Experimental study on the heat transfer characteristics of high blockage ribs channel

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Abstract

The heat transfer and pressure loss characteristics on a square channel with two opposite surfaces roughened by high blockage ratio ribs are measured by systematic experiments. Reynolds number studied in the channel range from 1400 to 9000. The rib height (e) to the height of ribbed channel (H) ratios are 0.2 and 0.33, respectively. The rib spacing (S) to height ratio (S/e) ranges from 5 to 15. The rib orientations in the opposite surfaces are symmetric and staggered arrangement. The experimental results show that (1) the heat transfer coefficients are increased with the increase of Reynolds number, though at the cost of higher pressure losses; (2) when the rib spacing to height ratio is 10, it keeps a highest heat transfer coefficient in three kinds of rib spacing to height ratio 5, 10 and 15; (3) the heat transfer coefficient of symmetric arrangement ribs is higher than the staggered arrangement ribs, but the pressure losses of symmetric arrangement ribs is larger than the staggered arrangement ribs; (4) compared with one-side ribbed channel, the heat transfer coefficient of two-side ribbed channel is distinct higher than that of one-side ribbed channel.

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Keywords: High blockage rib Gas turbine cooling Heat transfer Turbine blade

1. Introduction

Advanced gas turbine engines operate at high temperature to improve thermal efficiency and power output. As the turbine inlet temperature increases, the heat transferred to the turbine blades also increases. The level and variation in the temperature within the blade material must be limited to achieve reasonable durability goals. Because the operating temperature of modern gas turbines is far above the permissible metal temperature, there is a need to cool the blade for safe operation. There are two methods used for turbine blades to protect the blade material from exceeding the maximum allowable temperature, one is external cooling, such as film cooling, and another is internal cooling, such as impingement cooling, rib turbulated cooling, and pin-fin cooling. The internal cooling is achieved by passing the coolant through several enhanced serpentine passage inside the blade and extracting the heat from the outside of the blades. A common method of increasing the cooling capacity of the internal cooling circuit is the addition of rib turbulators to the internal coolant channel walls. The addition of the rib turbulators increases the overall internal convective heat transfer coefficient, causing a corresponding drop in the component metal temperature.

Rib turbulators are often used in the midsection ducts inside the blades to increase the convective heat transfer. The ribs lead to flow separation, reattachment, and strong secondary flow in the mainstream, and therefore it is very important to predict the heat transfer and friction characteristics in ribbed ducts accurately.

Over the last few decades, many investigators [1–6] have studied the flow and heat transfer characteristics in ribbed channels. Han and Wagner [7–11] studied the effect of Reynolds number on the centerline heat transfer coefficient of a square channel (W/H = 1) and two rectangular channels (W/H = 2.4) for two rib spacing (P/e = 10,20). The heat transfer distribution was presented by a Nusselt number ratio with several Reynolds numbers, and they showed similar trends except that the Nusselt number ratios decreased slightly with increasing Reynolds numbers. Bailey and Han [12,13] figured out that the best rib pitch to height ratio is between 7 and 15. Han et al. [14] pointed out that the channel with angled ribs performed better than orthogonal ribs due to the stronger secondary flow. Han et al. [15,16] showed that the orthogonal ribs in the narrow channel not only induce a high heat transfer coefficient but also cause a high pressure drop. The effect of the channel aspect ratio has been studied by Han and Park [17,18]. Though the ribbed side heat transfer augmentation is of the same order in all cases, the friction factor is much higher for channels

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http://dx.doi.org/10.1016/j.expthermflusci.2017.01.016 0894-1777/© 2017 Elsevier Inc. All rights reserved.
with wider aspect ratios. Studies by Taslim et al. [19] have focused on the cooling passages embedded in the leading edge of the blade. Zhang et al. [20] experimentally studies on a ribbed triangular channel. Taslim [21] stressed the contribution of the on-rib heat transfer coefficient to the total heat transfer. Maurer et al. [22–24] investigated the heat transfer performance of various rib geometries including W-shaped ribs for the combustor liner. The dimensionless rib height $e/D_h$ is 0.02 and pitch-to-height ratios $P/e$ are 5 and 10. Heat transfer coefficients were measured and flow fields were numerically simulated for Reynolds numbers up to 500,000. Increase in Reynolds number attenuates the heat transfer peak of the ribbed walls, which is more pronounced for the two-sided ribbed channel than for the one-sided ribbed channel.

Dees et al. [25] experimentally studied the heat transfer characteristics for 90 degree rib turbulator. They found that the internal rib turbulator can increase the overall effectiveness on the vane external surface by up to 50% relative to the non-ribbed model. Coletti [26] experimentally investigated the conjugate heat transfer in a rib-roughened trailing edge channel with crossing jets. The cooling scheme is characterized by a trapezoidal cross-section, one rib-roughened wall, and slots along two opposite walls. The Reynolds number is set at 67,500 for all the experiments. The measurements are performed using three different ribbed walls, with thermal conductivities ranging from 1 W/(mK) to 18 W/(mK). They found that the levels of the Nusselt number obtained in the purely convective regime (uniform heat flux at the wall-fluid interface) are off by up to 30% locally and 25% globally with respect to the conjugate results. Xie et al. [27] studied the flow and heat transfer characteristics in rectangular rib-roughened passages. They found that the side-wall heat transfer coefficients of the passage with ribs on opposite walls are about 20–43% higher than that of a smooth passage: the length of the ribs affects the heat transfer and friction characteristics. The level of augmentation of heat transfer becomes higher as the Reynolds number increases.

A flow visualization of the secondary flows is presented in Ref. [28], and a computational picture of secondary flows is painted in Ref. [29]. Su et al. [30] performed computations on a rotating channel with inclined ribs and presented predictions of secondary flows in the first channel. Liu et al. [31] studied the flow and heat transfer characteristics in a two-pass 90 degree ribbed parallelogram channel with infrared thermography and particle image velocimetry. It is found that the flow dynamic mechanisms responsible for the rib top and mid-rib heat transfer enhancement are different for the inlet and outlet passes. The pressures of a kewed high $N_{th}$ streak between the last inlet-leg rib and the bend as well as two high $N_{th}$ zones inside the bend are the new found features lacking in the corresponding two-pass 90-deg ribbed square channel. In addition, simple correlations of $N_{th}$ and $f_p$ with $Re$ are acquired. Thermal performance factors are about 66% and 28% higher than the previous reported smooth-walled counterpart at $Re = 5000$ and 20,000, respectively. Kim et al. [32] studied the effects of inlet velocity profile on flow and heat transfer in the entrance region of a ribbed channel. They found that in the entrance region, the location and shape of the reattachment and the recirculation region were altered by the local velocity distribution caused by the different inlet velocity profiles. Therefore, the distribution and the strength of the vorticity in the channel were changed, and the local heat transfer coefficient and pressure drop in the channel were affected by the inlet velocity profile.

In addition, the artificial roughness in the form of ribs has been extensively studied and widely used in the high-efficiency and low resistance heat exchange equipment such as heat exchanger and solar air heater over the past few decades. Since the inclined rib turbulators appear to be the most common in practice today, the majority of recent research papers are mainly focused on inclined, V and W-shaped rib turbulators [33–37]; Zheng et al. [38] investigated the effects of P-type and V-type rib arrangements on the flow pattern and heat transfer in an internally ribbed heat exchanger tube. The results reveal that the average Nusselt number and friction factor in the V-type ribbed tubes were about 57–76% and 86–94% higher than those in the P-type ribbed tube, respectively. The performance evaluation criterion (PEC) based on the same pumping power in the V-type ribbed tube varied from 1.32 to 1.74, which were about 27–41% higher than that in the P-type ribbed tube. Tang and Zhu [39] investigated on the turbulent flow and heat transfer behavior in the rectangular channel with inclined broken ribs for three kinds of rib arrays. They claimed that the heat transfer of the inclined broken ribbed channel was improved about 160–230% compared with smooth duct because of the generation of co-rotating longitudinal vortices. Moon et al. [40] evaluated the heat transfer performance in a rectangular channel of sixteen types of rib shapes, they found that boot-shaped rib design showed the best heat transfer performance with a pressure drop similar to that of the square rib. Tanda [41] experimentally investigated the effect of rib spacing on heat transfer and friction in a rectangular channel with 450 angled rib turbulators on one/two walls. The convective fluid was air, and the Reynolds number varied from 9000 to 35,500. The ratio of rib height to hydraulic diameter ($e/D$) was 0.09, while four rib pitch-to-height ratios ($p/e$) were studied: 6.66, 10.0, 13.33, and 20.0. Superior heat transfer performance was found at the optimal rib pitch-to-height ratio of 13.33 for the one-ribbed wall channel and at $p/e = 6.66–10$ for the two-ribbed wall channel.

From above we can know that the flow and heat transfer characteristics of ribbed channels has been deeply studied by a lot of researchers in the last few decades. Almost all studies in open literatures focus on higher Reynolds number (9000 at least) and a lower blockage ratio (2–10%). However, for the smaller gas turbine, the turbine blades have higher blockage ribs and lower coolant Reynolds number at closer spacing of turbine blade. The objective of this study is to measure the heat transfer coefficient and friction factor for the ribbed channels with large blockage ratio ($0.2,0.33$) and rib pitch-to-height ratio ($S/e$) ranging from 5 to 15. The Reynolds numbers tested are from 1400 to 8000. The study results of this paper are the benefit supplement for the internal cooling design of turbine blades.

2. Experimental setup

Fig. 1 shows the comprehensive scheme of experimental setup which consists of compressor, buffer tank, mass flow controller and test section. The air from compressor having environment temperature is ducted into the test section. A ball valve to protect and control the flow rate is located upstream of the test section. The flow is measured using a mass flow controller. The temperature of the air flowing into the test section is monitored by a T-type thermocouple. The spent air from the test section is directly exhausted into the atmosphere. The accuracy of the mass flow controller and T-type thermocouple is 1.0% and ±0.1°C, respectively. The temperature of the test wall is measured by an infrared thermography system (TVS-2000MK) with accuracy and measured temperature range of ±0.1°C (0–100°C) for the infrared thermograph system. All the measurement data of temperature are connected with 8-channel HP34970A data collection system.

Fig. 2 shows the schematic of the test section. There are two kinds of test sections in this experiment. Both of the test sections are rectangular channels. The geometrical dimensions of the test sections are 180 mm (length) × 60 mm (width) × 15 mm (height) and 180 mm (length) × 60 mm (width) × 9 mm (height), respectively. The dimensions of the pre-channel attached upstream of the test sections are 600 mm (length) × 60 mm (width) × 15 mm.
and 800 mm (length) × 60 mm (width) × 9 mm (height), respectively. At the entrance of the test section, there are a T-type thermocouple and a static pressure probe to measure the temperature and static pressure of the stream. The heating foil heated by passing DC power to provide uniform heat flux is made of 0.02 mm thick stainless-steel foil with 180 mm × 60 mm (length and width) dimensions. The surface of heating foil is painted with black paint to assure an uniform emissivity of 0.96. The channels are roughened with square cross section ribs made of stainless steel. The ribs with 3 mm × 3 mm × 60 mm dimensions are glued on heating foil and infrared grass, respectively. The ribs on the opposite wall are laid parallel to each other. The ribs are placed with a given spatial periodicity. The distance of the first rib from the entrance of the test section is 300 mm. The temperature of the heating foil is measured by an infrared camera. In order to measure the complete temperature field of the heating foil, it needs to measure the temperature fields from three angles for the infrared camera. Three T-type thermocouples to act as reference for the infrared thermograph system are fastened on the rear of the stainless foil with 502 glue.

\[ T = -13.6 + 1.46T_0 - 0.00528T_0^2 \]  

The insulated plate to minimize ambient heat transfer loss is made of a 20 mm thick Bakelite slab, which has a very low thermal conductivity of 0.06 W/(m·°C). Three T-type thermocouples are glued on the outside of the insulated plate.

There are two kinds of rib arrangements for the rib roughened channel, one is symmetric arrangement, another is staggered arrangement. The geometry dimensions of rib roughened channel are shown in Fig. 4 and Table 1.

3. Data reduction

The Reynolds number is computed using the expression

\[ Re = \frac{uD}{\nu} \]  

where \( u \) is the velocity in the test section, which is computed from the mass flow rate calculated from the mass flow controller. \( D \) is the hydraulic diameter of the test section, \( \nu \) is the kinematic viscosity.

The heat transfer coefficient, \( h \) is estimated as

\[ h = \frac{Q - Q_{\text{loss}}}{A(T_w - T_a)} \]  

where \( Q = UI \) is the heat flow of the stainless foil heated by passing DC power which has the accuracy of ±0.1 V and ±0.1 A for voltage \( U \) and electric current \( I \), respectively. \( A \) is the heat area of stainless foil. \( T_w \) is the wall temperature of stainless foil. The temperature \( T_a \) measured by the T-type thermocouple is the air temperature at the entrance of the test section. \( Q_{\text{loss}} = Q_{\text{con}} + Q_{\text{rad}} \) is the heat loss from the outer surface of the insulated plate to ambient, which includes natural convection heat loss \( Q_{\text{con}} \) and radiation heat loss \( Q_{\text{rad}} \). The natural convection heat loss \( Q_{\text{con}} \) is calculated by:

\[ Q_{\text{con}} = h_{\text{nat}}(T_{ow} - T_{amb}) \]  

where \( h_{\text{nat}} \) is the natural convection heat transfer coefficient which can be obtained by [42,33]:
\[ \text{Q}_\text{loss} = \frac{r \cdot A}{T_\text{rad}^4 - T_\text{amb}^4} \] (6)

where \( r = 5.67 \times 10^{-8} \text{ W/(m}^2 \text{ K}^4) \) is the Stefan-Boltzmann constant, \( e_\text{amb} = 0.6 \) is the outer surface emissivity of the insulated plate.

The temperature distribution on the surface of stainless foil (heated or unheated) was recorded by an infrared thermography system operating in the middle IR band (8–14 \( \mu \text{m} \)) of the infrared spectrum. The temperature calculation method of the temperature map obtained from the infrared thermography system was described in Ref.[43,34]. During all of the experiments, the outer surface temperature of the insulated plate was between 25.5 \( ^\circ \text{C} \) and 25 \( ^\circ \text{C} \) while ambient temperature was approximately 24 \( ^\circ \text{C} \). The difference between ambient temperature and outer surface temperature of the insulated plate was less than 1.5 \( ^\circ \text{C} \). So, the natural convection heat loss is about 0.1 \text{ W}, the radiation heat loss is about 0.145 \text{ W}, the total heat loss \( Q_\text{loss} \) is about 0.245 \text{ W} which is far less than the heat flow \( Q \). Thus, it is reasonable to ignore the heat loss from the outer surface of the insulated plate.

The average Nusselt number is defined as

\[ \text{Nu} = \frac{hD}{\lambda} \] (7)

where \( \lambda \) is the thermal conductivity of the air.

The friction factor is defined as

\[ f = \frac{1}{2} \frac{\rho u^2}{\rho L} \] (8)

where \( \rho \text{ in} \) and \( \rho \text{ out} \) are the pressures of the inlet section and outlet section of the test section, respectively. The average inlet velocity \( u \) is calculated using the channel mass flow rate. \( \Delta L \) is the length of the test section.

According to Holman [44,35], experimental uncertainties in average Nusselt number, friction factor measurement were estimated to be about \( \pm 9.5\% \) and \( \pm 6.3\% \), respectively. The individual uncertainty in air stream temperature \( T_a \) was \( \pm 0.4 \text{ } ^\circ \text{C} \); heating foil temperature \( T_w \) was \( \pm 0.1 \text{ } ^\circ \text{C} \).

### 4. Numerical setup

The computational results are used to analyze the results of the experiment. In this study, flow structures from numerical simulations are mainly employed in the analysis. FLUENT-6.3 was used to solve the Reynolds-averaged Navier-Stokes equation discretized by finite-volume method with second-order accuracy. The SIMPLEC algorithm was selected to handle the pressure and velocities. The diffusive terms were dispersed by central difference scheme, while second order upwind scheme was chosen for the discretization of the convective terms. The low-Reynolds number \( k--e \) model was used to simulate the turbulent flow and heat transfer in ribbed channel, which deals with a wide range of Reynolds number including laminar-turbulent transition. Flow was assumed to be incompressible, which is consistent with the flow in the experiment. Due to the application of the commercial software FLUENT, the mathematical description of continuity equation, N-S equation and energy equation can be seen from the Fluent Manual.
4.1. Boundary conditions

A constant and uniform heat flux that was the same as the experiment was applied on the bottom wall of the ribbed channel. No-slip velocity conditions were applied at all walls. Uniform inlet velocity related with the Reynolds number was set and the inlet temperature was fixed at 300 K. The turbulence intensity of inlet was 4%. The outflow boundary condition was chosen as outlet.

4.2. Grid independence

In general, the structured mesh provides greater accuracy than the unstructured mesh and was, thus, employed in this study. Fig. 5 shows the computational grid for symmetrical ribbed channel and staggered ribbed channel. From the figures, it can be seen that the grids near the boundary (the walls and ribs) are very dense to ensure the wall y-plus value to be around 1.0, while in other regions the grid is relatively sparse. The mesh number and the quality have an important impact on the accuracy of the numerical results. Thus, four different structured grid systems had been tested for the same ribbed channel. Fig. 6 shows the comparison of average Nusselt number of different computation meshes. A relatively small deviation is seen when the grids number is larger than 800,000, while a relatively larger deviation is found between 500,000 and 80,000. Therefore, to keep a balance between computational economy and grid independence of the numerical simulation, the grid intensity of 800,000 cells was chosen for all ribbed channels.

5. Results and analysis

5.1. The comparison of temperature contours and Nu between experiment results and numerical simulations

Fig. 7 shows the infrared temperature contours and the numerical simulation temperature contours and flow fields in the ribbed...
channel with symmetrical arrangement ribs at $Re = 3250$. It can be seen that, for the ribbed channel with $S/e = 5$, there is no any flow deflection in the symmetrical arrangement ribs, and a large vortex is formed in the groove between the adjacent ribs, which results in a larger high temperature region in the groove between the adjacent ribs. However, for the ribbed channel with $S/e = 10$, the streamline appears stronger deflection induced by the ribs, forming a wave flow pattern in the ribbed channel. A wave cycle of the streamline is in the range of two grooves. In one wave cycle, due to the stronger flow reattachment to the lower wall in the first half wave cycle, there is no obvious higher temperature region in the region I and III of the lower wall. However, a large vortex filled in the entire groove pushes the streamline upward to the upper wall, resulting in a higher temperature region in the groove II and IV. For the ribbed channel with $S/e = 15$, there exist a separation vortex behind the ribs, and the deflection and the reattachment of the streamline are weakened, which results in two high temperature regions and a low temperature in a groove on the lower wall. Clearly, compared with the ribbed channels with $S/e = 5$ and 15, the wall temperature in the ribbed channel with $S/e = 10$ is the lowest, and the wall temperature in the ribbed channel with $S/e = 5$ is the highest. Fig. 8 shows the comparison of Nusselt number between experimental results and numerical simulation results. It can be seen that the experimental results agree well with the numerical simulation results.

5.2. Effect of rib pitch-to-height ratio $S/e$

Fig. 9 shows the effect of rib spacing on Nusselt number for staggered and symmetrical rib channels. It can be seen that the rib pitch-to-height ratio has a large effect on the heat transfer coefficient of rib roughened channel. The rib pitch-to-height ratio equal to 10 corresponds to the largest heat transfer coefficient whether the rib arrangement is symmetrical or staggered. Clearly, the rib pitch-to-height ratio equal to 15 corresponds to the smallest heat transfer coefficient for symmetrical rib channel and staggered rib channel. The heat transfer coefficient of rib pitch-to-height ratio equal to 5 is between $S/e = 10$ and 15. This indicates that, for the ribbed channels with larger blockage ratios ($0.33, 0.2$), the effect of rib spacing on heat transfer coefficient is not in a monotonous increasing trend with the increase of rib spacing but an optimum rib pitch to height ratio. The optimum pitch to height ratio in this experiment is 10. At the same time, it also can be seen clearly that the heat transfer coefficient linearly increases with the increase of Reynolds number.

In order to have an intuitive understanding for the relationship between the flow structure and the convective heat transfer in ribbed channels, Fig. 10 shows the comparison of flow structures in three kinds of ribbed channels obtained from numerical simulation at $Re = 3750$ and $e/H = 0.33$. It can be observed that, for the ribbed channel with $S/e = 5$, the flow deflection in the staggered ribbed channel is very weakened, a separation region appears in the groove between the adjacent ribs, forming a big separation vortex and reattachment to the channel wall. However, there is no distinct flow reattachment and impingement to the ribbed wall. For the ribbed channel with $S/e = 10$, the streamline appears stronger deflection in the ribbed channel due to the staggered ribs, which results in distinct separation region and reattachment region between ribs, the flow strongly reattaches to the ribbed wall within the groove and detaches again quickly. For the ribbed channel with $S/e = 15$, the flow deflection is weaker than that in the ribbed channel with $S/e = 10$. Thus, there are three regions in the groove between ribs, that is separation region, reattachment region and flow over flat region. In general, when the fluid flows in ribbed channel, the main reasons enhancing convective heat transfer are flow separation vortex behind rib and reattachment to the channel wall. However, as the increase of the rib spacing, the effects of separation vortex and reattachment on convective heat transfer are weaker and weaker, but the effect of flow over a flat region on convective heat transfer is gradually increased. When rib spacing is large enough ($S/e = 15$), the convective heat transfer of flow over flat is the main reason influencing on the convective heat transfer in the ribbed channel, resulting in a lower convective heat transfer coefficient.

On the other hand, the flow reattachment to the rib wall will be gradually weakened as the decrease of rib pitch. When the rib pitch is small enough ($S/e = 5$), the phenomena of flow reattachment will be disappeared, which also results in a lower heat transfer coefficient in the rib roughened channel. Thus, only when the rib spacing is appropriate ($S/e = 10$) in ribbed channel, both of the flow separation vortex and reattachment to rib wall can play main roles on the convective heat transfer, leading to an optimum convective heat in the rib roughened channel.

5.3. Effect of rib arrangement

Fig. 11 shows the effect of rib arrangement on the heat transfer coefficient in ribbed channel with the blockage ratios $e/H = 0.33$ and 0.2. The plotted results show that rib arrangement has a larger effect on the heat transfer coefficient of ribbed channel. The average Nusselt number of symmetric ribs is higher than that of staggered ribs. The difference of average Nusselt number between symmetric ribs and staggered ribs is increased with the increase
Fig. 9. Effects of rib pitch-to-height ratio S/e on Nusselt number.

Fig. 10. Comparison of flow fields in ribbed channel (Re = 3750 and e/H = 0.33).
of Reynolds number. For the ribbed channel with blockage ratio $e/H = 0.33$, the difference is about 10 at the Reynolds number equal to 1500, but the difference is up to 40 at the Reynolds number equal to 4000. For the ribbed channel with blockage ratio $e/H = 0.2$, the difference is about 5 at the Reynolds number equal to 2600, and the difference is up to 40 at the Reynolds number equal to 9000. In addition, the difference of Nusselt number between symmetric ribs and staggered ribs is different for different blockage ratio $e/H$. Clearly, it is increased as increasing the blockage ratio $e/H$. The difference of Nusselt number for $e/H = 0.33$ is larger than that for $e/H = 0.2$.

These phenomena can be explained by the numerical simulation results. Fig. 12 shows the flow fields of different rib arrangements. It can be seen that, when the blockage ratio $e/H$ is 0.33, the streamlines reveal stronger deflection and disturbance induced by the ribs, forming a waved flow pattern in the rib roughened channel, which results in flow separation downstream of the rib and flow reattachment to the channel floor. However, there is a larger difference for the waved flow patterns when the fluid flows in staggered rib channel and symmetric rib channel. For staggered rib channel with $e/H = 0.33$ (Fig. 12a), a wave cycle of the streamline is in the range of a groove, which results in the regions of flow separation and reattachment within a groove. For symmetrical rib channel with $e/H = 0.33$ (Fig. 12b), the streamline also shows strongly waved pattern of the same as staggered rib channel, but the wave cycle of the streamline is in the range of three grooves, which results in a large difference from the staggered rib channel. A huge separation region exists in the first upper groove, which results in a large vortex of the same as the upper groove between ribs. The streamline pushed downward by the upper vortex attaches to the floor, accompanied by the strong reattachment and impingement to the ribbed wall. In addition, the flow cross-section

![Image](image-url)
area is also strongly compressed by the huge vortex, which results in increasing the flow speed in the symmetrical rib channel. In the second groove of the symmetrical rib channel, the flow pattern is opposite to the first groove, and the flow pattern is the same as the first groove in the third groove. Thus, when the blockage ratio e/H is 0.33, the disturbance of the fluid and turbulent mixing in the symmetrical rib channel is stronger than that in the staggered rib channel, which results in a stronger heat transfer characteristics than that in staggered rib channel. Fig. 12c shows the streamline profile of the staggered rib channel with e/H = 0.2. In contrast to the staggered rib channel with e/H = 0.3, the streamline deflection is weakened. Thus, the intensity and size of the separation vortex behind the rib are all smaller than that in the staggered rib channel with e/H = 0.33. Fig. 12d shows the streamline profile of symmetrical rib channel with e/H = 0.2. It can be seen that its streamline profile is different from symmetrical rib channel with e/H = 0.33, and the streamline appears continuous contraction and expansion as the fluid flows through the rib roughened channel, which results in continuous separation vortex behind the ribs. Because the streamline deflection does not appear in the symmetrical rib channel, the separation vortexes behind the ribs can expand freely in the groove between ribs. In addition, the decrease of the flow cross-section area between two ribs on top and bottom floors leads to an increase of flow speed, forming a stronger flow disturbance in the symmetrical rib channel with e/H = 0.2 than that in staggered rib channel with e/H = 0.2, which results in a higher heat transfer coefficient than that in staggered rib channel with e/H = 0.2. Fig. 12e shows the flow profile in one side rib channel. It can be seen that a boundary layer separates upstream and downstream of ribs, resulting in smaller separation vortexes before the ribs and larger separation vortexes behind the ribs. The flow separations reattach the boundary layer.
to the floor, thus increasing the heat transfer coefficient. Moreover, the separation vortexes in the rib channel enhance turbulent mixing, therefore the heat from the near surface fluid can more effectively get dissipated to the main flow, increasing the heat transfer coefficient. However, one side ribs in the rib channel disturb only the near-wall flow, and consequently the flow disturbance in the one side rib channel is smaller than the other cases.

5.4. Effect of blockage ratio e/H

Fig. 13 shows the effect of rib blockage ratio on heat transfer coefficient. It can be seen that there is a large effect for the blockage ratio on the heat transfer coefficient. In general, a larger blockage ratio in the ribbed channel corresponds to a higher heat transfer coefficient. So, the heat transfer coefficient of blockage ratio e/H = 0.33 is higher than that of blockage ratio e/H = 0.2. However, because of the difference of rib arrangements, the difference of heat transfer coefficient between the two blockage ratios is also different. Clearly, the difference of heat transfer coefficient for the symmetrical rib arrangement is higher than that for the staggered rib arrangement. The reason to explain this kind of phenomena can be obtained from the Fig. 12. It can be seen that whether symmetrical rib channel or staggered rib channel, the flow disturbance in the rib channel with e/H = 0.33 is significantly higher than that in the rib channel with e/H = 0.2, which results in a larger heat transfer coefficient for the rib channel with e/H = 0.3.

5.5. Comparison of heat transfer coefficient between one-side ribs and two-side ribs

Fig. 14 shows the comparison of heat transfer coefficient between one-side rib channel and two-side rib channel. Clearly, the heat transfer coefficient of two-side rib channel is distinct higher than that of one-side rib channel. The difference is increased with increasing flow Reynolds number, which is from the smallest value 10 at the Re = 3000 up to the largest value 60 at the Re = 9000. The reason for this phenomena can be obtained from Fig. 12. It can be seen from Fig. 12a, c and e that both of the flow separation vortexes and reattachment to the heat wall in the one-side rib channel are distinct weaker than that in two-side rib channel, thus the disturbance in the one-side rib channel is far less than that in the two-side rib channel, which results in a weaker turbulent mixing in one-side rib channel than that in two-side rib channel. So, the heat transfer coefficient of two-side ribbed channel is distinct larger than that of one-side ribbed channel.

5.6. Heat transfer enhancement

In order to analyze the heat transfer enhancement for the ribbed channel, the Nusselt number of smooth channel Nu0 must be obtained by an additional experiment in this paper. Fig. 15 shows the experiment results and empirical relation for the rectangular smooth channel of 180 mm (length) × 60 mm (width) × 15 mm (height) when the Reynolds number is from 950 to 8100. The empirical relation is 

\[ Nu_0 = 3.12 + 0.003Re \]

Fig. 16 shows the average Nusselt number ratio Nu/Nu0 for the ribbed channels of e/H = 0.33 as a function of the Reynolds number. It can be seen clearly that the Nusselt number ratios is almost unchanged with the increase of Reynolds number from 1000 to 8000. For the symmetrical rib channel of e/H = 0.33, when S/e = 15, Nu/Nu0 is about 5; when S/e = 10, Nu/Nu0 is in the range of 6–7, and when S/e = 5, Nu/Nu0 is in the range of 5–6. For the staggered rib channel of e/H = 0.33, when S/e = 15, Nu/Nu0 is about 5; when S/e = 10, Nu/Nu0 is about 6–7, and when S/e = 5, Nu/Nu0 is about 5. For the symmetrical rib channel of e/H = 0.33, when S/e = 15, Nu/Nu0 is about 6–7, and when S/e = 5, Nu/Nu0 is about 4. Obviously, the convective heat transfer is greatly enhanced by the ribs in the channel. The heat transfer enhancement of symmetrical rib channel is higher than that of staggered rib channel.
5.7. Pressure drop

Since the heat transfer enhancement in ribbed channel is typically accompanied by an increase in pressure drop, a comprehensive analysis of performance must include the effect of ribs on channel pressure drop. Fanning friction factor of the ribbed channels can be calculated according to Eq. (8). Fig. 17 shows the experimental results for the friction factor in different ribbed channels. The friction factor, deduced from the experimental data, is depicted as a function of \( Re \). It can be discovered that the friction factor is decreased gradually with the increase of Reynolds number. Whether symmetrical rib channel or staggered rib channel, the ribbed channel of \( S/e = 10 \) corresponds to the largest friction factor, and the smallest friction factor exists when \( S/e = 15 \). At the same time, it can be seen clearly that the friction factor of the symmetrical rib channel is larger than that of staggered rib channel under the same \( S/e \). The \( e/H \) also has a larger effect on the friction factor under the same \( S/e \). For example, for the symmetric rib channel, when \( S/e = 10 \), the largest and the smallest friction factors of the rib channel of \( e/H = 0.33 \) are 7.7 and 6.5 respectively, however, the largest and the smallest friction factors of the rib channel of \( e/H = 0.2 \) are 3.25 and 2.8 respectively. From above we can know that the optimum heat transfer coefficient corresponds to the largest friction factor, and the minimum heat transfer coefficient corresponds to the smallest friction factor.

6. Conclusions

Heat transfer characteristics and pressure drop in rectangular ribbed channels have been experimentally investigated by means of infrared thermography. Based on the results, the following conclusions are drawn:

(1) The rib spacing has a large effect on the heat transfer of the ribbed channel. The effect of rib pitch to height ratio on heat transfer coefficient is not in a monotony increase trend but an optimum rib pitch to height ratio. The optimum pitch to height ratio in this experiment is 10.

(2) The rib arrangement has a larger effect on the heat transfer coefficient of ribbed channels. In general, the average Nusselt number of symmetric ribs is higher than that of staggered ribs.

(3) A larger blockage ratio in the ribbed channel corresponds to a larger heat transfer coefficient. The heat transfer coefficient of blockage ratio \( e/H = 0.33 \) is larger than that of blockage ratio \( e/H = 0.2 \). The heat transfer coefficient of two-side rib channel is larger than that of one-side rib channel.

(4) The heat transfer enhancement effectiveness of ribbed channel in the laminar flow state is far higher than that in the turbulent state. The optimum heat transfer coefficient corresponds to the largest friction factor, and the minimum heat transfer coefficient corresponds to the smallest friction factor.
Acknowledgments

This work was supported by the Research Program of the National Natural Science Foundation of China, No. 51276088.

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