Numerical simulations on flow and heat transfer in ribbed two-pass square channels under rotational effects

N. Kaewchoothong, K. Maliwan and C. Nuntadusit
Department of Mechanical Engineering, Faculty of Engineering, Prince of Songkla University, HatYai, Songkhla 90110, Thailand
E-mail: chayut@me.psu.ac.th

Abstract. The main objective of this research is to study the flow and heat transfer characteristics in a rotating two-pass square channel with ribbed walls. In this study, the channel length-to-hydraulic diameter ratio of the rotating two-pass square channel ($L/D_h$), the rib height-to-hydraulic diameter ratio ($e/D_h$), rib angle of attack ($\alpha$) and the rib pitch-to-height ($p/e$) ratio are fixed at 11.33, 0.13, 60° and 10, respectively. The test fluid is air having the flow rate in terms of constant Reynolds number ($Re$) of 10,000. The rotation numbers ($Ro$) are varied from 0.1 to 0.4. The details of the local heat transfer distribution and the flow field of the rotating two-pass square channel are numerically studied by using commercial software ANSYS Fluent (ver.15.0). The results show that the ribbed walls enhance the heat transfer rate significantly. Under rotation, the average Nu in the first pass with radial outward flow is increased while that in the second pass is decreased, and also found that maximum heat transfer rate is observed for rotation number of 0.4 which is higher about 10-20% when compared with the other rotation number cases.

1. Introduction

Gas turbine engine is one kind of widely used in the powerful machineries which were applied in the aircraft propulsion, land-based power generation system and industrial applications. In order to achieve the higher thermal efficiency and power output. One of ways is by increasing the turbine inlet temperature (TIT) which current gas turbine engines can operate at TIT up to about 1,700°C. Thus, it need effective cooling technologies to augment heat transfer rate with less amount of cooling air, such as rib-roughened channel cooling, impingement cooling, film cooling, pin-fin cooling and dimple or protrusion cooling as shown in Fig. 1. Rib turbulated cooling in internal channel is the most commonly adapted approach in gas turbine blade. Han and his co-workers [1-7] have performed a large amount of research in this field including the discussions of aspect ratio, channel shape, Reynolds number, rib configurations, etc.

Walker and Zausner [8] used commercial software ANSYS CFX to predict local heat transfer distribution in a ribbed two-pass square channel and indicated that SST $k$- $\omega$ was an optimal turbulence model for engineering applications. Bonhoff et al. [9] numerically studied a two-pass channel with 45° ribs using the commercial solver ANSYS FLUENT with Reynolds stress model (RSM) and standard $k$-$\varepsilon$ model. After that, Ekkad and Han [10] studied the detailed heat transfer distributions in two-pass square channels with the angles of 60°, 90° and 60° discrete rib turbulators. They reported that the heat transfer rate for rib-roughened walls was larger than smooth walls due to the secondary flows induced the effect of the bend and rib turbulators. In addition, Shen et al. [11]
numerically investigated smooth and ribbed two-pass square channel under rotation number between 0.4 and 0.6 at constant Reynolds number fixed at 12,500. They explained that enhanced heat transfer areas were moved to radial inward channel from the sidewalls of radial outward channel as divider-to-tip wall distance was increased. Rotational effect in turbine blade affects the flow and heat transfer characteristics considerably. In general, rotation effect induces Coriolis and centrifugal forces that produce cross-stream secondary flow in the rotating channel. The effect of rotation on the first-pass and second-pass channels is different due to the different outward and inward flow.

From the review of obtainable literature on internal flow and heat transfer rate under rotation conditions, it can be noted that the heat transfer characteristics should be investigated more detailed in order to design more effective gas turbine blade cooling. Additionally, few researchers numerically studied both flow characteristics and heat transfer rate overall surface in cooling channel. Thus, the present study focused on the detailed examination of the effect of rotation on the flow behavior and local heat transfer distributions inside the two-pass square channel where the ribs are attached on the two opposite walls. The angle of 60° inclined ribs in two-pass square channel is numerically performed using commercial software ANSYS Fluent (ver.15.0) which all cases are carried out under high rotational number of up to 0.4.

2. Numerical simulation and methods

2.1. Physical model

The geometry of a two-pass square channel with the angle of 60° inclined ribs on the opposite walls for this study is shown in Fig. 2. The coordinate system of x, y, z represents the spanwise, normal to heat transfer wall and streamwise directions, respectively, where the rotating direction of the two-pass square channel is clockwise of x-axis direction. The cross section \((W \times H)\) of a channel was square of 15 mm \(\times\) 15 mm with aspect ratio \(AR\) of 1. In this study, the ribs with square cross section were used to locate on two opposite walls in rotating direction. The rib height-to-hydraulic diameter ratio \((e/D_h)\), the length-to-diameter ratio \((L/D_h)\) of each of the first and second passages, and the rib pitch-to-height \((p/e)\) ratio were fixed at 0.133, 11.33 and 10, respectively. The air inlet of channel was at \(R/D_h=22.0\).
For boundary conditions in these numerical simulation, the flow rate was determined at the entrance in the radial outward channel, which the value of flow rate is defined by Reynolds number \((Re)\) of 10,000. Inlet temperature and turbulence intensity of a channel entrance was imposed at 298.15 K and 5.0\%, respectively. The flow exit of the channel was determined to be maintained at a constant pressure of 1.0 atm. All wall surfaces of channel were no-slip conditions with zero velocity component. The constant heat flux of about 300 W/m\(^2\) was defined on trailing and leading surfaces. The rotation numbers of the two-pass square channel were varied between 0.1 to 0.4 into consideration similar with the aircraft blade condition [12].

Figure 2. Geometry of a rotating two-pass square channel.

2.2. Computational details
In this numerical study, the 3D flow field and heat transfer on surface of the two-pass square channel are studied by CFD commercial software ANSYS Fluent (ver.15.0). The computational model was operated by solving Reynolds averaged continuity and Navier-Stokes equations. The Reynolds stress model (RSM) was applied in solving turbulent flow [13-15]. For numerical calculations, the SIMPLE algorithm was used for the pressure-velocity coupling in order to successful computations of governing equations. Second order upwind scheme was designated for special discretization of all equations. The solutions were considered to be convergent when the residual values for continuity, velocities and energy were less than 1x10\(^{-6}\), 1x10\(^{-4}\) and 1x10\(^{-7}\), respectively [16-17].

Figure 3. Grid used for computational domain.
Computational domain was generated by structured hexahedral element as shown in Fig. 3. The more intensive grids were used near the walls, bend and ribbed surfaces of rotating channel. In all domains, the number of grid more than 3.0 million could be considered of the two-pass square channel, which the grid spacing of the first layer normal to the wall corresponding to $y^+$ values was less than 1.0.

### 2.3. Data reduction

There are three parameters of interest included Reynolds number ($Re$), Nusselt number ($Nu$) and Rotation number ($R_o$). The $Re$ is determined as

$$Re = \frac{V_i D_h}{\nu}$$  \hspace{1cm} (1)

where $V_i$ is the inlet velocity, $D_h$ is the hydraulic diameter of channel, which it can be calculated from $D_h = 2WH/(W+H)$ when $W$ and $H$ were width and height of channel ($D_h=H$), and $\nu$ is the kinematic viscosity of air. The local $Nu$ is calculated from Eq. (1).

$$Nu = \frac{hD_h}{k}$$  \hspace{1cm} (2)

where $h$ is the convective heat transfer coefficient of air and $k$ is the thermal conductivity.

The Nusselt number ratio ($Nu/Nuo$) is calculated using the correlation of Dittus and Boelter [18] for fully developed flow, a smooth wall with rotation to normalized Nusselt number.

$$Nu/Nuo = \frac{Nu}{0.023Re^{0.8}Pr^{0.4}}$$  \hspace{1cm} (3)

where $Pr$ is the Prandtl number of air ($Pr=0.71$).

The average Nusselt number ratio ($\overline{Nu}/Nu_o$) is considered by averaging local Nusselt number values (from Eq. (1)) over the heat transfer surface.

$$\overline{Nu}/Nu_o = \frac{\sum_{i=1}^{n} Nu_i}{n}$$  \hspace{1cm} (4)

where $Nu_i$ is the local Nusselt number at a measured point, $n$ is the total number of measured point over the heat transfer surface.

The $R_o$ is determined from Eq. (5).

$$R_o = \frac{\omega D_h}{V_i}$$  \hspace{1cm} (5)

where $\omega$ is the rotating speed.

### 3. Results and discussion

#### 3.1. Validation of CFD model

The computational model for a flow and heat transfer enhancement of the two-pass square channel is predicted using each turbulence model to validate with experimental results under stationary channel of A.K. Sleiti [17] as shown in Fig. 4. Comparison of calculation heat transfer on the first-pass leading
surface between the 5th and 6th ribs is considered at \( Re=10,000 \) under stationary case. The results show that the predictions using RSM was the most accurate of about 5%. RTD \( k-\omega \) and SST \( k-\omega \) predicted the trend exactly but over predicted \( Nu/Nu_o \) near the rib top by 15% and 25%, respectively. Whereas, the worst are predictions both the standard (STD) \( k-\varepsilon \) and RNG \( k-\varepsilon \) of scalable wall function option in this numerical study. Thus, the RSM model is the best choice for this flow model.

![Graph showing Nu/Nu comparison for different models](image)

**Figure 4.** Numerical validation of heat transfer along the centerline of the floor in stationary channel.

### 3.2. Flow structures

Figure 5 shows the comparisons of 3D flow-field velocity contours and the velocity vector on each plane of channel under rotation number \( (R_o) \) of 0.0 (Fig. 5(a)) and 0.2 (Fig. 5(b)). Plane (i), (ii), (iii) and (iv) are on the middle between ribs in the first and second channels, and plane (v) is selected in the center of the bend section. As shown in Fig. 5(a) for the stationary case \( (R_o=0.0) \), the results show that angle of 60° ribbed channel of each plane in the first-pass channel created simultaneously the large vortex near the inner and the outer sides, which it is induced by the effect of inclined ribs. On the bend region, similar flow behavior occurs large vortex near the inner wall and the outer wall due to the combined effect of the U-bend and ribs in the first pass channel can induce the small counter rotating flow near inner side. After the bend, the flow field in second-pass channel is similar with the flow before the bend in first-pass channel because the rib-induced counter rotating vortex pairs.

For rotating case, the complex secondary flow occurred near the trailing surface in the first-pass channel (see Fig. 5(b)) is created by the angle of inclined ribs and the effect of rotation which is affected to increase the heat transfer enhancements on the walls. On the bend section, it is observed that a counter rotating vortex pair creates near the top side of the bend due to the combined effect of the centrifugal-induced vortex and the Coriolis-induced complex secondary flow in the upstream. In addition, the secondary flow pattern in the second-pass channel is rather complicated due to the combined effect of the bend and the rib-roughened walls, which the flow structure in the bend region is influenced by the angle of 60° inclined ribs, curvature of the bend and the shape of the ribs upstream and downstream of the bend.
The Nusselt number distribution pattern for the ribbed two-pass square channel is shown in Fig. 6. The results show that trends of \( \frac{Nu}{Nu_0} \) distributions are significantly high near the inlet due to the flow entrance effect and decreases along the main flow direction until the bend of channel. In second pass, similar heat transfer distribution can be noted at the different sections cause by the deflection of main fluids by the bend and the rib-induced circulation, which it declines gradually.

For rotational case, it is observed that the heat transfer rate is significantly enhanced in the first-pass trailing surface and distribution patterns between the ribs are different from the stationary case due to the combined effects of the Coriolis and centrifugal forces which can create the main fluids shift and attach on the trailing surface. Similar heat transfer ratio distribution pattern in second-pass leading surface is gradually reduced. Additionally, the computational results are also concluded that the combined effects between the Coriolis and centrifugal forces which can create the main fluids shift and attach due to the combined effects of the Coriolis and centrifugal forces which can create the main fluids shift and attach on the trailing surface. Similar heat transfer ratio distribution pattern in second-pass leading surface is gradually reduced.

The average Nusselt number ratios in each region between the ribs on leading and trailing surfaces of the ribbed walls within the two-pass square channel are shown in Fig. 7. On the leading side, the overall \( \frac{Nu}{Nu_0} \) tends to increase in the first pass of the channel due to the effect of angle of inclined ribs. After that, heat transfer rate decreases slightly in the second pass channel after the bend due to the effect of the circulation which can induces a pair of counter rotating flows in the cross section of the channel. These circulation flows cause the flow to attach on the first-pass leading surface stronger than the second-pass leading surface. On the trailing side, the results show that the average heat transfer...
ratios are significantly increased in the first pass and gradually decrease in second pass after the bend of two-pass square channel. The results also show that increasing of the rotation number gives the highest heat transfer rate on the first-pass and second-pass surfaces when compared with the stationary case. This is because of the stronger rotation-induced secondary flow vortices and the combined effects between the Coriolis and centrifugal forces (see Fig. 5).

Figure 6. Detailed heat transfer distributions on leading and trailing surfaces of the two-pass square channel.

Figure 7. Regional average Nusselt number ratio between ribs for various rotation number. (Left: on leading side, Right: on trailing side)

4. Conclusions
In the present study, 3D flow field and heat transfer distributions were numerically performed in order to investigate the effect of the rotation number on flow and heat transfer characteristics in the two-pass square channel. The main results of this research can be summarized as follows.

1. Effect of the rotation creates heat transfer enhancement differences between the leading and trailing walls for ribbed channels.

2. The maximum value of average Nusselt number ratio is remarked for rotation number of 0.4 which is higher than about 10-20% when compared with the other rotation number cases.
3. Under the effect of rotation, the Coriolis force will increase the heat transfer rate in the first-pass channel for radial outward flow while decrease the heat transfer rate in the second pass as radial inward flow.

Acknowledgment
This research was financially supported by Faculty of Engineering, Prince of Songkla University, Thailand.

References
[1] Han J C, Glicksman L R and Rohsenow W M 1978 An investigation of heat transfer and friction for rib-roughened surfaces International Journal of Heat and Mass Transfer 21(8) 1143-1156
[2] Han J C 1984 Heat transfer and friction in channels with two opposite rib roughened walls Journal of Heat Transfer 106(4) 774-781
[3] Han J C 1988 Heat transfer and friction characteristics in rectangular channels with rib turbulators Journal of Heat Transfer 110(2) 321-328
[4] Han J C and Park J S 1988 Developing heat transfer in rectangular channels with rib turbulators International Journal of Heat and Mass Transfer 31(1) 183-195
[5] Park J S, Han J C, Huang Y, Ou S and Boyle R J 1992 Heat transfer performance comparisons of five different rectangular channels with parallel angled ribs International Journal of Heat and Mass Transfer 35(11) 2891-2903
[6] Azad G S, Uddin M J, Han J C, Moon H K and Glezer B 2002 Heat transfer in a two-pass rectangular rotating channel with 45-deg angle rib turbulators Journal of Turbomachinery 124(2) 251-259
[7] Huh M, Liu Y H, and Han J C 2009 Effect of rib height on heat transfer in a two pass rectangular channel (AR = 1:4) with a sharp entrance at high rotation numbers International Journal of Heat and Mass Transfer 52(19-20) 4635-4649
[8] Walker D and Zausner J 2007 RANS evaluations of internal cooling passage geometries: ribbed passage and a 180 degree bend ASME Paper GT2007-278
[9] Bonhoff B, Tomm U and Johnson B V 1996 Heat transfer predictions for U-shaped coolant channels with skewed ribs and with smooth walls ASME Paper 96-TA-7
[10] Srinath V Ekkad and Han J C 1996 Detailed heat transfer distributions in two-pass square channels with rib turbulators International Journal of Heat and Mass Transfer 40(11) 2525-2537
[11] Shen Z, Xie Y, Zhang D and Xie G 2014 Numerical Calculations on Flow and Heat Transfer in Smooth and Ribbed Two-Pass Square Channels under Rotational Effects Mathematical Problems in Engineering ID981376-1-7
[12] Shen Z, Xie Y and Zhang D 2015 Numerical predictions on fluid flow and heat transfer in U-shaped channel with the combination of ribs, dimples and protrusions under rotational effects, International Journal of Heat and Mass Transfer 80 494-512
[13] Gibson M M and Launder B E 1978 Ground effects on pressure fluctuations in the atmospheric boundary layer Journal of Fluid Mechanics 86 491-511
[14] Launder B E 1989 Second-moment closure: present and future International Journal of Heat and Fluid Flow 10 282-300
[15] Launder B E, Reece G J and Rodi W 1975 Progress in the development of a Reynolds-stress turbulence closure Journal of Fluid Mechanics 68 537-566
[16] Sewall E A and Tafti D K 2007 Large eddy simulation of flow and heat transfer in the developing flow region of a rotating gas turbine blade internal cooling duct with Coriolis and buoyancy forces Journal of Turbomachinery 130(1) 011005-011005-7
[17] Sleiti K and Kapat J S 2005 Fluid flow and heat transfer in rotating curved duct at high rotation and density ratios Journal of Turbomachinery 127(4) (2005) 659-667
[18] Dittus P W and Boelter L M K 1985 Heat transfer in automobile radiators of the tubular type *International Communications in Heat and Mass Transfer* 12 3-22

**Nomenclature**

| Symbol | Definition |
|--------|------------|
| AR     | Channel aspect ratio (-) |
| Dh     | Hydraulic diameter of channel (m) |
| e      | Rib height (m) |
| H      | Channel height (m) |
| h      | Heat Transfer coefficient (W/m² K) |
| k      | Thermal conductivity of air (W/m K) |
| L      | Length of the channel (m) |
| n      | Total number of pixel over the heated surface. (-) |
| Nu     | Nusselt number (-) |
| Nu̅    | Average Nusselt number (-) |
| Nu_l   | local Nusselt number at a pixel (-) |
| Nu_o   | Nusselt number of the smooth wall (-) |
| p      | Pitch of ribs (m) |
| Pr     | Prandtl number (-) |
| R      | Radius of rotation (m) |
| Re     | Reynolds number (-) |
| Ro     | Rotation number (-) |
| T      | Temperature of air (°C) |
| V_in   | Inlet velocity (m/s) |
| W      | Channel width (m) |

**Greek symbols**

| Symbol | Definition |
|--------|------------|
| ν       | Kinematic viscosity of air (m²/s) |
| ω       | Rotational speed (rad/s) |
| ρ       | Density of air (kg/m³) |