Study on Thermofluid Characteristics of a Lattice Cooling Channel

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ABSTRACT
Thermal fluid characteristics of the lattice cooling channel have been investigated by measuring distributions of heat transfer coefficient. The heat transfer coefficient distributions have been obtained by visualizing steady-state local temperatures using temperature sensitive liquid crystal and foil heater. The measurement has been performed for all surfaces consisting of a single passage, while past studies focused only on the bottom (primary) surface. By comparing heat transfer coefficient distributions on the primary surface, sidewalls and turning regions with velocity field which was measured in the previous study, the heat transfer mechanism of the lattice cooling channel is discussed.

The lattice cooling channel consists of two sets of inclined parallel ribs, which cross each other at right angles. Reynolds number is varied from 2,000 to 9,000, which is based on hydraulic diameter and bulk velocity in a sub-channel. Heat transfer patterns on the sub-channel before and after turning are compared.

Overall, the heat transfer distribution was well-correlated with the velocity field. In the sub-channel before turning, heat transfer is enhanced at the entrance of the primary surface due to flow acceleration. Heat transfer is also enhanced on the sidewalls. Whether the coolant turns or not, shear force exerted by the crossing flow at the diamond-shaped openings generates swirl flow motion, leading to enhanced heat transfer. In the sub-channel after turning, this trend is further increased due to longitudinal vortex formed by the turning. Moreover, comparison of heat transfer patterns on the sidewalls with that on the primary surface suggested that they are comparable due to the flow interaction. It suggests that contribution of the rib surface on total heat release should be taken into account for reasonable cooling design.

NOMENCLATURE

| Symbol | Description |
|--------|-------------|
| A      | Area        |
| $D_{hs}$ | Sub-channel hydraulic diameter |
| $H_s$  | Sub-channel height |
| $h$    | Heat transfer coefficient |
| $L$    | Streamwise distance from the sub-channel inlet |
| $Q$    | Heat flux |
| Nu     | Nusselt number |
| Nu_s   | Nusselt number of a smooth channel |
| Pr     | Prandtl number |
| Re_s   | Sub-channel Reynolds number |
| $T_a$  | Air temperature |
| $T_w$  | Wall temperature |
| $V_s$  | Bulk velocity of sub-channel |
| $W_s$  | Sub-channel width |

INTRODUCTION
Increasing turbine inlet temperature with a small amount of coolant is of great importance to realize more eco-friendly and economical power generation with gas turbine. Typically, the turbine blades or vanes are internally-cooled using air bled from the compressor to avoid failure due to high thermal load. The internal cooling channel has heat transfer enhancement devices such as ribs, dimples, and pin-fins. To improve their thermal performance, many research papers have been presented since the 1970s, which are summarized in Han’s textbook [1].

Lattice (matrix) cooling structure is one of the approaches to enhance heat transfer by refining the internal cooling structure. Its basic concept is illustrated in Fig. 1. The internal hollow space of the blade or vane is divided by two (or more) sets of equally-spaced, inclined parallel ribs to form small sub-channels on the backside of the airfoil. As shown in Fig. 1, two rib assemblies cross each other. Coolant entered into each sub-channel turns its flow direction at the endwall and moves to the following opposite sub-channel. The flow impingement at the side wall generates strong swirl, leading to formation of longitudinal vortex in the next sub-channel.

Fig. 1 Schematic of lattice cooling structure

In addition, by crossing the upper and lower ribs, diamond-shaped intersections are formed. Khalatov and Nam [2] explained that coolants flowing in the upper and lower sub-channels communicate each other at the intersections accord-
ing to pressure difference and exchange rotational momentum. The constant supplement of swirl contributes to sustain the longitudinal vortex structure over the sub-channel. As a result, complicated spiral streamlines are formed over the entire duct. Heat transfer in the sub-channel is enhanced compared to that in a smooth duct.

The experimental study on heat transfer in the lattice cooling channel has been conducted by several authors [3-6]. In the early stage, Goreloff et al. [3] has measured heat transfer and pressure loss of a turbine blade with lattice cooling structure using Zinc melt technique, varying sub-channel inclination angle ($\beta$) from 15 to 60 degrees. Measured results have shown that overall Nusselt number enhancement factor is from 1.0 to 2.0 for Reynolds number based on sub-channel hydraulic diameter between 5,000 and 20,000. It is suggested that the optimal value of $\beta$ for heat transfer enhancement is 45 degrees; on the other hand, pressure loss increases with increase in $\beta$.

Gillespie et al. [4] and Saha et al. [5] have reported detailed heat transfer coefficient distributions of primary (bottom) surfaces of a tapered lattice cooling channel. As mentioned in Ref. [2], higher heat transfer coefficients have been observed near the endwall due to flow impingement. Bunker [6] has measured local heat transfer coefficients on the primary surfaces of acrylic and metal lattice models to assess the effect of fin effectiveness of the ribs. The rib inclination angle is 45 degrees. Reynolds number based on sub-channel hydraulic diameter is varied from 20,000 to 100,000. Nusselt number distributions have shown that heat transfer enhancement at turning regions are sustained for long distances probably due to the existence of the longitudinal vortex [2]. Comparison of heat transfer enhancement between acrylic and metal models has proved that the thermal effect of the ribs, which act as fins, is not negligible.

To understand the flow and heat transfer mechanism, flow visualization of the lattice cooling duct is needed; however, the complicated geometry makes it quite hard due to difficulty with optical access. Therefore, numerical investigations have been performed to discuss the flow characteristics [7-11]. Most predicted results have shown similar trends to those described by Khalatov and Nam [2]; however, prediction accuracy of the velocity field has never been quantitatively evaluated.

The authors have measured three-dimensional velocity field inside the lattice cooling channel using Magnetic Resonance Velocimetry (MRV) [12]. The working fluid is a copper sulfate aqueous solution, and Reynolds number based on the sub-channel hydraulic diameter is approximately 8,000. Rib inclination angle varies from 30 to 60 degrees. For the rib inclination angle of 45 deg, overall flow structure is like that explained in Ref. [2]. However, it was found that there are three flow patterns in the openings between opposing sub-channels according to the number of pairs which have positive and negative values of the vertical velocity component. When the ribs are arranged with obtuse angle, a large vortex spreads across the contact surface while the vortex structure independently stays in each sub-channel for acute rib angle. The stripe pattern of the vertical velocity component, observed in the middle of the sub-channels, is due to a pair of the large vortex and accompanying small vortex. Moreover, the measured results are compared with numerically predicted ones. Although overall trends are captured with numerical calculation, flow interaction between the upper and lower sub-channels is overestimated at the turnings and underestimated in the intermediate regions.

The goal of the present study is to clarify flow and heat transfer enhancement mechanism of the lattice cooling channel and to provide data for validation of the numerical work. For this purpose, local heat transfer coefficient distributions are measured for all surfaces composing one sub-channel. In Refs. [6] and [10], it is suggested that heat transfer from the side walls significantly contributes to the overall heat release. By comparing measured heat transfer coefficient distributions among three surfaces, the amount of contribution will be discussed.

### EXPERIMENTAL APPARATUS

General arrangement of the heat transfer test apparatus is shown in Fig. 2. Air is supplied to the test section through an inlet bell mouth using an electric suction-type blower (2.2kW). Flow rate is controlled using an orifice flow meter and ball valves. The test section, colored yellow green in the figure, is 1,000mm long, 700mm wide and 120mm height. The material of the model is rigid foam PVC (polyvinyl chloride), which is one of engineering plastics that has favorable workability.

Dimensions of the sub-channel are listed in Table 1. The inclination angle of the ribs is set at 45deg in this study. Reynolds number based on the sub-channel hydraulic diameter and bulk velocity is varied from 2,000 to 9,000. The heat transfer experiment is conducted for one passage (red, blue and green areas in Fig. 3). The passage consists of the lower sub-channel, upper sub-channel, and turning region. Steady state inner wall temperature distributions of those sub-channels are measured under constant heat flux. Thin foil heater (thickness = 10 $\mu$m) is adhered on all surfaces of the passage to give Joule heating. To measure spatial distributions of temperature, temperature-sensitive cholesteric liquid crystal sheet (Japan Capsular Products Inc., Type RW-3035) is adhered to the heater of each surface. It has wide-band temperature response in the range between 30 and 35 degC. Temperature variation due to the sheet thickness, estimated at the Reynolds number of 9,000, is $\pm 0.017$ degC.

When the heater is energized with a pair of electrodes located at both ends of the wall (see Fig. 4(b)) under the certain air flow condition, the liquid crystal changes its color depending on temperature. Color distribution is obtained by taking a photograph of the liquid crystal. The color information is digitized as hue value using an image processing software. For this purpose, the passages are made of transparent acrylic resin. For the sidewalls, it is impossible to face the camera toward the surfaces due to the existence of the neighboring sub-channel. In this experiment, a mirror is obliquely placed between the photographing target and the neighboring sub-channel to take a picture from the outside, as shown in Fig. 4(a). By correcting the projection angle, temperature distributions on the sidewalls are visualized. In addition, several K-type thermocouples are installed on the back of the foil heater to check color indication of the liquid crystal, as shown in Fig. 4(b).

As a matter of course, heat transfer characteristics vary before and after turning at the duct endwall. In this paper, the sub-channels before and after turning are called entrance and basic region, respectively. To compare the heat transfer mechanisms of those regions, surface temperature distributions of both sub-channels are measured by changing flow direction. That is, temperature distribution of the entrance region is measured supplying air from the right hand side in Fig. 3(a). On the other hand, when measuring temperature variation in the basic region, the flow direction is reversed as shown in Fig. 3(b). The experiment is therefore performed twice for each Reynolds number setting.

| Table1 | Dimensions of the sub-channel |
|--------|-----------------------------|
| Sub-channel height $H_1$ [mm] | 50 |
| Sub-channel width $W_s$ [mm] | 75 |
| Rib inclination angle $\beta$ [deg] | 45 |
| Number of the sub-channel | 6 |
Fig. 2 Test apparatus

Fig. 3 Flow direction for measurement

Fig. 4 Cross section of measurement surface
DATA REDUCTION

As mentioned above, wall temperature is measured to obtain heat transfer coefficient under steady state condition. Thermal balance on the wall is described as

\[ Q = hA(T_w - T_a) \]  

Heat transfer coefficient is therefore derived as

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

The heat transfer coefficient is non-dimensionalized using Nusselt number based on the sub-channel hydraulic diameter as

\[ Nu = \frac{hD_{ha}}{\lambda_a} \]  

Nusselt number is normalized to evaluate the amount of heat transfer enhancement. Nusselt number of developed turbulent flow in a smooth pipe is chosen as reference, which is determined by Dittus and Boelter as follows;

\[ Nu_e = 0.023Re^{0.8}Pr^{0.4} \]  

As well as the definition of Nusselt number, Reynolds number is based on hydraulic diameter and bulk velocity of the sub-channel. The left hand side of eq. (1) is a heat input to the heater, calculated using the electric current and voltage. Air temperature \( T_a \) is measured at the inlet of the test apparatus using a K-type thermocouple.

Color response of the liquid crystal is converted into temperature by the hue capturing technique \[13\]. From a color image captured with a digital camera, red, green and blue signals at each pixel are obtained and converted into hue value. Calibration of the liquid crystal response is performed at three points (entrance, center and exit of the sub-channel) on the primary surface by comparing the crystal response to the infrared absorption on the mirror or brightness. Therefore, the calibration curve for the primary surface is applicable to the sidewalls. Uncertainty of temperature is estimated to be 7.1 % at 95% confidence.

RESULTS AND DISCUSSIONS

Spatial distributions of heat transfer enhancement factor \( (Nu/Nu_{s}) \) on the sub-channel before and after turning are shown in Fig. 6. The sub-channel Reynolds number \( (Re) \) is set at 8,000 to match the velocity measurement condition. In Fig. 7, \( Nu/Nu_{s} \) plots on equally dividing lines quadrisecting each surface (illustrated in Fig. 6) are compared between the entrance and basic regions. The percentage shown on the sidewalls corresponds to vertical distance from the bottom (primary surface). The heat transfer patterns for other Reynolds numbers tested have shown similar trends in both regions; therefore, the result for \( Re = 8,000 \) is regarded as the representative. For comparison, velocity distributions measured in the previous study are shown again in Fig. 8. In addition, contour plots of vertical velocity component on the contact surface between two sets of crossing sub-channels are also shown in Fig 9 to discuss the effects of flow interaction on the heat transfer.

Local heat transfer distribution in the entrance region

Comparison between Figs. 6 and 8 suggests that the heat transfer pattern is well correlated with velocity field on the primary surface. It shows similar trends as that of typical channel flow. Around the inlet, heat transfer is enhanced due to flow acceleration associated with separation at the corner. From Fig. 7(a), the maximum Nusselt number is found at the apex of the inlet and gradually decreases to the smooth channel value at \( L/D_w = 5 \). Near the turning zone, heat transfer increases again at 75% of the channel width while it constantly decreases at other locations. As can be seen in Fig. 8, the second heat transfer enhancement is correlated with strong downward flow at the turning region. Swirling flow at the turn that has been reported in the past studies is generated there, and heat transfer is accordingly enhanced.

On the sidewall A, the heat transfer is enhanced between the first and second ribs, where the influx reattaches. Note that the pale blue rectangles in Figs. 7(b) and (c) show the ribs forming the opposite sub-channels. Around the third rib, heat transfer increases again at 25% of the channel height from the bottom (primary surface). The heat transfer increase is more noticeable for the sidewall B. On the sidewall B, heat transfer has a peak above the third rib for all three vertical locations. The Nu/Nu_{s} value becomes greater toward the top of the sidewall, that is, the opposite sub-channels. As described in the Introduction, coolant flowing in the lower and upper sub-channels contacts at the diamond-shaped openings in the middle of the duct. Crossing of two flows at the openings generates counterclockwise swirl motion when viewing from the outlet in the present rib arrangement. Therefore, the inflow from the lower sub-channel occurs at the sidewall B and impingement of swirling flow occurs at the sidewall A in the upper sub-channel where temperature is measured. Those are supported by the distribution of measured vertical velocity component on the contact surface, Fig. 9. As a result, Nusselt number of the sidewalls is increased to approximately twice the smooth channel value. It is found that the flow interactions between the opposing sub-channels have already started to raise the amount of heat transfer on the sidewalls in the entrance region. The above-mentioned, increased heat transfer in the middle of the primary surface is therefore caused by the swirl generated by the inflow as shown in the velocity vectors near the primary surface (Fig. 8(b)); however, it seems smaller than that for the sidewalls.
Fig 6  Distributions of heat transfer enhancement factor (Nu/Nu₀) on the sub-channel
(upper: entrance region, lower: basic region)

Fig 7  Comparison of longitudinal Nu/Nu distributions

(a) Primary surface
(b) Sidewall A
(c) Sidewall B
Local heat transfer distribution in the basic region

Spatial distribution of Nu/Nu_{str} on the sub-channel after the turn is shown in Fig. 6(b). It is clearly seen that heat transfer is substantially increased over the entire sub-channel. On the primary surface, Nusselt number increases to about four times the smooth pipe value at the impingement region and the increased heat transfer is sustained to the end of the sub-channel in the upper half of the surface. The longitudinal Nu/Nu_{str} plots in Fig. 7(a) show almost flat profile at 25% of the channel width while two other profiles show monotonical decrease downstream. That is due to accelerated flow derived from the longitudinal vortex formed at the turning. Near the sidewall B, the coolant is lifted up from the primary surface along the vortex. The Nu/Nu_{str} value thus decreases at 50% and 75% of the channel width. Nevertheless, the heat transfer is substantially enhanced over the channel.

The vortex generated in the turning region is categorized to forced vortex. In a typical straight pipe, increased tangential velocity in the outer radius substantially decreases downstream due to friction of the wall. In the lattice cooling channel, on the other hand, angular momentum is constantly added in the contact surfaces, which probably leads to sustained vorticity in the sub-channel. That is supported by the swirl velocity near the primary surface, shown in Fig. 8(b), distributed over the sub-channel. More quantitative discussion should be made by comparing the present results with flow and heat transfer of a simple pipe with inlet swirl, for example.

Moreover, there is a pair of wide and narrow bands of increased heat transfer. That is probably attributed to the stripe pattern of the vertical velocity component discussed in the previous study [12]. The previous study has suggested that the stripe pattern of the vertical velocity component is caused by generation of several vortices in the sub-channel. The similar stripe pattern has also been reported by Saha et al. [5] although the Reynolds number is an order of magnitude greater than that for the present study.

As well as the entrance region, higher heat transfer is found on the sidewalls; however, the heat transfer pattern is quite different. It implies that direction and angle of the rib inclination determines the heat transfer patterns. In the sidewall A, heat transfer at the edge connected to the opposite sub-channels periodically increases. The increased heat transfer region is not only on the contact surface, but also extended to the intersection of the opposing ribs. From Fig. 8(a), swirl flow direction is inclined at approximately 45 degrees against cross section of the sub-channel. Impingement region of influx from the opposing sub-channel therefore shifts downstream along the streamline. The discontinuous heat transfer pattern is also seen in the primary surface as wavy isotherms. On the other hand, heat transfer near the primary surface (25 and 50% of the channel height) is almost constant at around Nu/Nu_{str}=3.

In the sidewall B, the periodic behavior like the sidewall A does not appear. Nusselt number is strongly enhanced at the inlet and gradually decays downstream to similar Nusselt number as the sidewall A. However, it is still greater than that on the primary surface. That is attributed to shear force exerted by the swirling flow. On the sidewall B, the flow swirls toward the top of the rib, that is, the opposite sub-channels. In addition to the effect of the shear force, ventilation to other sub-channels may contribute to increase the heat transfer.

Heat transfer distribution in the turning region

Distributions of Nu/Nu_{str} on the turning region are shown in Fig. 10. Note that the color range is larger than that for previous figures to improve their visibility. In the figures, coolant enters the turning region from the lower left and exits to upper right. Most coolant impinges the square endwall although some bypass flow occurs before turning [12]. Heat transfer enhancement becomes the maximum due to the impingement. Unlike the above-mentioned heat transfer behavior in the sub-channel, heat transfer characteristics in the turning depend on Reynolds number. For Re=4,000, the upper and lower sidewalls show high heat transfer region due to the impingement. However, they become inconspicuous for Re=8,000.

Compared with simple impingement heat transfer on a flat surface, Nusselt number of the stagnation point on the endwall seems substantially larger. Typically, Nusselt number for the impingement region on the flat surface from a round hole is roughly five times higher than Nu_{str}, while the Nu/Nu_{str} on the endwall exceeds 20 for...
the present configuration. There are two plateaus of heat transfer on the endwall. They are derived from strongly whirling flows shown in Fig. 11. The left one is due to the impingement flow from the lower sub-channel, and the right one is a secondary vortex behind turning flow to the upper sub-channel. Heat transfer enhancement by the secondary vortex is less than that by the impingement flow for \( Re_s = 4,000 \); for \( Re_s = 8,000 \), on the other hand, heat transfer of the secondary vortex highly increases to exceed that of the impingement. In addition to the effect of impingement flow, turbulence in the secondary vortex would be closely related to the heat transfer enhancement on the endwall.

In a cooling design of actual blade or vane using the lattice cooling structure, heat transfer pattern in the turning region should be more uniform to effectively cool the material. To provide such configuration, geometry of the turning region needs to be optimized considering favorable combination with the sub-channel geometry. Most studies, including the present paper, have dealt with flat surface as the endwall of the turning region. However, when the lattice cooling structure is arranged in spanwise direction, shown in ref. [3] for example, the turning regions correspond to interiors of leading and trailing edges. The endwall is therefore concavely curved, and the sidewalls are connected to the endwall more smoothly compared with the present model. Further investigation of flow and heat transfer mechanism for such concave turning geometry would be useful for the effective cooling design.

![Fig. 10 Distributions of heat transfer enhancement factor \((Nu/Nu_s)\) in the turning region](image)

**Comparison of area-averaged Nusselt number**

From the local Nusselt number distributions, area-averaged Nusselt numbers for Reynolds number from 2,000 to 9,000 are calculated. Fig. 12 shows a comparison of the averaged Nusselt numbers on all surfaces in each region. Not surprisingly, Nusselt number of the entrance region increases with Reynolds number with similar exponential relationship as turbulent channel flow. Nusselt number of the sidewall A is less dependent on Reynolds number than that of the sidewall B. As discussed in Fig. 6, the impingement flow dominates heat transfer characteristics of the sidewall A. Stagnation region in the lower half of the sidewall A is covered with a laminar boundary layer, leading to the decreased Reynolds number dependence.

The averaged Nusselt number in the basic region increases about two to three times as those in the entrance region. The Reynolds number dependence on the sidewalls furthermore decreased compared with those in the entrance region. It implies that the flow communication highly accelerates coolant in the boundary layers on the sidewalls.

An interesting thing is that a similar amount of heat transfer occurs on the sidewalls as the primary surface. Moreover, for smaller Reynolds number, heat transfer is more enhanced on the sidewalls than the primary surface. It is demonstrated that the contribution of the sidewalls should be taken into account when estimating heat release of the lattice cooling channel.

In the turning region, Nusselt number suddenly increases for \( Re_s = 8,000 \). There might be a transition point of heat transfer at Reynolds number between 6,000 and 8,000. Further investigation of the existence of the transition behavior is needed in the near future.

![Fig. 11 Streamlines calculated using the velocity field [12]](image)
CONCLUSION

To clarify heat transfer enhancement mechanism of the lattice cooling channel, local heat transfer coefficient distributions are measured for all surfaces composing one sub-channel. Distributions of heat transfer enhancement factor are compared with velocity field measured in the previous study [12]. The following conclusions are drawn;

- The heat transfer patterns on the sidewalls are significantly different depending on the location of inflow and outflow. Therefore, direction and angle of rib inclination determines the heat transfer distribution.
- Nusselt number of the turning region is 2-3 times larger than that on the primary surface. Strongly swirling corner vortices further increases heat transfer on the endwall, but the heat transfer on the sidewalls is less enhanced.
- The area averaged Nusselt number depends upon the Reynolds number with similar exponential relationship as that for a turbulent heat transfer. On the other hand, Nusselt number of the sidewalls less depends on Reynolds number.

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