailing side in stream wise under rotation conditions at \( Re_1 = 30,000, 40,000, 50,000 \) and \( Re_2 = 37,000, 50,000, 62,000 \), respectively. For the leading side, the \( Nu/Nu_0 \) ratio at entrance region of the inlet pass is higher than the position far from the entrance. Then the heat transfer enhances at the upstream of the 180-degree turn, which due to the turn bend increase the turbulence of the main stream. After the 180-degree turn, under the effect of the turn bend, the \( Nu/Nu_0 \) ratio is up to 3.0 at the downstream of the turn. The \( Nu/Nu_0 \) ratio diminishes along the stream wise. Due to the effect of the turn bend and the shrinking hydraulic
diameter of the outlet pass, the $\frac{Nu}{Nu_0}$ ratio is approximately 1.5 times of that in the inlet pass with the same Reynolds number and rotate speed.

For the trailing side, the trend of heat transfer is almost same along the stream wise. However, due to the Coriolis directing to the trailing side and increasing the heat transfer in the inlet pass, the $\frac{Nu}{Nu_0}$ ratio is almost same on the trailing side between the inlet pass and outlet pass.

Moreover, the Coriolis directs to the trailing side in the inlet pass and leading side in the outlet pass, which leads to the different heat transfer trend with rotate speeds. On the trailing surface of the inlet pass, the $\frac{Nu}{Nu_0}$ ratio increases with rotate speed. The $\frac{Nu}{Nu_0}$ ratio can be up to 4.3 at 900 rpm with the $Re_1$ about 30,000, which is almost 2 times of the stationary case. However, the heat transfer enhancement generated by rotating decreases with the increase of Reynolds numbers. It is interesting to observe that the rotating effect on the trailing surface is not obvious in the outlet pass. The weakened influence of rotation can be attributed to the nearly rectangular cross-section shape of the outlet pass, which decreases the effect of rotation. Moreover, with the unique rotation orientation, the Coriolis is not perpendicular to the trailing side, which also weakens the effect of Coriolis.

Fig. 7 shows the distribution of $\frac{Nu}{Nu_0}$ ratio for both leading and trailing side with different Reynolds numbers under rotating condition. In the inlet pass, the $\frac{Nu}{Nu_0}$ ratio on the trailing side is almost 2.5 times of that on the leading side at different rotate speeds. While in the outlet pass, the $\frac{Nu}{Nu_0}$ ratios on the leading and trailing side are at the same level, converging to approximately 2.5.

For all cases, with the increase of Reynolds numbers, the $\frac{Nu}{Nu_0}$ ratio decreases. Moreover, the trend is remarkable on the trailing side in the inlet pass, but not obvious on the trailing side in the outlet pass.

### 4.4. The effect of rotation number

#### 4.4.1. Inlet pass

The distribution for the heat transfer enhancement on the leading and trailing side under rotation numbers in the inlet pass is shown in Fig. 8. Due to the Coriolis pushing the main flow to the trailing side, the $\frac{Nu}{Nu_0}$ ratio is higher than the leading side ($Nu_0$ is regionally averaged Nusselt number at stationary). However, at position 8, the difference of $\frac{Nu}{Nu_0}$ ratio between leading and trailing side is very small. The reason for the phenomenon has been mentioned above in the turn bend, the decreased hydraulic diameter causes the mainstream accelerating, which increases the mixture of the stream and deduces the effect of rotation, then the difference between leading and trailing surface of heat transfer is weakened.

When the channel rotating, $\frac{Nu}{Nu_0}$ ratio on the trailing side monotonically increases with the rotation number, which due to the increase of the Coriolis-induced cross-stream vortices impinging directly onto the trailing side. The $\frac{Nu}{Nu_0}$ ratio can be enhanced more than 2.4 times when the rotation number reaches to 0.72. On the contrary, the heat transfer at the entrance is not
increasing so highlighted because of the effect of inlet boundary condition. On the leading side, heat transfer weakens with the increase of rotation number at the lower rotation number \( (Ro < 0.2) \), and then enhances at higher rotation number \( (Ro > 0.2) \). The phenomenon of the critical point of heat transfer in the leading side has been reported by many researchers \([5,9,12,23,25]\) in square or rectangular ducts. On the leading side, under the effect of the Coriolis, the main flow is shifted to the trailing side, which causes the inertia force weakened, and the heat transfer weakened. However, when the inertia force is less than buoyancy force (in the opposite direction of main flow), a regional separation is formed near the leading surface, after that, the Coriolis acts this local flow toward leading surface, which results in the re-enhancement of heat transfer on the leading side.

Fig. 6. Distribution of \( \frac{\text{Nu}}{\text{Nu}_0} \) ratio on leading and trailing side in stream wise under rotation conditions at different Reynolds numbers.

Fig. 7. \( \frac{\text{Nu}}{\text{Nu}_0} \) ratios for both leading and trailing side at all Reynolds numbers under rotation condition.
4.4.2. Outlet pass

Unlike the inlet pass, the Coriolis force in the outlet pass directs most main flow to the leading side. Therefore, the heat transfer on the leading surface is expected to be higher than trailing side. However, due to the influences of the turn bend, the $\frac{Nu}{Nu_s}$ ratio on leading surface is lower than the trailing surface. When the rotating number is higher, the advantage of leading side appears gradually (see Fig. 9).

However, it should be noted that, for all positions on the trailing surface, the $\frac{Nu}{Nu_s}$ ratio is not enhanced remarkably under the effect of rotation. What’s more, at high Reynolds number ($Ro > 0.2$), there is a slight drop of the heat transfer on the leading side. In order to present the interesting phenomenon more clearly, Fig. 10 shows the $\frac{Nu}{Nu_s}$ ratio of the trailing and leading surface in the outlet pass.

As shown in Fig. 10, on the leading side of outlet pass, with different Reynolds numbers, the heat transfer is not enhanced monotonically with the increase of the rotation number. At low Reynolds numbers ($Re_2 = 31,000,37,000,43,000$), the $\frac{Nu}{Nu_s}$ ratio enhances with rotation number, and then drops slightly, and finally, re-increases at a high rotation number. With higher Reynolds numbers ($Re_2 = 50,000,60,000$), the phenomenon of the decline is more significant, but the heat transfer is not enhanced due to the limit of the rotation number in the current study. Moreover, it is
interesting to find out that the trend of the heat transfer under rotation on the trailing surface is similar to that on the leading side. It can be concluded that for the current engine-similar rotation channel, the trend of $Nu/Nus$ ratio with rotation number is different at different Reynolds numbers, which means that the $Nu/Nu$ graph cannot eliminate the influence of $Re$ in the outlet pass. Comparing with previous work, this phenomenon is found for the first time in the public literatures. The new phenomenon can be attributed to the unique cross-section, rotating orientation and the decreasing hydraulic diameter of the outlet pass. However, in the current work, the phenomenon cannot be explained in detail, and more work should be done to give an accurate explanation.

As shown in Fig. 10, it is obvious that, with the increase of rotation number, the $Nu/Nu$ ratio on both leading (no more than 1.35) and trailing surface (no more than 1.1) in outlet pass, is not increasing notably as much as that in the inlet pass (up to 2.0 for both leading and trailing side, according to Fig. 8), which means that the rotation effect in the outlet pass is not significant comparing with that in the inlet pass.

4.5. The effect of buoyancy number

In a rotation channel, the centrifugal-buoyancy effects cannot be ignored especially when the rotation effects are evident. In the current study, Huh and Han's method [22] were followed to study the effect of buoyancy number. The buoyancy number ($Buo$) is always employed to quantify the strength of buoyancy force. According to the definition of $Buo$ which were defined as:

$$Buo = \frac{T_w - T_i}{T_{av} \cdot Ro \cdot \frac{d}{d'}}.$$  \hfill (7)

The buoyancy number is proportional to d.r. (defined by d.r. = $T_w - T_i/T_{av}$, where $T_w$ is the average temperature of the wall and $T_i$ is the temperature of inlet mass flow), $Ro$ and radius-to-diameter ratio ($ratio/d$). Fig. 11 shows the $Nu/Nu$ ratio on trailing and leading side with $Buo$ in the inlet and outlet pass at different Reynolds numbers.

According to Fig. 11, in the inlet pass, the $Nu/Nu$ ratio increases monotonically with $Buo$ on the trailing side. However, on the leading side, there is a slight drop for $Nu/Nu$ ratio, then increases with the increase of $Buo$. In the outlet pass, the similar phenomenon is also found in trailing and leading surface with that in Fig. 10. For the trailing side, the $Nu/Nu$ ratio is not enhanced remarkable with the increase of $Buo$, and on the leading side, the $Nu/Nu$ ratio shows different trend with $Buo$ at different Reynolds number. This is due to that, the d.r. in the current study is not varied prominently (at the level of 0.138), and the radius-to-diameter ratio is also a constant number for the discussion about the $Nu/Nu$ ratio. This implies that the extensive variation of buoyancy number attributes to the change of rotation number, and the similar trend of $Nu/Nu$ ratio can be found between Figs. 10 and 11.
4.6. Surface averaged heat transfer correlations

From Fig. 11, it can be seen that the \( \frac{N_u}{N_u_0} \) correlations on leading or trailing surface can be easily obtained. The power law expression is used to fit the data, which were used by the previous work of Huh et al. [22] to predicted the heat transfer in a 2:1 channel under rotating conditions. The coefficients and exponents for the \( \frac{N_u}{N_u_0} \) correlations are shown in Table 2.

In Fig. 12, correlation validations on trailing and leading surfaces in both inlet and outlet pass are given. It can be seen that for the inlet pass leading side, inlet pass trailing side, and outlet pass trailing side, all the points are in the ±5% error band, which is acceptable accuracy for predicting the heat transfer in a rotating channel. However, on the leading side of the outlet pass, the error band is ±10%. The bigger errors can be attributed on the leading side in the outlet pass. It can be found in Fig. 10 that the \( \frac{N_u}{N_u_0} \) ratio cannot eliminate the influence of \( Re \) in the outlet pass for the unique cross-section in current study.

5. Conclusion

An experimental study has been undertaken in a rotating engine-similar two-pass internal smooth cooling channel with irregular cross-section. Experiments were performed in a comprehensive range of Reynolds number (from 25,000 to 50,000 for the inlet pass, and 31,000 to 62,000 for the outlet pass) and rotation number (from 0 to 0.72 for the inlet pass, and 0 to 0.37 for the outlet pass). Due to the special cross-section shape and unique rotation orientation, some unusual phenomena were found. The following are the main conclusion reached.

(1) For the stationary cases, due to the irregular cross-section shape, the \( \frac{N_u}{N_u_0} \) ratio on the trailing side is up to 1.5 times of that on the leading side in the inlet pass, which is different from the irregular cross-section channel. However, in the outlet pass, the effect of the turn bend and decreased hydraulic diameter, the difference between trailing and leading decreases.

(2) Rotation induced–Coriolis enhances the \( \frac{N_u}{N_u_0} \) ratio up to 4.3 on the trailing side in the inlet pass, but not remarkable on the leading side and all sides in the outlet pass. The \( \frac{N_u}{N_u_0} \) ratio on the leading side in the outlet pass is approximately 1.5 times of that in the inlet pass with the same Reynolds number and rotate speed.

(3) The \( \frac{N_u}{N_u_0} \) ratio decreases with the increase of Reynolds number for all sides, which is remarkable for the trailing side.
in the inlet pass, but not obvious on the trailing side in the outlet pass. The $\frac{Nu}{Nu_0}$ ratio on the leading and trailing side converge to approximately 2.5 in the outlet pass.

(4) In the outlet pass, the $Nu/Nu_0$ ratio shows different tendency with $Ro$ at different Reynolds numbers, which means that the $Nu/Nu_0$–$Ro$ graph cannot eliminate the influence of $Re$ in the current work. This phenomenon indicates that the cross-section shape influences the trend of heat transfer in a rotation channel and is an important factor for blade designer to be considered.

References

[22] M. Huh, J. Lei, Y.-H. Liu, J.-C. Han, High rotation number effects on heat transfer in a rectangular (AR = 2:1) two-pass channel, J. Turbomach. 133 (2011) 021001.