Fluid Flow and Infrared Image Analyses on Endwall Fitted with Short Rectangular Plate Fin

Kenyu Oyakawa[1], Islam Md. Didarul[2] and Minoru Yaga[3]

1. Graduate student, Faculty of Engineering, University of the Ryukyus, Okinawa, Japan.
2. Assoc. professor, Dept. of Mechanical Systems Engineering, University of the Ryukyus, Okinawa, Japan.
3. Professor, Dept. of Mechanical Systems Engineering, University of the Ryukyus, Okinawa, Japan.

An experimental investigation is carried out to study fluid flow and heat transfer characteristics on the endwall fitted with arrays (7 × 7) of short rectangular plate fins of different pattern (co angular and zigzag) for different pitch ratio. Experiments were conducted in a rectangular duct of 50 mm height for an air flow of Reynolds number ranged from 18750 to 62500 based on the equivalent diameter and air velocity of the duct. Infrared image analysis technique was employed to make clear the characteristics of local heat transfer coefficients on fin base, endwall and overall surface. Flow pattern around the short rectangular plates were visualized by inducing fluorescent dye in a water channel and longitudinal vortices and others were observed. Increasing the distance between plates in flow direction causes heat transfer enhancement for co angular pattern, while decreasing the distance causes heat transfer enhancement for zigzag pattern. Zigzag pattern with pitch ratio 2 is found to be more effective in heat transfer enhancement than any other cases investigated.

Key words: infrared image, rectangular plate, convection heat transfer, flow visualization.

INTRODUCTION

With an increasing demand of more compact light weight heat exchanger researchers are doing great effort to study heat transfer characteristics of longitudinal rectangular fin arrays. There needs a very efficient technique for the cooling of a powerful gas turbine engine blades for aircrafts as well as for chemical and process industries. Forced convection to fin array inside the turbine blade is a good technique for cooling. Heat transfer application is also important factor in thermal management of microelectronic component of digital computers as well as spacecraft. Heat transfer enhancement with arrays of rectangular blocks especially for the fully developed flow has been investigated by many researchers. Among them Sparrow et al [1,2] studied heat transfer and pressure drop in arrays of rectangular blocks with barriers and mixing blocks and focused the effect of the missing block and barriers on thermal hydraulic behavior of the array. Sparrow and Liu [3] studied heat transfer and pressure drop for laminar flow from arrays of plates of inline and staggered arrangement. For constant power and constant surface area heat transfer for the segmented array shows higher values than that of the parallel plate channel and the staggered arrays shows better performance than that of inline array. Recently, heat transfer enhancement by inclined rectangular plate which is set perpendicular to a duct is reported as the effective vortex generator by many researchers, because the longitudinal vortex and others produced by generator keeps its intense far downstream. Molki et al [4] studied experimentally heat transfer at the entrance region of an array of rectangular heated blocks and presented empirical correlations for the adiabatic heat transfer coefficients and thermal wake effects. Turk and Junkhan [5] also showed measuring the span wise heat transfer downstream of rectangular fin mounted on a flat plate. Oyakawa et al [6] also studied the heat transfer on plate setting rectangular fins with having attack angle of 20°. Bilen et al [7] investigated heat transfer enhancement from a surface fitted with rectangular blocks at different attack angle and found that the maximum heat transfer was obtained at attack angle of 45°, but did not inform about local heat transfer characteristics. T. Igarashi [8] studied heat transfer from a square prism with different attack angle and found reattachment flow at attack angle of 14° -35°. Kadle and Sparrow [9] investigated numerically and experimentally heat transfer from an array of parallel longitudinal continuous fins to a turbulent air stream passing through the inter-fin spacing. In the literature heat transfer from a surface having rectangular block, distance between blocks in flow direction and transverse to the flow direction was taken as parameter and in some cases attack angle as a parameter. But no work is found by the authors about the investigations of the heat transfer from the endwall surface for different pattern (co angular and zigzag) of short rectangular plate fins.

The aim of the present work is to analyze the heat transfer characteristics from fin base, endwall and overall surfaces due to different arrangement of fins for different pitch ratio in flow direction and to observe the flow vortices that enhance heat transfer. Detail analysis of heat transfer coefficients of endwall are observed by infrared image technique. With flow visualization the regions where the vortices attach are observed and found a relation with heat transfer enhancement from the endwall.

EXPERIMENTAL APPARATUS AND PROCEDURE

Schematic view of the experimental apparatus and fin patterns are presented in Fig. 1. It is a rectangular duct of 50 mm height, 250 mm in span width and 784 mm in length. Fins were set by 7 in line and 7 in row respectively. The first row of fins was located 200 mm away from the duct entrance. The fins were made of aluminum, of rectangular type of dimensions 20 mm long, 5 mm thick and three kinds of height of 5, 10 and 15 mm. The pitch ratio of separation distance S_x, S_z between fins to fin length L was chosen to be S_x/L= 2, 3 in stream wise and S_z/L=1 in span wise direction. The angle of the fins to the flow direction was kept 20 degree. Arrangement of the fins in the same angle to the same direction is called co angular pattern and the others that have same angle, but alternative directions after each row is called zigzag pattern. Air blower supplies air to the test section. Top wall of the duct, made of bakelite plate of thickness 10 mm and a 30 μm thick stainless steel foil (200 mm x 784 mm) was attached to inner side of the Bakelite plate in order to form the heating surface by supplying direct current. Fins were attached to the heating surface by means of 100 micrometer double sided thin tape. A window was formed by cutting the bakelite of top wall, exposing the foil in order to observe the infrared image of the representative 3rd and 4th row fins. Exposed foil of the window was perfectly blackened and was covered by polyvinyl chloride film that has a transmissivity for infrared energy to be nearly unity. As the fins are attached to the inner side of the top wall to prevent free convection through the window, window was covered by two layers of polyvinyl chloride film keeping nearly 2 mm gap between heating surface and 1st layer. An infrared camera of TVS 8000 that has a sensor of Indium-Antimony (In-Sb), which can measure temperature of 160x120 points with a resolution of 0.025°C on a black body was employed to take infrared images from the upper side of the top wall so that temperature of fin base and overall surfaces may be clearly observed. During the experiment temperature difference between mainstream flow and endwall was kept constant and heat flux was varied in the range of (q/A= 288-1552 W/m^2). Inlet temperature of the mainstream was measured by a copper constantan thermocouple located just at the entrance of the duct. All data were taken when the steady state was reached and it took a hour to reach the steady state.

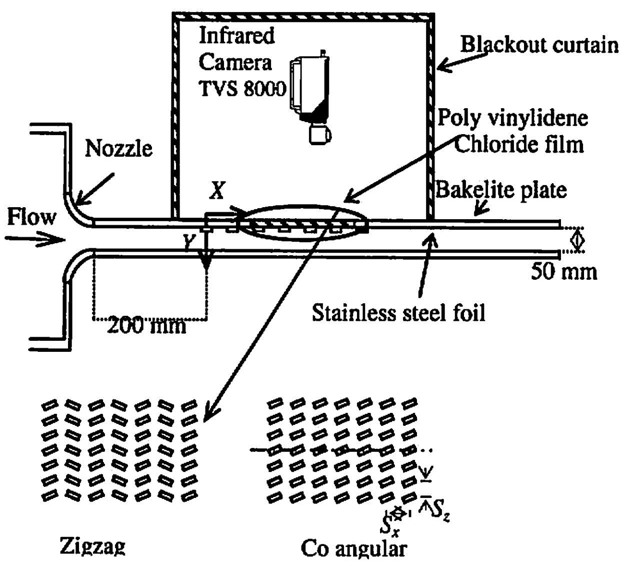


Fig.1 Experimental apparatus with fin pattern

RESULTS AND DISCUSSIONS

Flow visualization

In order to observe the flow behavior and to get insight the flow pattern around the rectangular fin, the vortices generated by fin of co angular and zigzag pattern were observed by fluorescent dye flow in a water channel. Vortices at co angular pattern and zigzag pattern for different *PR* are shown in Fig. 2. In case of co angular pattern for *PR*=2, it is shown that the dye flow stagnates front of the fin, forms the horse shoe vortex surrounding it. Rear of the fin, the vortex rolls up itself, rises along its end, and show the longitudinal behavior and then it joints to the flow separated from upstream portion of fin. It is also shown that vortex introduced by the 2nd row fin some what bypasses 3rd row fin just washing its top surface without sweeping the inter fin region strongly. With increasing the distance between fins in flow direction it is observed that vortex also generated by front corner of the 2nd row fin, touches the endwall, bottom wash the surface and then reattaches the

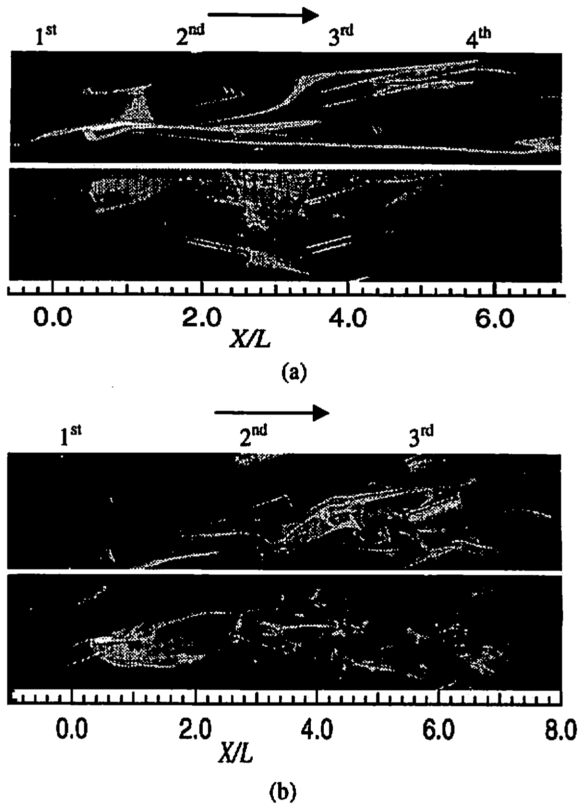


Fig. 2 Flow visualization for co angular and zigzag pattern for (a) *PR*= 2, (b) *PR* = 3

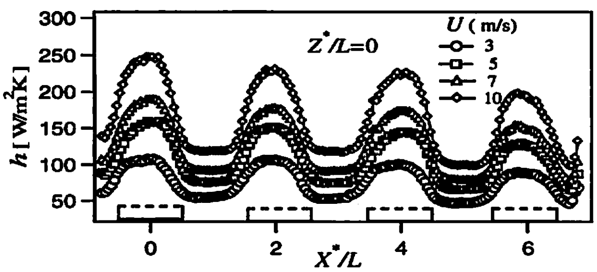
following fin. For the zigzag pattern flow behavior is somewhat different. For PR=2, 1st row fin creates vortex which touches front of the second row fin and bottom wash nearby heating surface, flow direction is changed because of zigzag settings and then strong longitudinal behavior appears between 2nd and 3rd row fin which advances and touches the endwall and front corner of the 3rd row fin. Flow separated from the front corner of 3rd row fin reattaches the following fin. As the pitch ratio becomes smaller, water flows look like a sinusoidal wave channel. But when the pitch ratio increases flow trends is little bit different. Though vortex introduced by the corner of the 1st row fin touches the endwall and 2nd row fin, but after 2nd row fin vortex is leaned.

Local heat transfer distributions

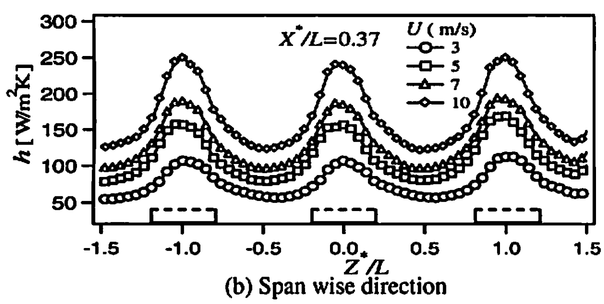
First to check the accuracy of the heating surface pre experiments were carried out with no fins varying mainstream velocity, which will be called smooth surface hereafter. The average Nusselt numbers in the downstream region almost agree with the Nusselt numbers of fully developed smooth duct flow of Eq.(1)

$$Nu_s = 0.023 Re^{0.8} Pr^{0.4} \tag{1}$$

Here, the rectangular plates were mounted on the heating surface keeping the attack angle of 20 degree as in previous experiment it was found that 20 degree angle is most effective among 0, 20 and 25 degree for co angular and zigzag pattern. Also T. Igarashi [8] and Bilen et al [7] got same result about the attack angle. Test run between the ranges of 3-10 m/s which correspond to the range of Reynolds number Re=18750- 62500. As a typical case of heat transfer distribution varying the velocity, distributions along stream wise and span wise



(a) Stream wise direction



(b) Span wise direction

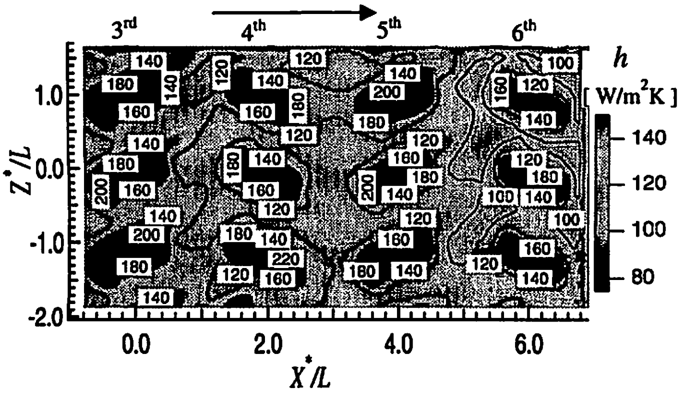
Fig. 3 Variation of local heat transfer distributions with velocity for zigzag pattern of PR=2, H=10 mm

directions are shown in Fig. 3(a) and (b) respectively, where X^*,Z^*(0,0) is at the center of 3rd row fins. Distributions show periodical pattern with higher values at fin base and lower ones at endwall and is analogous regardless of velocity.

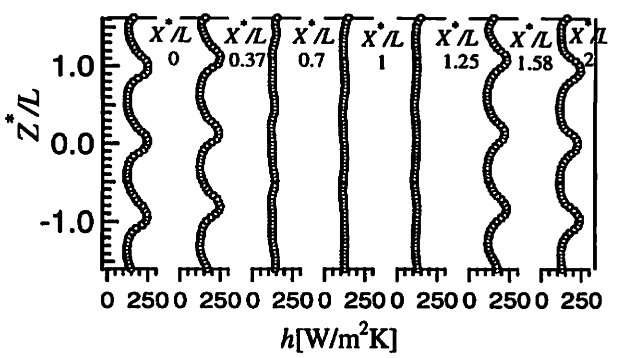
Detailed heat transfer coefficient distributions on the endwall surfaces of rectangular fin duct are shown in Fig. 4 for zigzag pattern as for typical example. Fig. 4(a) shows the infrared image on the endwall of zigzag pattern. From the overall heat transfer coefficient profile it is shown that higher heat transfer coefficient contour are close to the fin base for every rows of fin. Heat transfer in the region just ahead of 3rd row fin is higher and the region between fins in span wise direction also shows higher heat transfer. Enhancement ahead of the fins are due to the corner effect of the fin, as the corner edges increases the turbulence intensity due to vortices and attaches that region. Enhancement between fins in span wise direction may be due to lateral mixing caused by the side top edges of the fins. Heat transfer ahead of 4th row fins is somewhat different because of zigzag pattern of the fins but the corner effects are still observed. Heat transfer ahead of 5th row is similar to 3rd row.

In Fig. 4(b) heat transfer coefficients along centerline shows periodical distributions. Heat transfer coefficient sharply increases, reaches to a maximum and then sharply decreases to a minimum value following a nearly flat distribution and continues same pattern for every rows of fin investigated. It is clear that the maximum value is at the center point of fin base and higher heat transfer is observed due to the extended surface effect. Distributions between fins are almost flat and showing actually the endwall heat transfer and is nearly half of maximum values. Heat transfer coefficient along midline though shows nearly straight line distributions but shows somewhat higher values just after the fin and it indicates that in this region, flow attaches the endwall strongly.

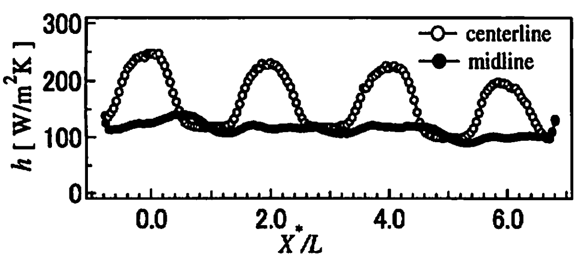
Fig. 4(c) shows the heat transfer distributions along span wise direction. At X^*/L=0, distribution is periodical, showing maximum heat transfer coefficient at fin center position and then slowly decreases, reaches to a minimum and then slowly increases. This pattern is repeated for every fin along Z direction. At X^*/L=0.37, slightly higher heat transfer coefficient is observed though distribution is periodical. When we move to downstream, i.e. at X^*/L=0.7, distribution tends to become straight, and at middle position between 3rd and 4th row fins (X^*/L=1) distribution almost straight. At further downstream it tends to become periodical again. And at X^*/L=1.58, 2 periodical distribution is appeared. It can be assumed that flow attaches strongly just front of the fin and behind the fin, not at the middle between fin rows. Periodical distribution also indicates that the heat transfer coefficients through other fins are identical. From Fig. 4(d), when PR is increased, heat transfer ahead of 3rd and 4th rows are found similar to the corresponding fin rows of PR=2. With increasing the PR, lower heat transfer region are increased ahead of representative fin rows. Also heat transfer contour are degraded around the fins with increasing PR.



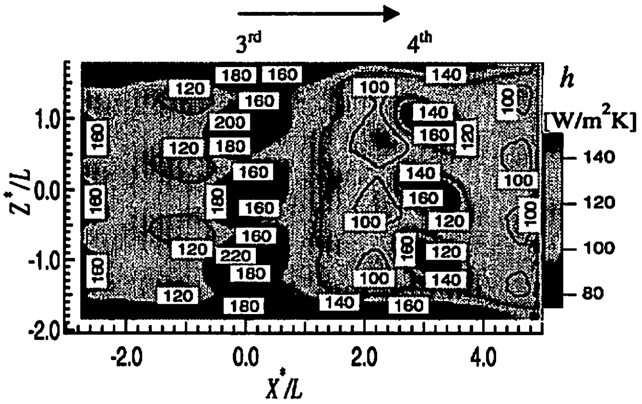
(a)



(c)



(b)

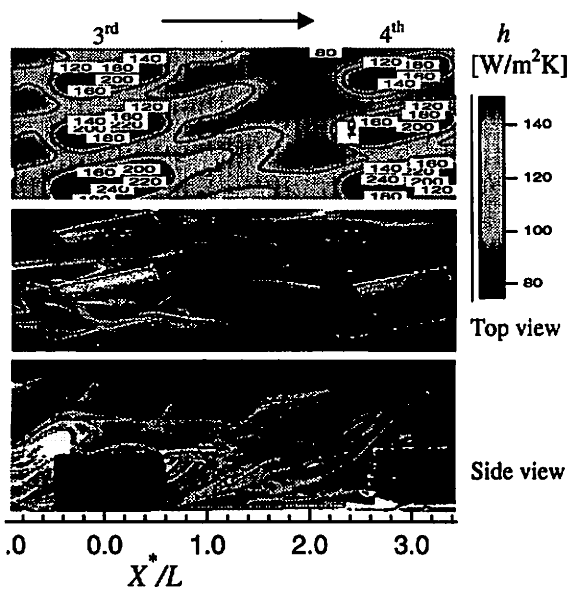


(d)

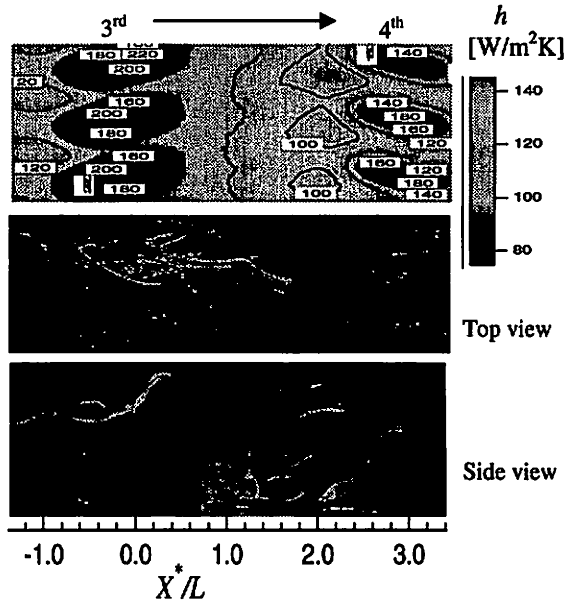
Fig. 4 Heat transfer coefficient distributions around the fins of zigzag pattern on the endwall for Re= 62500 and H=10 mm, (a) contour for PR =2, (b) centerline and midline distributions for PR=2, (c) span wise distributions for PR =2 and (d) contour for PR = 3

Comparison between infrared data and flow visualization

Fig. 5 shows the isotherms of infrared image and we then compares with flow visualization observed for the 3rd and 4th row regions of co angular and zigzag pattern of PR =3 and H=10 mm. Higher heat transfer contour are observed surrounding the fins for both pattern as the flow touches the front of 3rd row fins, sweeps the fin side wall along with bottom wash of the endwall surrounding the fins. Corner effect behind 3rd row fins are also observed



(a)



(b)

Fig. 5 Comparison between infrared image and flow visualization for PR =3, H =10mm of (a) co angular pattern (b) zigzag pattern and clearly visible at infrared image shown in Fig. 5(a). Comparatively higher heat transfer contour are shown on endwall between fins in stream wise direction for zigzag pattern than that of co angular pattern. Higher heat transfer coefficient contour is shown in fin base in case of co angular pattern than that of zigzag pattern and it is already discussed that flow separated from corner edge reattaches the following fins more strongly while *PR* is increased. From the side view it is clear that flow touches the endwall strongly for zigzag pattern while for co angular case flow touches the fin strongly. Thereby higher heat transfer contour appeared at fin base and endwall for co angular and zigzag pattern respectively for *PR*=3.

Area averaged heat transfer from endwall, fin base and overall surface

Average heat transfer coefficients on fin base, endwall and overall surfaces are measured by following equation,

$$\bar{h}_{overall}=\frac{A_{fin}}{A_{overall}}\bar{h}_{fin}+\frac{A_{endwall}}{A_{overall}}\bar{h}_{endwall} \qquad (2)$$

where, $A_{overall}$ is the overall surface area including fin base, A_{fin} and $A_{endwall}$ are surface area of fin base and end wall respectively. Fig. 6(a),(b) and (c) shows the relationship between the area-averaged Nusselt number and the Reynolds number. The area averaged Nusselt number is represented by curve fitting in the form of Eq.(3), where the coefficient *c* and exponent *m* of the

$$Nu = cRe^{m} \qquad (3)$$

equation are listed in Table 1. Figure 6(a) shows the variation of Nusselt number with Reynolds number when the average heat transfer coefficient of fin base is considered. Nusselt number is increased with Reynolds number for any case investigated. 15 mm fin shows the highest value and it might be due to extended surface effect. When compare Nusselt number between co angular pattern and zigzag pattern of *PR* =2, 3 of same height of *H* =10 mm, zigzag pattern shows the highest values for *PR*=2. Fig. 6(b) exhibits heat transfer enhancement from the endwall due to fin pattern. Between *PR*=2 and 3, for the zigzag pattern of 10 mm fin, average Nusselt number is somewhat higher for *PR*=2 than that of *PR*=3. So increasing the space in stream wise direction for zigzag pattern decreases heat transfer *PR*=2 than that of *PR*=3. So increasing the space in stream wise direction for zigzag pattern decreases heat transfer enhancement, but for co angular type pattern this behavior is opposite. In that case increasing the space in stream wise direction enhances heat transfer more.

Fig.6(c) shows the behavior of heat transfer enhancement with Reynolds number for different fin pattern on the overall surface area. Average Nusselt number for zigzag pattern of *PR*=2 and of *H*= 10 mm fin is higher than that of all cases investigated. As *PR* increases overall heat transfer increases incase of co angular pattern. This agrees with the data obtained by K. Bilen et al [7], where he found that as *PR* increases, heat transfer increases and the effect of attack angle is little

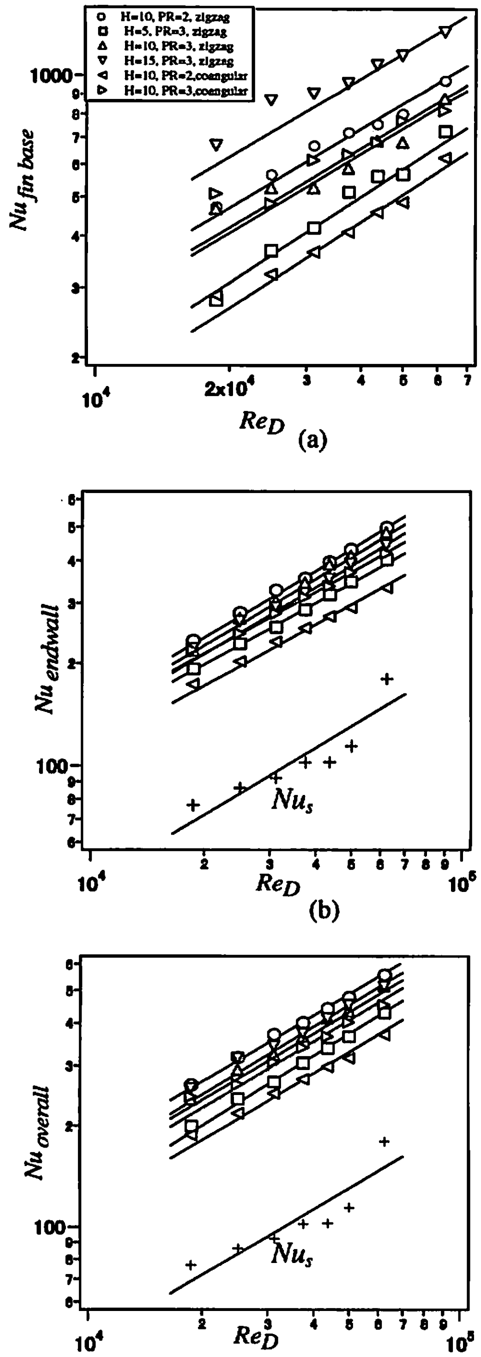


Fig. 6 Average *Nu-Re* relationship for (a) fin base. (b) endwall and (c) overall surfaces.

for larger than 22.5°. In case of co angular pattern, as the distance bet^n fin decreases, fluid bypasses over the top surface of the fins, resulting in bypass of the flow over the fins and increasing the formation of recirculation region between the sequential fins. When the distance between the fins in flow direction increases the effect of dead region and flow decreases; flow separated from the front corner reattaches the following fin. And so heat transfer increases in co angular case. On the other hand, for zigzag pattern as the sequential fins alternately changes the direction, flow reattaches the large surface area of fins as well as endwall. While increasing distance between fins in flow direction vortex reattaches endwall and pass out without touching fin surface.

Table 1 Coefficient c and exponent m of eq.3

			Fin base		Endwall		Overall	
Pattern	H	PR	c	m	c	m	c	m
zigzag	5	3	0.31	0.7	0.52	0.6	0.12	0.65
zigzag	10	3	0.65	0.65	0.36	0.65	0.4	0.65
zigzag	15	3	1.0	0.65	0.34	0.65	0.38	0.65
zigzag	10	2	0.75	0.65	0.39	0.65	0.43	0.65
coangular	10	2	0.26	0.7	0.45	0.6	0.29	0.65
coangular	10	3	0.67	0.65	0.56	0.6	0.36	0.65

	Nomenclature	PR	pitch ratio
D_h	equivalent diameter of the duct	X, Z	stream wise and span wise coordinate (X, Z= 0,0)
L	fin length = 20 (mm)		at the center of 1st row fin)
H	height of fin = 5 ~15 (mm)	X^*,Z^*	Stream wise and span wise coordinate, (X^*,Z^*= 0,0
$\bar{h}$	average heat transfer coefficient (W/m^{2}K)		at the center of a rectangular fin of 3rd row fin)
h	local heat transfer coefficient (W/m^{2}K)	ν	kinematic viscosity of fluid (m^2/s)
T_∞	bulk temperature of duct flow (°C)	λ	thermal conductivity of fluid (W/mK)
T_w	duct wall surface temperature (°C)		Subscripts
Re	Reynolds number (UD_h/ν)	s	smooth surface (no fin)

CONCLUSIONS

Experiments were carried out to investigate heat transfer and flow characteristics on fin base, endwall and overall surfaces in details for various rectangular fin pattern. The results are summarized as follows:

[1] Flow pattern and the vortices developed by co angular and zigzag pattern has been made clear by flow visualization. Longitudinal vortices along with corner vortices and others are observed which enhances heat transfer.

[2] Local heat transfer coefficients distribution for different fin pattern with different *PR* has been investigated.

[3] Area average Nusselt number for fin base, endwall and overall surface is measured and found that though extended surface effect is higher but overall heat transfer depend on that of endwall and the *Nu-Re* relationship is made clear.

[4] Zigzag pattern of *PR*=2 is found to be most effective for heat transfer enhancement while heat transfer tends to increase with decreasing PR for zigzag pattern and heat transfer increases with increasing *PR* for co angular pattern.

ACKNOWLEDGEMENT

This research work is supported by Heiwa Nakajima Foundation. This support is gratefully acknowledged.

REFERENCES

[1] Sparrow, E.M., Niethammer, J.E, Chaboki, A.: Heat transfer and pressure drop characteristics of arrays of rectangular modules encountered in electronic equipment, Int. J. of Heat Mass Transfer, Vol.25, pp. 961-973,(1982)

[2] Sparrow, E.M., Vemuri, S.B., Kadle, D.S.: Enhanced and local heat transfer, pressure drop, and flow visualization for arrays of block-like electronic components, Int. J. of Heat Mass Transfer, Vol.26, pp. 689-699,(1983)

[3] Sparrow, E.M., Liu, C.H.: Heat Transfer, pressure drop and performance relationships for in line, staggered and continuous plate heat exchanger, Int. J. of Heat Mass Transfer, Vol.22, No.7, pp. 1613-1625,(1979)

[4] Molki, M., Faghri, M., Ozbay, O.: A correlation for heat transfer and wake effect in the entrance region of an inline array of rectangular blocks simulating electronic components, ASME J. of Heat Transfer, Vol. 117, pp. 40-46, (1995)

[5] Turk, A.Y., Junkhan, G.H.: Heat transfer enhancement downstream of vortex generators on a flat plate, Heat Transfer, C.I. Tien et al. ed., Hemisphere Washington, Vol. 6, pp. 2903-2908, (1986)

[6] Oyakawa, K., Furukawa, Y.,Taira, T., Senaha, I.: Effect of vortex generators on heat transfer enhancement in a duct, Experimental Heat Transfer, Fluid Mechanics and Thermodynamics, M.D. Kelleher et al. (Edit), pp. 633-640, (1993)

[7] Bilen, K., Yapici, S.: Heat transfer from a surface fitted with rectangular blocks at different orientation angle, J. of Heat Mass Transfer, Vol.38, pp. 649-655, (2002)

[8] Igarashi, T.: Local heat transfer from a square prism to an airstream, Int. J. of Heat and Mass Transfer, Vol.29, No.5, pp. 777-784, (1983)

[9] Kadle, D.S., Sparrow, E.M.: Numerical and experimental study of turbulent heat transfer and fluid flow in longitudinal fin array, ASME J. of Heat Transfer, Vol.108, pp. 16-23, (1986)