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Heat Transfer Characteristic of a Triangular Channel with Turbulence Promoter

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ABSTRACT

An experimental and analytical study on the heat transfer and pressure loss of a triangular cooling flow channel with and without turbulent ribs has been conducted for high reliability cooling design of gas turbine blades. In addition, the means of enhancing and controlling the local heat transfer on the surface of a triangular ribbed channel, utilizing the secondary flow from the gap between side-wall and rib, are proposed.

The contours of experimental and analytical Nusselt number on the ribbed channel with the gap show the enhancement of heat transfer by re-attachment flow, swirl flow, and secondary flow from the gap. Optimizing the length of the gap enables the enhancement of the heat transfer around the trailing edge and to assure mean heat transfer in the whole of the cooling flow passage.

NOMENCLATURE

d_e	Equivalent hydraulic diameter
f	Friction factor
L	Distance
Nu	Nusselt number
Δp	Pressure drop
q	Heat flux
Re	Reynolds number (= $\rho v d_e / \mu$)
T	Temperature
v	Velocity
h	Heat transfer coefficient
λ	Thermal conductivity
μ	Viscosity
ρ	Density
δ	Wall thickness

Subscript:

f	Fluid
loss	Heat loss
m	Material
wi	Inner surface of the wall
wo	Outer surface of the wall

INTRODUCTION

Advanced gas turbine blades are exposed to high heat load and thermal stress. The internal convection cooling with ribbed channels is often adopted in cooled turbine blades to satisfy their life. Many studies of the heat transfer and pressure loss in the square or rectangular ribbed channel have been conducted. (Bargraff et al., 1970 Han et al., 1985 Anzai et al., 1991 Taslim et al., 1994)

However, there are various geometries of the cross section of ribbed channel installed in cooled turbine blades. Therefore, the heat transfer and pressure loss characteristics of various cross-sectioned channels have also been studied. Metzger et al. (1987) investigated the heat transfer and pressure loss characteristics of a triangular channel which simulate the cooling channel at the leading edge. The effect of aspect ratio (height/width) in rectangular channel was reported by Han et al. (1989) and Tasilim et al. (1988). The heat transfer and pressure loss characteristics of a thin rectangular channel which simulates the cooling channel around a trailing edge is the subject of the study by Kiml et al. (2000, 2001, 2002).

At the trailing edge of cooled turbine blades, the cross section of the cooling channel is a thin triangle. The heat transfer distribution in such a channel is non-uniform and complicated. In addition, it is difficult to cool the trailing edge where the heat transfer area of the gas side is large, compared with that of the cooling side. Therefore, cooling air ejection has often been utilized. Taslim et al. (1997) investigated the heat transfer and pressure loss characteristics of a thin triangular channel with and without air ejection. However, it degrades the aerodynamics performance and cannot be applied to closed circuit steam-cooled turbine blade as shown in Fig.1. To attain high reliability cooling design of the trailing edge, it is indispensable to understand and to predict the local heat transfer coefficient in a triangular ribbed channel precisely.

In this study, the local heat transfer and pressure loss in a triangular ribbed channel is investigated experimentally, and the means of predicting them by CFD (Computational Fluid Dynamics) is investigated. Also, the means of enhancing and controlling the local heat transfer coefficient on the surface of a triangular ribbed channel, by utilizing the secondary flow from the gap between side-wall and rib, is introduced and confirmed the effect experimentally and analytically.

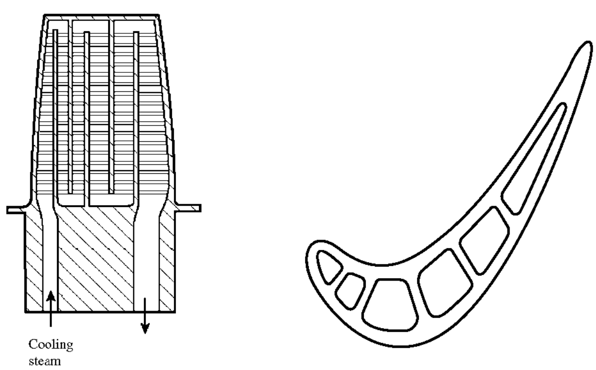

Fig.1 Cross section of steam cooled turbine blade

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EXPERIMENTAL PROGRAM

Experimental Apparatus and Method

Fig.2 shows the schematic drawing of the test section that simulates the cooling channel at the trailing edge. The test channel has a thin triangular cross section whose mean height and width are 17mm and 125mm, respectively. Ribs are installed and staggered on the left and right walls in the channel. The rib height and width are 4mm and 5mm, respectively. The rib pitch-to-height ratio is 12.5 and the attack angle against flow direction is 60 degrees. Four types of heat transfer test models with different rib lengths are shown in Fig.3. One of the models is no-ribs and the others have the gap between narrow side-wall and edge of the rib, whose length is from 0mm to 25mm. The heat transfer model, including the channel wall and rib, is made of low thermal conductivity material, Bakelite.

The local heat transfer is derived from the measured heat flux of an electrically heated thin stainless steel foil adhered to one side of the ribbed walls. Local wall temperatures are measured by approximately 130 thermocouples embedded in the channel wall. The details of the measuring locations are shown in Fig.4. Dry air is used as the test fluid. The Nusselt number Nu is calculated as

$$Nu = hd_e / \lambda_f \qquad (1)$$

$$h = (q - q_{loss}) / (T_{wi} - T_f) \qquad (2)$$

$$q_{loss} = \lambda_m (T_{wi} - T_{wo}) / \delta \qquad (3)$$

Equations (1), (2), and (3) consist of the heat transfer coefficient h, the equivalent hydraulic diameter d_e, the thermal conductivity of test fluid λ_f, the measured net heat flux q, heat loss by heat conduction through the wall to backside q_{loss}, the local wall temperature at the position of thin stainless steel foil T_{wi}, the wall temperature at the outside of the channel T_{wo}, the mainstream temperature T_f, the thermal conductivity of the channel material λ_m and the wall thickness of the channel δ. The mainstream temperature T_f is determined from the measured fluid temperatures at the starting and end point of the heated range. The typical measured wall and fluid temperatures are 40~70°C and 20~40°C, respectively. The uncertainty of this data reduction mainly depends on the measurement of the temperature and heat flux. The estimate of the error in calculating heat transfer is at most 10 %.

In a fully developed channel flow, the friction factor f can be determined by measuring the pressure drop across the flow channel Δp, the main stream velocity calculated from the mainstream veocity v, the density of test fluid ρ, the equivalent hydraulic diameter d_e, and the distance L between the pressure measuring points at inlet and outlet of the channel. These data are applied to the following equation.

$$f = \Delta p / (4L / d_e) / (1 / 2\rho v^2) \qquad (4)$$

Tests have been conducted at Reynolds number, based on the hydraulic diameter, from 8,000 to 78,000, changing the mass flow rate. Reynolds number range covers typical conditions of air cooled turbine blades.

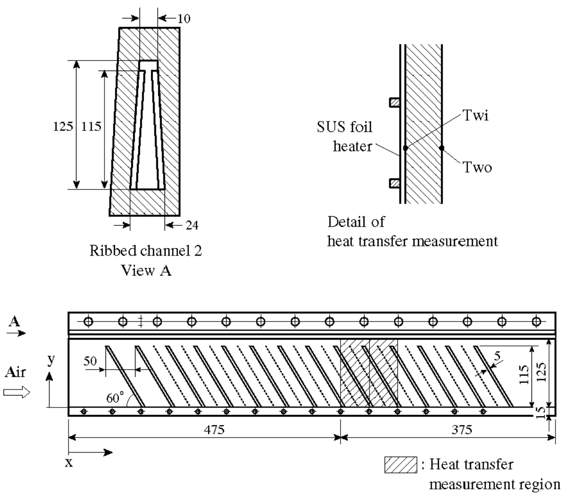

Fig.2 Schematic of test section

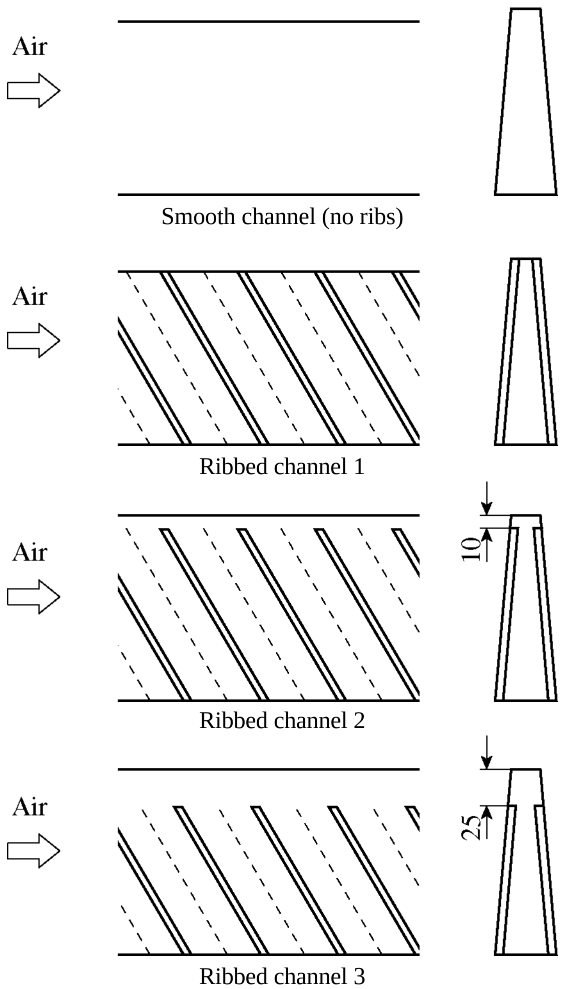

Fig.3 Test channel

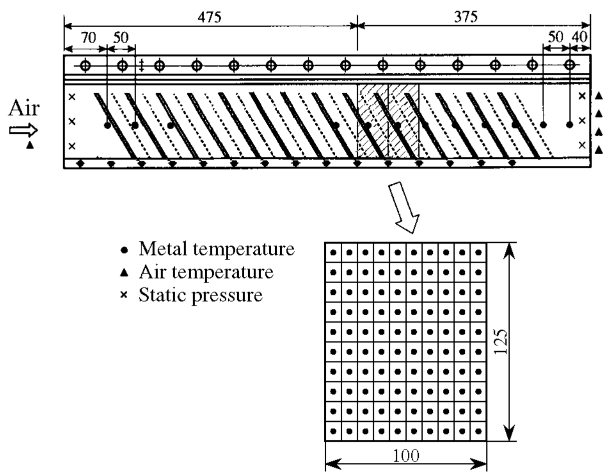

Fig.4 The detail of measuring location

Experimental Results and Discussion

The streamwise distributions of Nusselt number for the smooth channel are shown in Fig.5. The streamwise variation indicates the flow developing in the entrance region of the channel. The Nusselt number decreases with increasing downstream length. It reaches constant value when the flow is fully developed. Intensive measurement region with 100 embedded thermocouples is exposed to this fully developed flow.

Fig.6 and 7 show the variation of the mean Nusselt number, which is the arithmetic average of local Nusselt number at the intensive measurement region, and the friction factor with Reynolds number. The results calculated by Blasius, Kay and Han's Correlation (1984) are presented for comparison.

The mean Nusselt number and the friction factor measured by using the smooth channel model agree very well with Kay and Blasius's correlation. The validity of the measurement with this test section is confirmed.

The mean Nusselt number of the ribbed channel 1 (without gap) is approximately 20% lower than that of Han's correlation. The friction factor of the ribbed channel 1 is, nevertheless, approximately 60% higher than that of Han's correlation. These facts are attributed to the effect of the geometry and aspect ratio in the cross-section of the channel. Friction factor of all ribbed channels increases with an increase of Reynolds Number, although that of Han's correlation is constant. This result, which is similar to that of Kiml et al. (2000), is due to staggered array of ribs on the left and right surface in a thin channel. Concerning the effect of the gap, the smaller the length of the gap, the larger the mean Nusselt number and the friction factor.

Experimental Nusselt number contours of the smooth channel and all ribbed channel are shown in Fig.8. The distribution of mass flow rate in the channel affects the heat transfer. The velocity in the thick region may be large, compared with that in the thin region. The contours with all ribbed channels show the heat transfer enhancement by re-attached flow and swirl flow. In addition to them, the heat transfer enhancement is indicated around the downstream of rib edges in the ribbed channel 2 and 3 (with gap). Therefore, the ribbed channel 2 with small gap is effective for cooling the trailing edge. Fig.9 shows the variation of the local Nusselt number around the thin region with Reynolds number. The local Nusselt number of the ribbed channel 1 and 2 are approximately same and the highest compared to all. In particular, the local Nusselt number of the ribbed channel 2 is slightly higher than that of the ribbed channel 1 under comparatively high Reynolds number condition.

Eventually, optimizing the length of the gap enables to enhance heat transfer around the trailing edge and to prevent the large reduction of the average heat transfer in the whole of the channel. The optimum gap size must be between 0mm and 10mm in this experiment.

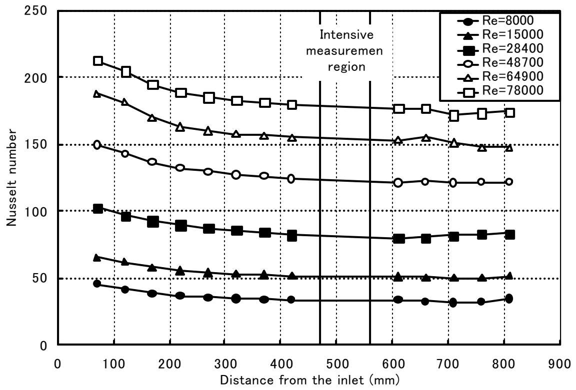

Fig.5 Streamwise distributions of Nusselt number for the smooth channel

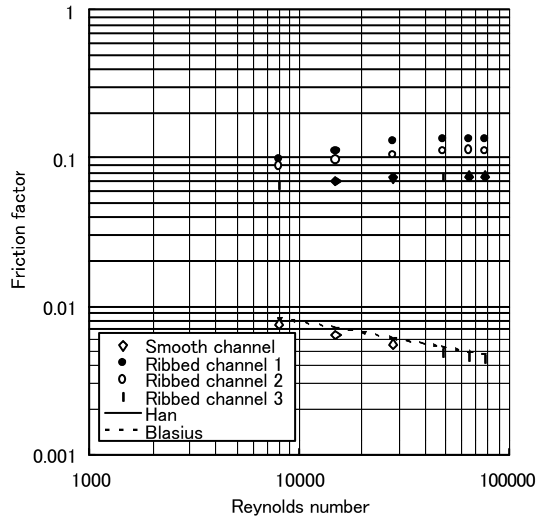

Fig.6 Friction factor versus Reynolds number

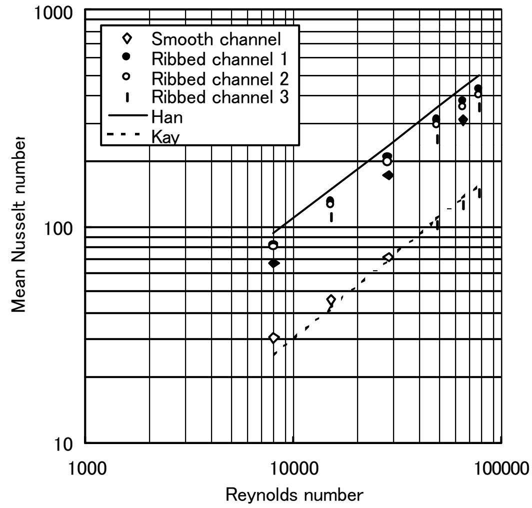

Fig.7 Mean Nusselt number versus Reynolds number

NUMERICAL ANALYSIS

Numerical Process

Three kinds of ribbed channels stated in the previous section are analytical system. The computational domain, as shown in Fig.10, is only one rib-module based with rib position in streamwise direction of the test channel. Fig.11 shows the example of analytical grid with approximately 200,000 cells. Periodic boundary condition is imposed along the streamwise direction. The similar analytical method is adopted in the previous studies (Braun et al., 1999 Ciofalo et al., 1992 Hermanson et al., 1989). Three-dimensional incompressible analysis is performed with Fluent5.0. Eddy-viscosity turbulence model according to Realizable k-ε model (Shih et al., 1995) is used. The analyses have been conducted at Reynolds number, from 28,000 to 65,000, based on hydraulic diameter. The uniform heat flux condition is applied in the heated wall except ribs.

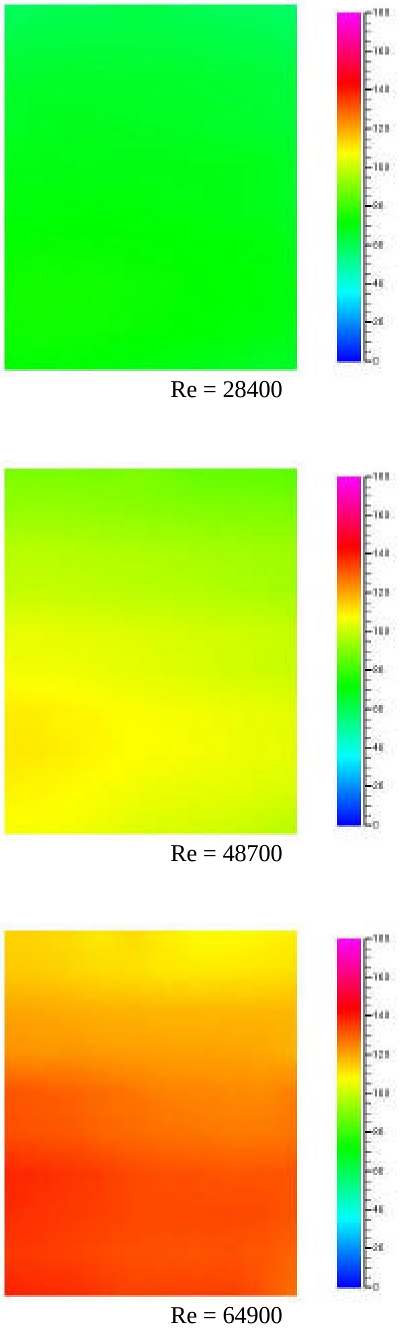

Fig.8a Experimental Nusselt number contour (Smooth channel)

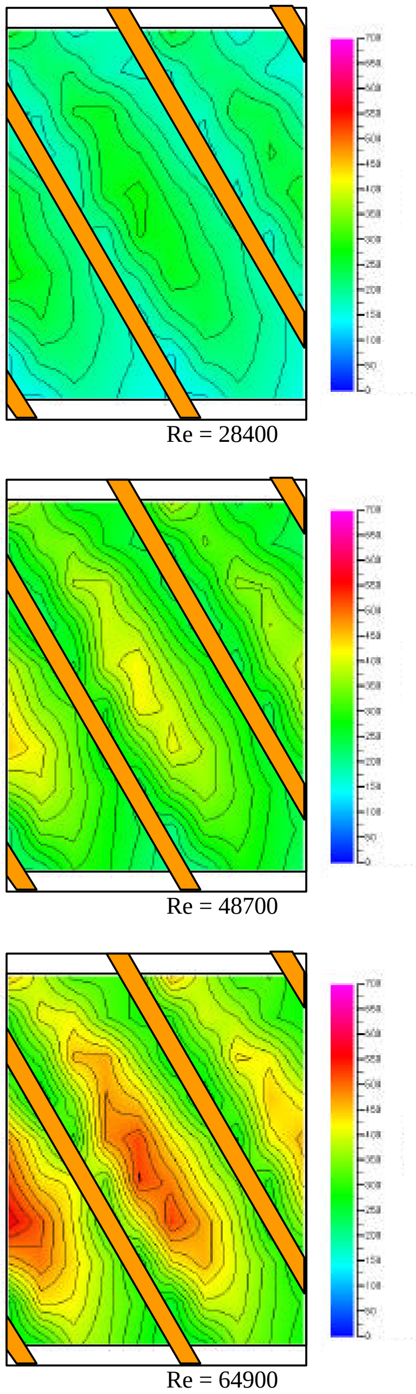

Fig.8b Experimental Nusselt number contour (Ribbed channel 1)

Comparison with experimental data and discussion

Analytical Nusselt contours of ribbed channels are presented in Fig.12. All contours agree with experimental contours shown in Fig.12 qualitatively.

In the ribbed channel 1, the maximum region is moved to the wide region of the channel with the increase of Reynolds number. This tendency is also somewhat indicated in the experimental contour shown in Fig.8. This is caused by the distribution of mass flow rate under each Reynolds number condition. The velocity at the wide region is larger than that at the narrow region. The difference between the two kinds of velocities increases with the increase of Reynolds number.

The heat transfer enhancement around the downstream of the rib edges in ribbed channels with the gap is indicated remarkably. The velocity contours at the cross section of channels and the velocity vectors at the ribbed surface are shown in Fig.13 and 14 respectively. The ribbed channel 1 (without gap) has the high velocity in the comparatively wide region of the channel. However, the ribbed channel 2 and 3 (with gap) have the highest velocity in the narrow region of the channel. Namely, the powerful secondary flow from the gap exists in the ribbed channel with the gap in addition to re-attachment flow and swirl flow.

The heat transfer enhancement around the downstream of the rib edges shown in Fig. 8 and 12 is attributed to the secondary flow from the gap, which is similar to the result of Kiml et al. (2001). The secondary flow from the gap between the narrow side-wall and the ribs contributes to the enhancement of local heat transfer around the trailing edge in a ribbed channel.

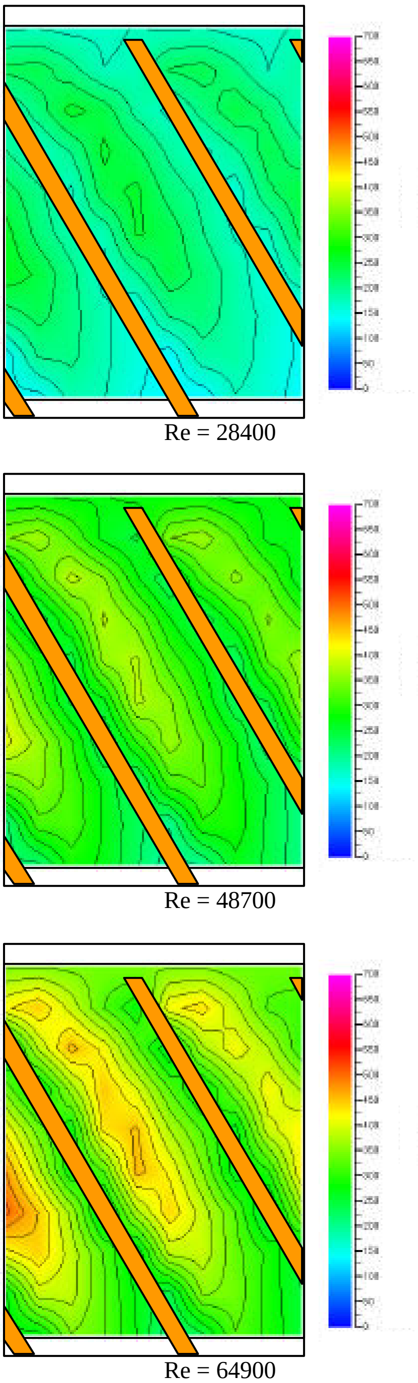

Fig.8c Experimental Nusselt number contour (Ribbed channel 2)

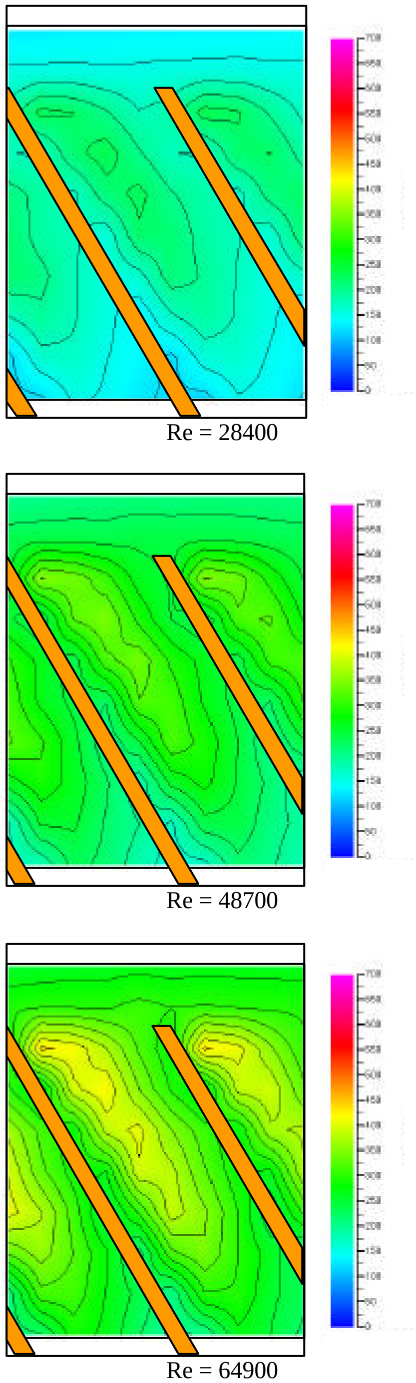

Fig.8d Experimental Nusselt number contour (Ribbed channel 3)

Zhengjun et al. (1996) reported the flow structure in the ribbed channel with the gap. It includes the above-mentioned secondary flows and the longitudinal vortex flow which generates from the top surface of the ribs. In general, the longitudinal vortex flow somewhat, has contributed to the heat transfer enhancement in the whole of the ribbed flow channel with the gap, too.

Finally, variations of mean Nusselt number and friction factor versus Reynolds number are shown in Fig.15 and 16. The accuracy of numerical analysis relating heat transfer and friction factor are ±20% and ±10%, respectively. This analytical method is capable of predicting the heat transfer and the pressure loss in a triangular ribbed channel. We have a confidence to apply CFD as the tool for the design of a enhanced cooling flow channel and to optimize the cooling structure.

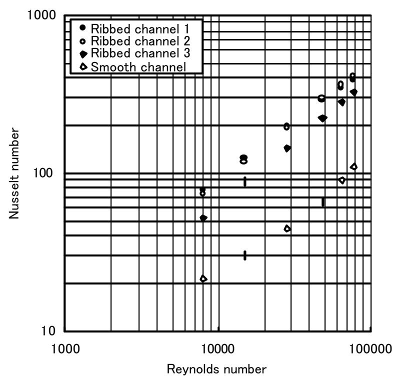

Fig.9 Local Nusselt number around the thin region of channels versus Reynolds number

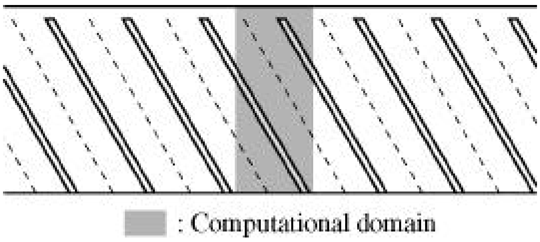

Fig.10 Computational domain (Ribbed channel 2)

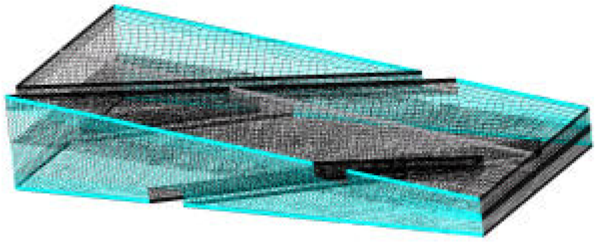

Fig.11 Analytical grid (Ribbed channel 2)

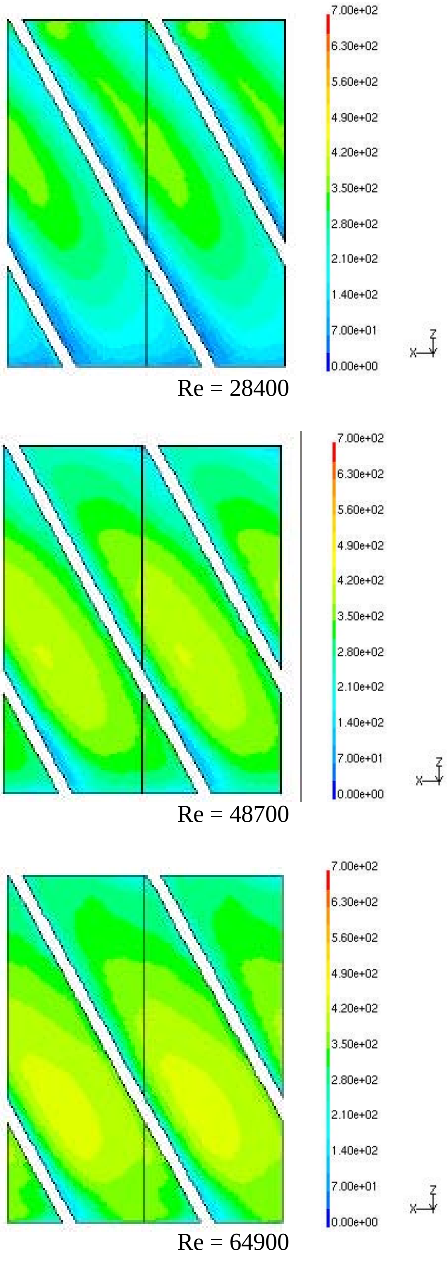

Fig.12a Analytical Nusselt number contour (Ribbed channel 1)

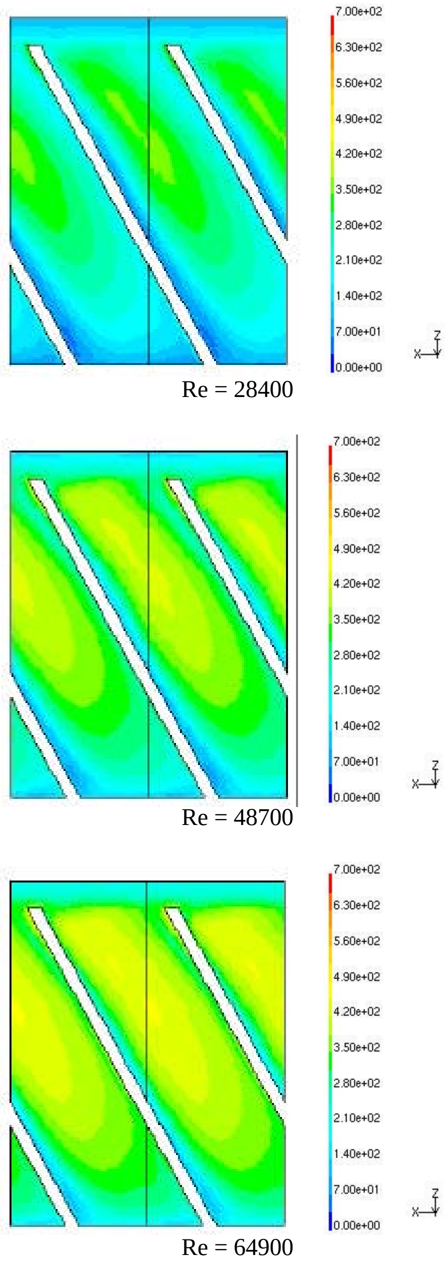

Fig.12b Analytical Nusselt number contour (Ribbed channel 2)

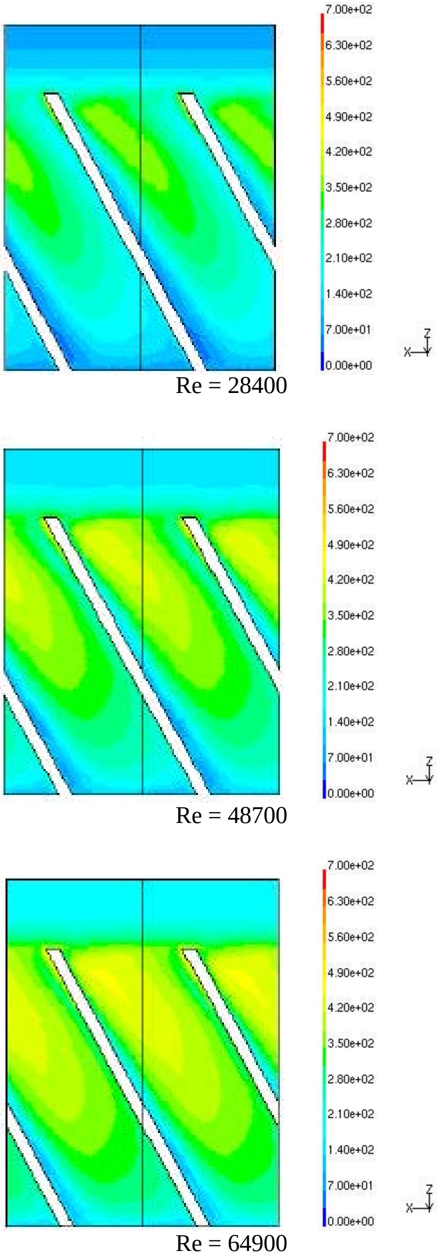

Fig.12c Analytical Nusselt number contour (Ribbed channel 3)

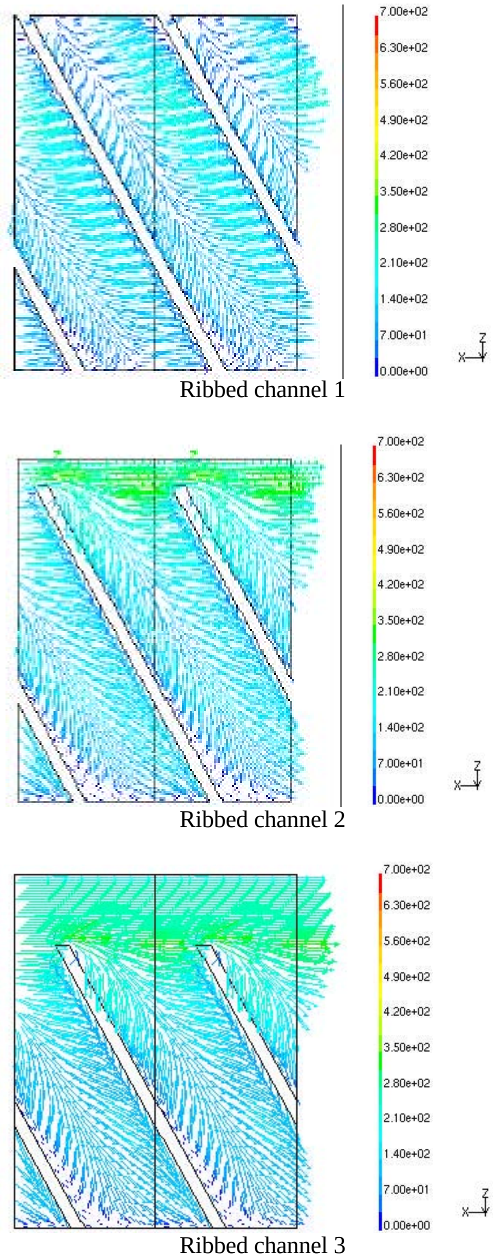

Fig.14 Velocity vector at the ribbed surface (Re=64900)

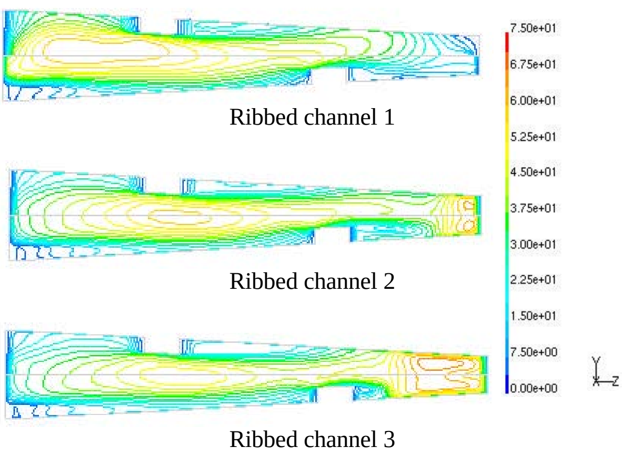

Fig.13 Velocity contour at the cross section of the channel (Re=64900)

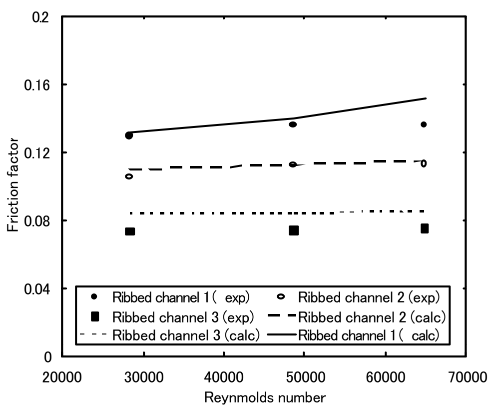

Fig.15 Friction factor versus Reynolds number

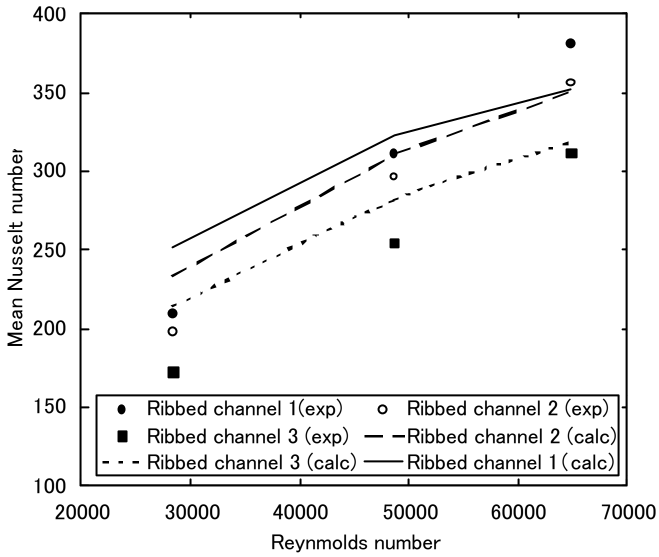

Fig.16 Mean Nusselt number versus Reynolds number

CONCLUSIONS

The following conclusions were obtained through the experimental study and prediction by CFD on the heat transfer and pressure loss of a triangular flow channel.

1) The secondary flow from the gap between narrow side-wall and ribs contributes to the control of local heat transfer around trailing edge in a ribbed channel. In particular, it enables to enhance heat transfer under comparatively high Reynolds number condition.

2) Optimizing the length of the gap enables the enhancement of the heat transfer around the trailing edge and to prevent the large reduction of the average heat transfer in the whole of the cooling flow passage.

3) CFD is capable of predicting the heat transfer and the pressure loss in a ribbed channel and can be the tool for fundamental design of a cooling channel with ribs.

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