GT2011-453) *

HEAT TRANSFER ENHANCEMENT DUE TO COMBINATION OF DIMPLES, PROTRUSIONS, AND RIBS IN NARROW INTERNAL PASSAGE OF GAS TURBINE BLADE

Akira Murata

Tokyo University of Agriculture and Technology Koganei, Tokyo, Japan

Kaoru Iwamoto Tokyo University of Agriculture and Technology Koganei, Tokyo, Japan Satomi Nishida Tokyo University of Agriculture and Technology Koganei, Tokyo, Japan

Yoji Okita IHI Corporation Nishitama, Tokyo, Japan Hiroshi Saito Tokyo University of Agriculture and Technology Koganei, Tokyo, Japan

Chiyuki Nakamata IHI Corporation Nishitama, Tokyo, Japan

ABSTRACT

Internal convective cooling of gas-turbine airfoil is essential because turbine inlet temperature becomes higher for pursuing higher thermal efficiency. For higher cooling performance, heat transfer is often enhanced by installing ribs and/or pin-fins in the internal passage. In this study, in order to enhance heat transfer, the combination of spherical dimples, cylindrical protrusions, and transverse square ribs was applied to one wall of a narrow passage. As for the cylindrical protrusions, two different diameter cases were examined. The heat transfer enhancement was measured by a transient infrared thermography method for the Reynolds number of 2,000, 6,000, and 10,000. The pressure loss was also measured in the experiments, and RANS simulation was performed to give a rationale for the experimental results. The present results clearly showed the spatial variation of the local Nusselt number: the high Nusselt number was observed on the rib top-surface and also near the leading edge on the protrusion top-surface. In addition, the areas around the dimple's trailing-edge on the oblique line connecting the neighbor dimples showed moderately enhanced heat transfer. When two different protrusiondiameter cases were compared, both the mean Nusselt number and the friction factor were similarly higher in the larger protrusion case than the smaller protrusion case, and therefore the larger protrusion case was more

effective in cooling even when the pressure loss was taken into account.

NOMENCLATURE

а	:	thermal diffusivity of wall [m ² /s]						
d_{dmp}	<i>i</i> :	dimple diameter [m]						
d_{prot}	:	protrusion diameter [m]						
F	:	function used in transient technique [-]						
h	:	heat transfer coefficient [W/(m ² •K)]						
H	:	height of channel [m]						
Hara	.:	height of protrusion [m]						
Hrib		height of rib [m]						
L	:	distance between pressure taps [m]						
D_{r}	:	streamwise	pitch	of	roughness			
elemer	nts[m]				J			
p_{τ}	: '	lateral pitch of	f dimples a	nd prot	rusions [m]			
ΛP	:	pressure diffe	rence [Pa]	•				
<i>a</i>		wall heat flux	ſW/m²l					
Re		Revnolds number. $Re = U 2H/v$						
+		time [e]		m=11 / /				
ι	:	tomporaturo [Z 1					
	•		nj - [
U_m		mean velocity	[m/s]	_				
x	:	streamwise co	pordinate [r	nj				
Z	:	lateral coordir	nate [m]					
δ	:	dimple depth	[m]					
\varDelta_{th}	:	temperature p	enetration	depth [m]			
		•			-			

 λ : thermal conductivity [W/(m•K)] ν : viscosity [m²/s]

Subscripts

f:	fluid				
<i>i</i> :	initial condition				
smooth:	smooth wall case				
w :	wall				

INTRODUCTION

High turbine inlet temperatures are desired to achieve better gas turbine performance. Current turbine inlet temperatures far exceed the ability of metal alloys or ceramics to perform over required service lives. The turbine components are maintained hundreds of degrees below the hot gas temperatures by the use of intensive cooling. The vast majority of turbine cooling is accomplished with air drawn from the compressor. While this air is readily available, its use conflicts additional penalties on gas turbine performance due to the power required for compression and also usually causes aerodynamic losses when the coolant re-enters the hot gas stream in the turbine.

The cooling schemes in turbine components have been exhaustively investigated in the past several decades to assure more efficient ways of providing cooling while minimizing the amount of coolant. Nowadays, a variety of techniques are used for enhancing convective heat transfer rates in aeroengines/gas-turbines for internal cooling of turbine airfoils. These include rib turbulators, pin-fins, dimpled surfaces, surfaces with arrays of protrusions, or similar repeated features. All of these devices act to generate fluid turbulence, vortices, and/or even larger scale secondary flows to enhance mixing. Such vortices and secondary flows not only act to increase advection of heat, but also to increase turbulence production by increasing shear stress. These give larger magnitudes of turbulent transport. All of the aforementioned enhancing devices also provide some heat transfer augmentation by increasing surface areas. The overall objective of each device is then significant enhancement of turbulent heat transfer, with minimal increases in streamwise pressure drop penalties. These devices are utilized in separate regions of the airfoils where they are best suited due to geometry constraints, pressure losses, and other reasons.

Pin-fins are generally used in the parts of turbine airfoils where higher levels of heat transfer augmentation are required and where pressure drops are tolerated to some extent or often even desired (such as in trailing edge regions). In the past numerous researches, considered are the influences of pin geometry, materials, array configuration, channel geometry, Reynolds number, etc. Of the earlier investigations, Sparrow et al. [1], Metzger et al. [2], and Simoneau and Van Fossen [3] reported Nusselt numbers and indicated the presence of large overall heat transfer augmentations for certain flow conditions and pin-fin array arrangements. The effects of different pin-fin shapes on the heat transfer were observed by reviewing different pin-shape results of circular pins [4], elliptic pins [5], diamond pins [6], cubic pins [7], threedimensional protrusions [8], and pins with endwall fillets [9].

Rib turbulators are generally rectangular crosssectional bars mounted along the surface, which are often angled with respect to the bulk flow direction. Because they protrude into the flow, they act to trip the flow, mix the flow, and also generate vortices and three-dimensional velocity gradients. When the main geometric parameters (rib height, channel blockage, orientation, and spacing) are varied, it is possible to optimize the cooling scheme. A significant number of experimental and numerical studies address the effects of rib turbulators on heat transfer. Among several of the earliest studies, Han et al. [10] addressed the effects of rib shape, angle of attack, and pitch to height ratio. In later study, Han et al. [11] also investigated V-shaped ribs with 45- and 60-deg arrangements in square channels and indicated that the best heat transfer enhancements are produced by this configuration. Taslim et al. [12] studied twelve different geometries of ribs that are placed on all four walls of channels with both square and trapezoidal cross sections. Compared to channels with ribs on two walls, heat transfer coefficients and thermal performance factors are enhanced. Cho et al. [13] employed continuous and discrete, parallel and cross arrays of ribs in a square duct. Discrete ribs with gaps in between are found to produce more uniform heat transfer coefficient distributions than continuous ribs. Also numerous computational studies of flow and heat transfer with rib turbulators considered orthogonal ribs [14][15], angled ribs [16]-[18], 45-deg Vshaped ribs [19], etc.

Dimples are arrays of indentations along surfaces. These are often spherical in shape. Surface with dimples is an attractive method for internal cooling because they produce multiple vortex pairs that augment heat transfer as they advect downstream. They are notable for the lowpressure drop penalties that they produce, which is because they do not protrude into the flow to cause form drag. The history of published research of dimples in context with gas turbine applications is relatively new. Schukin et al. [20] presented heat transfer results with a single hemispherical cavity in a diffuser channel and in a convergent channel. They concluded that heat transfer is augmented with either positive or negative pressure gradients. Chyu et al. [21] presented data showing heat transfer on surfaces imprinted with staggered arrays of two different shapes of concavities. Compared with the conventional spherical dimple, the teardrop dimple generally induces somewhat higher surface heat transfer. Gortyshov et al. [22] employed spherical dimples placed at different relative positions on the opposite sides of a narrow

channel. They suggested that the flow structures produced by each side interact with each other in a nonlinear fashion, with behavior that cannot be determined using superposition. Mahmood et al. [23] considered heat transfer in a channel with dimples and protrusions on opposite walls. As a result, thermal performance parameters are generally slightly lower when both protrusions and dimples are employed, compared to a smooth-dimple arrangement. More recently, Borisov, et al. [24] investigated an alternative geometry, banked dimple, where dimple and toroidal bank are concentrically arranged. As expected, the surface with banked dimples produces higher heat transfer at the cost of higher friction losses.

Though there have been numerous works done with these three key features as summarized above, seldom are researches combined together in an attempt to further improve cooling effectiveness through synergistic effects[25]. One of such a few pioneering works investigating the interaction of these augmentation features was done by Bunker et al. [26]. They evaluated channel averaged heat transfer and pressure loss of complete "double-walled" arrangement with various configurations using pin-fins, rib-turbulators, and concavities and showed capabilities more than three times the heat transfer of smooth channels. Though the heat and fluid flow mechanism was not clarified, this research well demonstrated the promising potential of the combined effect of the different surface devices. The present authors [27] also investigated the combination effects of the surface devices (dimples, ribs, and protrusions) by performing a large eddy simulation. For the geometries in that study, the highest heat transfer enhancement was obtained for the combination of dimples and ribs though the pressure loss penalty was three times larger for 20% heat transfer increase as compared to the passage with only dimples.

The present paper is directed at combined cooling techniques designed for significant heat transfer enhancement with the three key augmentation devices, (cylindrical protrusions), transverse pin-fins rib turbulators, and spherical dimples. The primary contribution of the paper is to stress not only on the average heat transfer capabilities but more on the detailed local heat transfer distributions aiming to better understand the complex physics. A transient infrared thermography method is adopted for this purpose. Also steady three-dimensional RANS simulation is performed to give insight into the fluid dynamics and heat transfer mechanism of the synergistic interaction of the devices. In the current study, the two opposite walls are not joined via pin-fins (i.e. thermally unchained) and only the one side of the walls is roughened. Thus, the arrangement is fundamentally different from the configuration of Bunker et al. [26] where the complete double-walled airfoil is assumed. The choice of the present configuration is



Fig.1 Narrow internal passages of gas turbine vane.

motivated by intention of applying this novel cooling technology directly to the existing cooled vanes/blades in affordable manner without drastic change in whole airfoil design (see Fig.1).

2. EXPERIMENTS

2.1. Transient Technique

When a semi-infinite wall with initial temperature of T_i is exposed to flow of which temperature is changed stepwise from T_i to T_f at time, t = 0s, surface temperature of the wall, T_w , at time, t, is expressed by the following theoretical solution of one-dimensional heat conduction problem of the semi-infinite plate [28]:

$$\frac{T_w - T_i}{T_f - T_i} = 1 - \exp\left(\frac{h^2 at}{\lambda_w^2}\right) \operatorname{erfc}\left(\frac{h\sqrt{at}}{\lambda_w}\right), \tag{1}$$

where *h*: heat transfer coefficient, *a*: thermal diffusivity of wall, and λ_w : thermal conductivity of wall. When the fluid temperature changes not stepwise but gradually, Duhamel's theorem [28] is applied and the fluid temperature change is approximated by *N* small temperature changes whose contributions are superimposed to give the solution. In this case, the temporal variation of wall temperature becomes as follows [29]:

$$T_{w} - T_{i} = \sum_{j=1}^{N} F(t - t_{j}) \left(T_{f,j} - T_{f,j-1} \right) \quad , \tag{2}$$

where

$$F(t-t_j) = 1 - \exp\left(\frac{h^2 a(t-t_j)}{\lambda_w^2}\right) \operatorname{erfc}\left(\frac{h\sqrt{a(t-t_j)}}{\lambda_w}\right).$$
(3)

In equation (2), $T_{j,j}$ is fluid temperature at time, t_j (*j*=0,1,2,..., *N*). Note that heat transfer coefficient, *h*, cannot be explicitly solved from equations (2) and (3), and therefore *h* has to be solved by an iterative method.

As equations (1)-(3) show theoretical surface temperature of a semi-infinite plate, the experimental condition has to satisfy this "semi-infinite" assumption. When the surface temperature is changed, the depth to which the temperature change penetrates is called temperature penetration depth, Δ_{th} , and it is approximated as equation (4):

$$\Delta_{th} = \sqrt{12at} \ . \tag{4}$$

If Δ_{th} becomes sufficiently small as compared to the plate thickness, the semi-infinite assumption holds. In this study, the penetration depth becomes 6.25mm after 30-second heating, and from this result the thickness of 20mm is adopted for the acrylic plate.

In addition, when the surface is not flat, multidimensional heat conduction in the wall affects the result. The authors (Nishida et al.[30]) have examined this effect for dimpled surface, and confirmed that the highly affected area is confined to the region close to the dimple rim, and its effect on mean Nusselt number is small (3% in [30]). However, in the present case, the protrusions and ribs further introduce the multi-dimensional heat conduction, and those effects are not quantitatively identified yet.

2.2. Experimental apparatus and method

In this study, the same apparatus and procedure were used as those in Nishida et al. [31]. Schematic of experimental apparatus is shown in Fig.2, and details of test section are shown in Fig.3. Air was used as a working fluid, and it was sucked by a turbo blower. At the inlet of the test section, mesh-type heater (SUS316, 250 mesh/inch, ϕ 30µm) was installed and used to heat the air by applying electric current. The heated air flowed into acrylic test section of which width was 21.6H (H=9.25mm is the channel height) where only one surface was roughened with dimples, protrusions, and ribs, and the opposing surface was smooth. The total length of the test section was 58.4H. Inlet and outlet air temperatures were measured by using platinum-wire sensor which can measure the cross-sectionally averaged temperature from electric resistance change of platinum wires (650µm) covering the channel crosssection.

On the opposing smooth surface at 45.4*H* downstream from the inlet, a view window of germanium



Fig.2 Schematic of experimental apparatus.



Fig.3 Details of test section.

glass (ϕ 75mm) with anti-reflection coating was installed. Through the window, an infrared camera (FLIR SC-620, 640x480pixel²) measured unsteady wall-temperature change. The roughened surface was painted in black in order for the surface to have higher emissivity. The surface emissivity was determined such that the surface temperatures measured by a thermocouple (ϕ 50µm) buried in very surface of the wall and by the infrared camera agreed to each other.

As shown in Fig.4, the spherical dimples had the diameter of $d_{dmpl}/H=2.38$, the depth of $\delta/d_{dmpl}=0.30$. The square ribs had common height and width of $H_{rib}/H=0.31$ and streamwise pitch of $p_x/H_{rib}=10.35$. As for the cylindrical protrusions, two different diameter cases were examined: the diameters of $d_{prol}/H=2.54$ and 1.70 with the common height of $H_{prol}/H=0.857$. The dimples and protrusions were arranged in the staggered manner. In the lateral direction, the dimples and protrusions were linked by the transverse ribs. Therefore, the dimples and protrusions had common streamwise and lateral pitches of $p_x=p_z$ ($p_x/d_{mpl}=1.34$ or $p_x/d_{prol}=1.25/1.87$ for two d_{prot} cases). The Reynolds number defined by the mean velocity and two times of



(c) Detailed dimension of roughness elements for case of d_{prot}/H =1.70.

Fig.4 Details of heat transfer surfaces.

channel-height (2H) was varied among 2,000, 6,000, and 10,000.

As emissive intensity of infrared ray changes directionally, temperature measurement by the infrared camera is affected by the surface inclination with respect to the camera optical axis. The dimpled surface of this study made an angle of 60deg at the dimple edge with respect to the flat part. In addition, the camera optical axis was angled by 10deg from the perpendicular direction to the view window; this angle (10deg) was needed for avoiding the reflection from the camera itself on the view window. Therefore, the smallest camera optical angle with respect to the surface was 20deg (perpendicular case corresponds 90deg). In order to this angle effect on the examine temperature measurement, preliminary experiments were performed by measuring a heater's surface from various angles and the infrared camera results were compared with The results showed that the thermocouple output.



Fig.5 Computational domain of numerical analysis.

infrared camera indicated about 0.1°C lower than the thermocouple output in the angle of 30-50deg with respect to the measured surface. This factor was considered in the measurement uncertainty analysis [32]. It should be noted that the area giving the camera optical angle with respect to the surface less than 30deg is only 8.9% of the dimple projection area.

In this study, the temperature of main flow was heated up to about 90°C. In order to avoid the period of small temperature increase which leads to larger measurement uncertainty, the time, t=30s, was chosen as a measurement time. The estimated measurement uncertainty was 7% for both Nu and f for Re=10,000 [32]. Here, the effect of multi-dimensional heat conduction in the wall explained in section 2.1 is not included.

3. NUMERICAL ANALYSIS

Three-dimensional steady RANS simulation was performed for the geometry identical to the experimental test section. FLUENT 6.3 was used, and the unstructured grid using hexahedral elements was generated by using GAMBIT 2.3. Heat and fluid flow was simulated as a steady problem using a realizable k- ε turbulence model with enhanced wall treatment as a wall function. Considering geometric symmetry and periodicity of the test section, computational domain extended for lateral one pitch and for streamwise two pitches as shown in Fig.5. The computational nodes were located more densely near the wall boundaries, and total 1.2million nodes were allocated in the computational domain. When the grid resolution was doubled near the wall boundaries in which the node number was 1.7 million, the differences of mean Nusselt number and friction factor due to this resolution change were within 3.3% and 1.2%, respectively. As for the boundary conditions, constant heat flux was assigned on the walls. The other boundary conditions were symmetric at the lateral boundaries, and periodic at the streamwise boundaries. When temperature was assigned at the inlet,



Fig.6 Contour of local Nusselt number: left and right figures are for d_{prot}/H =2.54 and 1.70, respectively (experimental results).



Fig.7 Contour of local Nusselt number (numerical results for Re = 10,000).

the streamwise periodicity was attained by subtracting the bulk-temperature increase from the outlet temperature.

4. RESULTS AND DISCUSSION

The measured heat transfer coefficient, h, is shown as the Nusselt number,



Fig.8 Velocity vectors and fluid temperature of numerical results at distance of $0.1H_{rib}$ from roughened surface (*Re*=10,000).

$$Nu = \frac{h2H}{\lambda_f} \tag{5}$$

and it is normalized by the empirical correlation for smooth pipe (Kays and Crawford [33]):

$$Nu_{\infty} = 0.022Re^{0.8} Pr^{0.5}.$$
 (6)

Although Re=2,000 is usually treated as laminar flow, the turbulent correlation of equation (6) was used even for Re=2,000 considering the highly disturbed flow due to the roughness elements on the channel surface. For reference, it should be noted that Nu_{∞} of laminar and turbulent (equation (6)) flows for Re=2,000 are 8.23 and 8.00, respectively. In this study, considering the periodicity and symmetry of the flow field, the minimum unit area becomes the rectangle shown by broken lines in Fig.4(c). The presentation and averaging of the results are performed for this minimum unit area.

Figure 6 shows the contour of local Nusselt number. Though the color range differs for Re=2,000, the following tendencies are common for all Re and two protrusion diameters. Nu is the highest near the leading edge on the protrusion top-surface and also on the rib top-surface. In addition, the moderate increase of Nu can be seen around the dimple's trailing-edge on the oblique line connecting the neighbor dimples. The convection in the oblique direction must be dominating due to the flow blockage by the protrusions. When the effect of the protrusion diameter is



Fig.9 Streamwise variation of local Nusselt number for *Re* = 10,000 (experimental results).

examined by comparing the left and right figures, the peak values of Nu on the protrusion and rib top-surfaces are higher for the larger protrusion diameter case $(d_{prot}/H=2.54)$. In the case of smaller protrusion diameter (right figures, $d_{prot}/H=1.70$), the isolated high Nu area behind the transverse ribs is observed which must be caused by the flow reattachment of the flow separation caused by the ribs. In the case of $d_{prot}/H=2.54$ with laterally shorter ribs, the corresponding area may be merged with high Nu area around the dimple's trailingedge on the oblique line. On the other hand, the low Nu areas are observed near trailing edge on the protrusion top-surface, behind the protrusions, just behind the ribs, and near leading edge within the dimple on the oblique line connecting the neighbor dimples. These low Nu areas are corresponding to the wake and flow separation areas. Figure 7 shows the local Nusselt number of the numerical result for Re=10,000. Almost all the



Fig.10 Mean Nusselt number and friction factor versus Reynolds number (open symbols are for numerical results, and plot of Bunker et al.[26] is for their "small diamond with dimpulators").

experimentally observed tendencies are seen in the numerical result, although the enhancement on the dimple's trailing-edge on the oblique line seems higher in the numerical analysis. Figure 8 shows the velocity vectors and fluid temperature of the numerical results at a distance of $0.1H_{rib}$ from the roughened wall. In the figure, the linearly increasing component was subtracted from the fluid temperature, and the streamwise two-span was shown for easier understanding the flow field. The fluid temperature in Fig.8 is scaled by the temperature scale of $q_w H / \lambda_f$. As expected, the main flow is meandering due to the blockage of the protrusion and the high Nu on the dimple's trailingedge on the oblique line is corresponding to the high velocity and low fluid temperature region of this meandering main flow. This meandering main flow makes high temperature (and low heat transfer) area behind the

	(A) <i>d_{prot}/H</i> =2.54			(B) <i>d_{prot}/H</i> =1.70		
Re	2000	6000	10000	2000	6000	10000
Nu _m	18.4	80.0	130	13.1	58.1	108
Nu_m / Nu_∞	2.25	4.23	4.45	1.70	3.09	3.79
$Nu_{m,A}/Nu_{m,B}$	1.40	1.38	1.20	-	-	-
f	0.358	0.294	0.282	0.272	0.233	0.219
f/f_∞	30.3	32.7	35.6	22.9	25.9	27.6
$f_{\rm A}/f_{\rm B}$	1.32	1.26	1.29	-	-	-
$\eta_{e\!f\!f}$	0.788	1.44	1.48	0.653	1.14	1.37

Table 1 Mean Nusselt number and friction factor for each Reynolds number.

protrusion smaller. When two protrusion-diameter cases are compared in Figs.6 and 7, the larger diameter case shows smaller area of low Nusselt number.

Figure 9 shows the streamwise variation of local Nu on the three center-lines of protrusion, rib, and dimple for Re=10,000. In the bottom part of the figure, the shapes of protrusion, rib, and dimple are drawn for easier identification of the location. The above-mentioned tendencies for Fig.6 are more clearly seen in Fig.9: the highest Nu near the leading edge on the protrusion top-surface and the second highest Nu on the rib top-surface. The increase of Nu around the dimple's trailing-edge is less than those of protrusion and rib, though the centerline location differs from the location where locally high Nu is observed.

In Fig.10, the mean Nusselt number, Nu_m , and the friction factor are shown. In the present study, Nu_m was based on projected surface area. It should be noted that difference of Nu_m based on the actual/projected areas in the numerical results are only 0.3% by dimples' concavity effect and 3% by the rib/protrusion side-wall effect. The friction factor was calculated by the following equation:

$$f = \frac{\Delta P}{2\rho_f U_m^2 \frac{L}{2H}}$$
(7)

In the figure, *f* is normalized by the Blasius equation:

$$f_{\infty} = 0.0791 \ Re^{-0.25} \ . \tag{8}$$

Here again, the case of Re=2,000 is assumed to be turbulent, and it should be noted that the laminar $(f_{\infty}=16/Re)$ and turbulent (equation(8)) cases give $f_{\infty}=0.008$ and 0.012, respectively. Detailed values of Nu_m and f are shown in Table 1. In the figures, the experimental results of Bunker et al.[26] for their "small



Fig.11 Heat transfer efficiency index including surface area effect (legend is same as that in Fig.10).

diamond with dimpulators" are included. Their surface geometry can be summarized as follows (the numbers in the parentheses are for this study): $d_{dmpl}/H=3.0$ (2.38), $\delta/d_{dmpl} = 0.33$ (0.30), $H_{rib}/H=0.15$ (0.31), $d_{prot}/H=3.5$ (2.54/1.70), $p_x/d_{prot}=2.14$ (1.25/1.87). It should be noted that their roughness arrangement was different from that of this study and their test section had the roughness on both surfaces. Furthermore, in Fig.10, the original data of Bunker et al.[26] are divided by the factor of 1.052 which is the surface-area ratio as compared with the smooth surface, because they show hA ratio where A is the surface area. Within the present experimental range, the increase of Re gives higher Num. The agreement between the experimental and numerical results of Nu_m in Fig.10(a) is good: numerical results are lower than the experimental results by 6.4% and 6.6% for $d_{prot}/H=2.54$ and 1.70, respectively. The results of Bunker et al. [26] are lower than the present results except for Re=2,000. On the other hand, the friction factor in Fig.10(b) only gradually increases with the increase of Re. The numerical results of f are higher than the experimental results by 24.8% and 17.8% for $d_{prot}/H=2.54$ and 1.70, respectively. As seen in Fig.10 and Table 1, the case of $d_{prot}/H=2.54$ gives 20-40% larger Nu_m for the friction factor penalty of 26-32% as compared to that of dprot/H=1.70. The friction factor of Bunker et al. [26] is lower than the present results.

In order to examine the heat transfer efficiency taking the pressure loss into account, the heat transfer efficiency index, η_{eff} , was derived by Gee and Webb [34], and it means the index of the heat conductance for equal pumping power and heat transfer surface area. In this study, in order to compare among different heat transfer surface areas, *A*, the following modified definition was used by multiplying *A* to the Stanton number, *St*:

$$\eta_{eff} = \frac{\left(St \ A\right) / \left(St \ A\right)_{\text{smooth}}}{\left(f \ / \ f_{\text{smooth}}\right)^{1/3}}$$
(9)

In the above equation, smooth data were calculated by using equations (6) and (8). As the side surfaces of the ribs and protrusions are not measured in this study, those area are not counted in A. In the numerical results of Fig.11, for the comparison on the same basis, the side surfaces were excluded from A. As seen in Fig.11 and Table 1, the case of $d_{prot}/H=2.54$ gives higher value than that of $d_{prot}/H=1.70$. When the present results are compared with the result of Bunker et al.[26] the present results of $d_{nrot}/H=2.54$ have higher η_{eff} except for Re=2,000. For example, at Re=10000, the present result of $d_{nrot}/H=2.54$ is 30% higher than that of Bunker et al.[26]. The contribution from the side surfaces of ribs and protrusions is estimated by using the numerical results. The increase by including the side surface contribution becomes 31% and 24% for $d_{prot}/H=2.54$ and 1.70, respectively. Therefore, the present values in Fig.11 are considered to increase in the real performance.

5. CONCLUSIONS

In this study, in order to enhance heat transfer in a narrow passage, the combination of spherical dimples, cylindrical protrusions, and transverse square ribs was applied to one wall of the passage. As for the cylindrical protrusions, two different diameter cases were examined. The heat transfer enhancement was measured by a transient infrared thermography method for the Reynolds number of 2,000, 6,000, and 10,000. The pressure loss was also measured in the experiments. In addition, steady three-dimensional RANS simulation was performed for the identical geometry. The present results clearly showed the spatial variation of the local Nusselt number: the high Nusselt number was observed on the rib top-surface and also near the leading edge on the protrusion top-surface. The areas around the dimple's trailing-edge on the obligue line connecting the neighbor dimples showed moderately enhanced heat transfer. The main features of the experimental results were reproduced in the numerical results, though some discrepancy was seen in the absolute values of local Nusselt number and friction factor. When two different protrusion-diameter cases were compared, the mean Nusselt number was higher in the larger protrusion case by 20-40% than the smaller protrusion case for the friction loss penalty of 26-32%, and therefore the larger protrusion case was more effective in cooling even when the pressure loss was taken into account.

REFERENCES

[1] Sparrow, E. M., Ramsey, J. W., and Altemani, C. A. C., 1980, "Experiments on In-Line Pin Fin Arrays and Performance Comparisons with Staggered Array", *Journal of Heat Transfer*, Vol.102, No.1, pp.44-50.

- [2] Metzger, D. E., Barry, R. A., and Bronson, J. P., 1982, "Developing Heat Transfer in Rectangular Ducts with Staggered Arrays of Short Pin-Fins", *Journal of Heat Transfer*, Vol.104, No.4, pp.700-706.
- [3] Simoneau, R. J., and Van Fossen, G. J., 1984, "Effects of Location in an Array on Heat Transfer to a Short Cylinder in Cross Flow", *Journal of Heat Transfer*, Vol.106, No.1, pp.42-48.
- [4] Ligrani, P. M., and Mahmood, G. I., 2003, "Variable Property Nusselt Numbers in a Channel with Pin Fins", *Journal of Thermophysics and Heat Transfer*, Vol.17, No.1, pp.103-111.
- [5] Uzol, O., and Camci, C., 2001, "Elliptical Pin Fins as an Alternative to Circular Pin Fins for Gas Turbine Blade Cooling Applications. Part 1: Endwall Heat Transfer and Total Pressure Loss Characteristics", ASME Paper, 2001-GT-180.
- [6] Hwang, J. J., and Lu, C. C., 2000, "Lateral-Flow Effect on Endwall Heat Transfer and Pressure Drop in a Pin-Fin Trapezoidal Duct of Various Pin Shapes", ASME Paper, 2000-GT-232.
- [7] Chyu, M. K., Hsing, Y. C., and Natarajan, V., 1998, "Convective Heat Transfer of Cubic Fin Arrays in a Narrow Channel", *Journal of Turbomachinery*, Vol.120, No.2, pp.362-367.
- [8] Chyu, M. K., and Natarajan, V., 1996, "Heat Transfer on the Base Surface of Three-Dimensional Protruding Elements", *International Journal of Heat and Mass Transfer*, Vol.39, No.14, pp.2925-2935.
- [9] Chyu, M. K., 1990, "Heat Transfer and Pressure Drop for Short Pin-Fin Arrays with Pin-Endwall Fillet", *Journal of Heat Transfer*, Vol.112, No.4 pp.926-932.
- [10] Han, J. C., Glicksman, L. R., and Rohsenow, W. M., 1978, "An Investigation of Heat Transfer and Friction for Rib-Roughened Surfaces", *Journal of Heat and Mass Transfer*, Vol.21, No.7, pp.1143-1156.
- [11] Han, J. C., Zhan, Y. M., and Lee, C. P., 1991, "Augmented Heat Transfer in Square Channels with Parallel, Crossed, and V-Shaped Angled Ribs", *Journal* of *Heat Transfer*, Vol.113, No.3, pp.590-596.
- [12] Taslim, M. E., Li, T., and Spring, S. D., 1998, "Measurements of Heat Transfer Coefficients and Friction Factors in Passages Rib-Roughened on All Walls", *Journal of Turbomachinery*, Vol.120, No.3, pp.564-570.
- [13] Cho, H. H., Lee, S. Y., and Wu, S. J., 2001, "The Combined Effects of Rib Arrangements and Discrete Ribs on Local Heat/Mass Transfer in a Square Duct", *ASME Paper*, 2001-GT-175.
- [14] Rigby, D. L., 1998, "Prediction of Heat and Mass Transfer in a Rotating Ribbed Coolant Passage with a 180 Degree Turn", ASME Paper, 98-GT-329.
- [15] Murata, A. and Mochizuki, S., 2000, "Large Eddy Simulation with a Dynamic Subgrid-Scale Model of

Turbulent Heat Transfer in an Orthogonally Rotating Rectangular Duct with Transverse Rib", *International Journal of Heat and Mass Transfer*, Vol.43, No.7, pp.1243-1259.

- [16] Lin, Y.-L., Shih, T. I.-P., Stephens, M. A., and Chyu, M. K., 2001, "A Numerical Study of Flow and Heat Transfer in a Smooth and a Ribbed U-Duct with and without Rotation", *Journal of Heat Transfer*, Vol.123, No.2, pp.219-232.
- [17] Murata, A. and Mochizuki, S., 2001, "Large Eddy Simulation of Turbulent Heat Transfer in an Orthogonally Rotating Square Duct with Angled Rib Turbulators", *Trans. ASME, Journal of Heat Transfer*, Vol.123, pp.858-867.
- [18] Murata, A. and Mochizuki, S., 2004, "Effect of Rib Orientation and Channel Rotation on Turbulent Heat Transfer in a Two-Pass Square Channel with Sharp 180° Turns Investigated by Using Large Eddy Simulation", *International Journal of Heat and Mass Transfer*, Vol.47, pp.2599-2618.
- [19] Jia, R., Saidi, A., and Sunden, B., 2002, "Heat Transfer Enhancement in Square Ducts with V-Shaped Ribs of Various Angles", *ASME Paper*, GT-2002-30209.
- [20] Schukin, A. V., Kozlov, A. P., and Agachev, R. S., 1995, "Study and Application of Hemispheric Cavities for Surface Heat Transfer Augmentation", ASME Paper, 95-GT-59.
- [21] Chyu, M. K., Yu, Y., Ding, H., Downs, J. P., and Soechting, F. O., 1997, "Concavity Enhanced Heat Transfer in an Internal Cooling Passage", ASME Paper, 97-GT-437.
- [22] Gortyshov, Y. F., Popov, I. A., Amirkhanov, R. D., and Gulitsky, K. E., 1998, "Studies of Hydrodynamics and Heat Exchange in Channels with Various Types of Intensifiers", *Proceedings of the 11th International Heat Transfer Conference*, Vol.6, pp.83-88.
- [23] Mahmood, G. I., Sabbagh, M. Z., and Ligrani, P. M., 2001, "Heat Transfer in a Channel with Dimples and Protrusions on Opposite Walls", *Journal of Thermophysics and Heat Transfer*, Vol.15, No.3, pp.275-283.
- [24] Borisov, I., Khalatov, A., Kobzar, S., and Glezer, B., 2004, "Comparison of Thermo-Hydraulic Characteristics for Two Types of Dimples Surfaces", *ASME Paper*, GT2004-54204.
- [25] Nakamata, C., 2009, "Internal Cooling Structure for Hot Section Components", Japanese Patent Application No.2009-167860.
- [26] Bunker, R. S., Bailey, J. C., Lee, C.-P., and Stevens, C. W., 2004, "In-Wall Network (Mesh) Cooling Augmentation of Gas Turbine Airfoils", ASME Paper, GT2004-54260.
- [27] Murata, A., Mochizuki, S., Nakamata, C., and Okita, Y., 2008, "Large Eddy Simulation of Turbulent Heat Transfer in Stationary Channels with Dimples,

Protrusions, and Ribs", *International Journal of Transport Phenomena*, Vol.10, No.4, pp.323-336.

- [28] Carslan, H. S. and Jaeger, J. C., 1956, Conduction of Heat in Solids, 2nd ed., Oxford, pp.29-33 and pp.70-73.
- [29] Metzger, D. E. and Larson, D. E., 1986, "Use of Melting Point Surface Coatings for Local Convection Heat Transfer Measurements in Rectangular Channel Flows With 90-deg Turns", *Trans. ASME, Journal of Heat Transfer*, Vol. 108, pp. 48–54.
- [30] Nishida, S., Murata, A., Saito, H., and Iwamoto, K., 2011, "Compensation of Three-Dimensional Heat Conduction inside Wall in Heat Transfer Measurement of Dimpled Surface by Using Transient Technique", *J. Enhanced Heat Transf.*, submitted(Paper No. JEH(T)-3016).
- [31] Nishida, S., Murata, A., Saito, H., and Iwamoto, K., 2009, "Measurement of Heat and Fluid Flow on Surface with Teardrop-Shaped Dimples", *CD-ROM Proc. of Asian Congress on Gas Turbines*, Tokyo, Japan, August 2009, ACGT 2009-TS41.
- [32] ASME, 1998, Test Uncertainty: Instruments and Apparatus, *ASME Standard PTC 19.1-1998*, New York, NY, USA, ASME.
- [33] Kays, W., Crawford, and M., Weigand, B., 2005, Convective Heat and Mass Transfer, 4th Edition, McGraw-Hill Int. Editions, p.297.
- [34] Gee, D.L. and Webb, R.L., 1980, "Forced Convection Heat Transfer in Helically Rib-Roughened Tubes", *International Journal of Heat and Mass Transfer*, Vol.23, pp.1127-1136.