HEAT TRANSFER AND FLOW CHARACTERISTICS IN DEVELOPING REGION THROUGH SQUARE DUCTS WITH SMOOTH AND RIBBED WALLS.

Khan R K[1], Ali M. A.T[2] and Akhanda M A R[3]
[1](Chief Instructor, Mechanical Engineering,
Bangladesh College of Leather Technology, Dhaka, Bangladesh.)
[2&3] (Professor, Department of Mechanical Engineering,
Bangladesh University of Engineering and Technology, Dhaka, Bangladesh.)

Abstract The effect of a rib wall over a smooth wall on flow characteristics, friction factors and heat transfer coefficients in square ducts with asymmetric heating have been investigated. Bottom walls (test specimens) for both ducts are made of 12.5mm thick aluminium sheets and are heated by passing a c current through the heater placed under the walls. The uniform heating is controlled using a digital temperature controller and a variac. The rib pitch to height ratio (p/e) of the rib surface is 16. The heat transfer phenomenon and flow characteristics across different sections of the test ducts have been investigated. The result shows that the average heat transfer coefficient and the average friction factor for the rib wall increases by 6.24 percent and 3.38 percent respectively over the smooth wall. Flow measurements show that the rib wall has flatter velocity profiles than that of the smooth wall. The flatter velocity profiles are responsible for producing higher heat transfer. For the evaluation of friction factor and Nusselt number two Semi-analytical correlation's have been developed for the use of heat transfer designers.

INTRODUCTION

In various types of heat transfer augmentation techniques, people are now showing more interest with large-scale surface roughness in the form of ribs of different geometry and orientations called turbulent promoter. These promoters at the wall surface help to eliminate the viscous sub-layer and enhanced heat transfer augmentation. However, the increase in heat transfer is accompanied by an increase in the pressure drop of the fluid flow. Many investigations have been directed towards developing predictive correlation's for a given rib geometry and establishing suitable geometry which gives a better heat transfer performances for a given pumping power.

Akhanda has been carried out an experimental study on ribbed surfaces with eight different geometric variations. Fully developed turbulent heat transfer in tubes or in non-circular ducts with repeated-rib-roughness has been studied extensively. Considerable data also exist for repeated rib-roughness in an annular flow geometry, in which the inner annular surface is rough and the outer surface is smooth, to simulate the geometry of fuel bundles in advanced gas-cooled nuclear reactors. Heat transfer and friction in rib-roughened non-circular ducts have been extensively investigated to understand the heat transfer augmentation mechanism and flow-characteristics against the pumping power. In these experiments, the effects of rib height, rib spacing, aspect ratio, rib flow attack angle, etc. on heat transfer augmentation mechanism and flow phenomenon have extensively been investigated, and using these experimental data efficient types of heat exchanger equipment are developed.

Experimental studies on convection heat transfer for laminar flow in entrance region on circular tubes have been carried out to understand the heat transfer mechanism and flow characteristics. An investigation with laminar forced convection heat transfer in entrance region through non-circular ducts has been carried out to have an idea on heat transfer augmentation technique and friction phenomenon. In the literature review, it is seen that a few experimental studies have been conducted on forced convection heat transfer through entrance region of ribbed-roughened non-circular ducts. So, to have a clear understanding on heat transfer augmentation technique, friction factors and flow characteristics in turbulent flow through the developing region in rib-roughened square ducts and to compare those with a smooth square duct, the present experimental studies have been carried out.

EXPERIMENTAL APPARATUS AND TEST PROCEDURES

An experimental facility has been constructed to test the augmentation technique and to provide both the smooth and the ribbed duct reference data. Air at room temperature and atmospheric pressure is flown into the

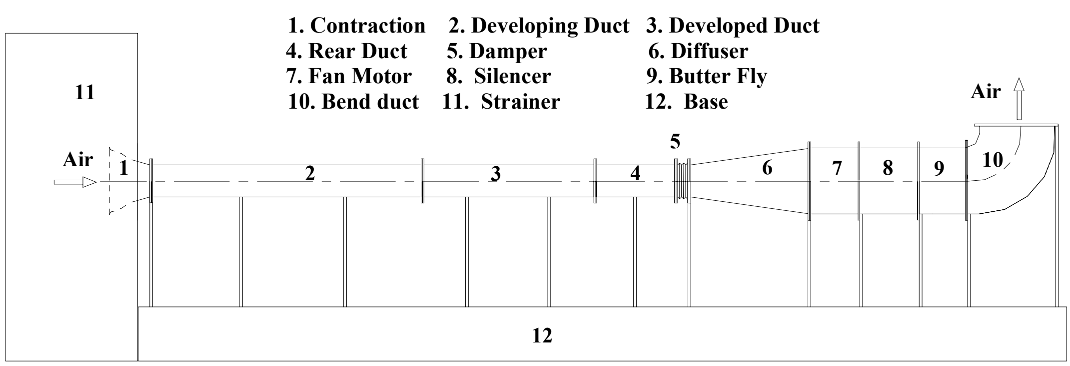

Fig.1 The schematic diagram of the experimental apparatus

square duct through a contraction. The hydraulic diameter of the duct is 50mm and its overall length is

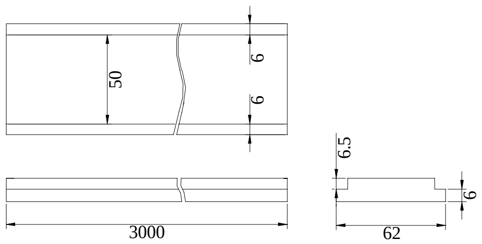

Fig.2 Orthographic views of the smooth wall.

5755mm. The nomenclatures of different parts and sections are shown in the experimental apparatus given in Fig.1. The lengths of the developing duct, developed duct and rear duct are 3000mm, 1830mm and 925mm respectively. The fully developed velocity profiles are

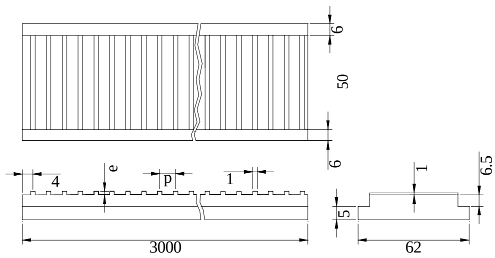

Fig.3 Orthographic views of a rib wall

attained about $55D_h$ (2750mm) downstream from the entrance of the developing duct. Bakelite sheet of 16mm thick is used to make its side walls and supporting wall. The top wall is made of 12.5mm thick clear perspex sheet and is fastened with the side walls by means of Allen bolts. A flexible duct (damper) is placed at the outlet of the rear duct.

A diffuser of length 920mm is placed between the outlet of the flexible duct and the inlet of the fan motor. The specifications of fan motor are of 2.75 hp and 2900 rpm. A silencer is introduced at the outlet of the fan motor to minimize creating sound and vibration. A butter-fly is set-up at the other end of the silencer to control the flow rate of air for measuring the required Reynolds number. At the end of the butter fly, a 90^o-bent duct is used to exhaust air to vertically upward. All elements of the experimental set-up are mounted on

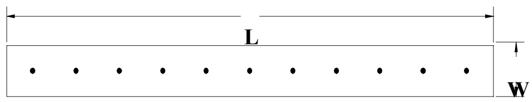

Fig.4 Distributions of thermocouples along the bisector of bottom surfaces of aluminium walls.

stands with proper levelling, at a convenient height of 1020mm from the base and different sections have been assembled to one another by means of flanges and nuts-bolts.

At the beginning of every set of experiment, a specified target temperature is fixed to the temperature controller. The voltage and current supplied to the heater are recorded from a digital voltmeter and a digital ammeter and thus heat energy generated to the heater is calculated. Eleven thermocouples are fitted to the bottom surface of the aluminium wall to measure its temperatures. Orthographic views of the smooth wall and ribbed wall are shown in Fig.2 and Fig.3

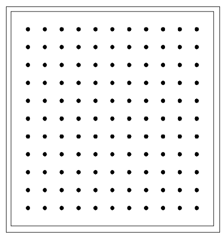

Fig.5 121 Specified points at inlet and outlet of the test duct.

respectively. The square sized rib having the dimensions of 1mm$\times$1mm$\times$50mm are prepared on aluminium plate by metal removing process. The ribs are oriented at 90^o with the flow direction. The distributions of thermocouples are at a pitch of 282mm, beginning at 90 mm from the leading edge and along the bisector of the bottom surface is shown in Fig.4. A digital thermocouple thermometer equipped with a selector switch is used to record the temperatures at their respective positions. The mean temperature of the bottom surface is calculated byaveraging the recorded temperatures and the mean temperature of its inner surface is corrected by using one dimensional conduction equation. The temperatures and velocities of the heated air flowing in the test duct are measured for

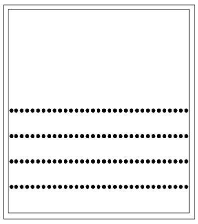

Fig.6 85 Specified points to measure vel. & temp. at four different levels.

121 specified points across its inlet and outlet as shown in the Fig.5.

From the recorded data average temperature and average velocity at inlet and outlet are calculated for every set of experiments. Thus, mean temperature and mean velocity of flow are evaluated by averaging the calculated average temperatures and average velocities. Temperatures and velocities are also measured at 85

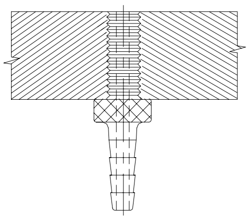

Fig.7 Setting of a pressure tap

specified points along Z-axis for four different levels like Y/B = 0, - 0.25, - 0.5 and - 0.75 across the duct cross-section, which is shown in Fig.6. These measurements are taken for three different lengths of X/D=3, 33 and 57. For each set of experiments, static pressures at wall are measured from eleven pressure taps. The taps are mounted at a pitch of 285mm in the range between X/D=3 to X/D=57 and at the same level of 25mm below its top-side. From the pressure differences along the length of the test duct the corresponding friction factors are evaluated. The setting of a pressure tape with the side wall is shown in Fig.7. The total uncertainties involve in measured values are estimated as 8.0 percent.

3, RESULTS AND DISCUSSIONS:

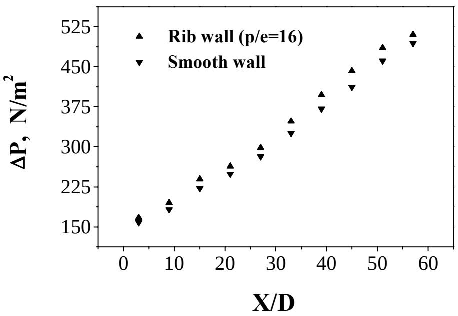

Fig.8 Comparisons of pressure drop along the length of ribbed and smooth duct.

(i) Effect of Reynolds number on Pressure Drops:

Figure 8 compares static pressure drops along the length of ribbed and smooth duct at a Reynolds number of 5.0×10^4. Pressure drops for both ducts increase with an increase of Reynolds number. The rib wall retards the flow velocity and develops more pressure drop rather than the smooth wall. For this reason, ribbed duct always exhibits higher pressure drops at every corresponding point than that of the smooth one. The analysis also concludes that the average pressure drop for rib duct is 3.32 percent higher over the smooth duct in the same range of Reynolds number.

(ii) Effect of Ribs on Air Temperatures between Outlet and Inlet of Two Test Ducts:

Figure 9 shows a comparison on temperature difference of air between the outlet and inlet of rib and smooth walls within the same range of Reynolds number. Ribs break the laminar sub layer & buffer layer and create local wall turbulence due to flow separation, reversal and reattachment between the ribs, which helps to exchange heat among the air particles. So, air particles absorb more heat while flowing in the rib wall than that of the smooth wall. For this reason, temperature differences for the ribbed duct always shows the higher value at every corresponding Reynolds number. In the analysis, it is seen that average temperature difference of air between the outlet and inlet of the ribbed duct increases by 5.10 percent than that of the smooth duct in the same range of Reynolds number.

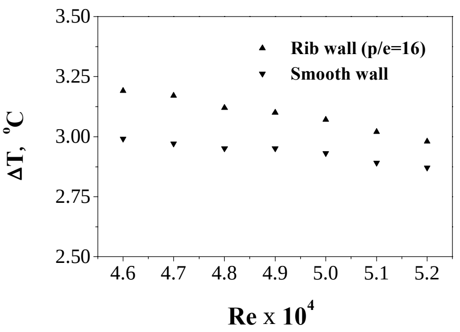

Fig.9 Comparisons on temp. differences bet^{n}outlet and inlet of test ducts.

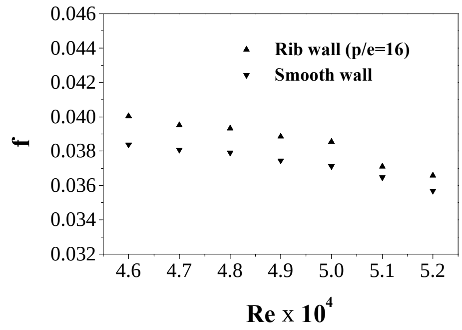

Fig.10 Effect of Re on friction factors between ribbed and smooth ducts.

(iii) Effect of Reynolds number on Friction Factors.

Effect of Reynolds number on friction factors between rib and smooth walls have been compared in Fig.10. For making this comparison, seven sets of experiments have been conducted for seven different Reynolds numbers. In the analysis, it is seen that due to ribbed surface, velocity of flow retards in the rib wall, which develops more pressure drop than that of the smooth duct. More pressure drop in the rib wall induced more friction factor and in the experiments it is observed that the average friction factor in the rib duct increase by 3.38 percent over the smooth duct in the same range of Reynolds number.

(iv) Comparisons on Heat Transfer Coefficients:

Figure 11 shows the comparisons on heat transfer coefficients between rib and smooth walla. As discussed in the previous article (ii) that Ribs break the laminar sub layer & buffer layer and create local wall turbulence due to flow separation, reversal and reattachment between the ribs, which helps to exchange heat among the air particles. So, air particles absorb more heat while flowing in the rib wall than that of the smooth wall. That is why, heat transfer coefficient in the rib wall always shows the higher value at every corresponding Reynolds number. In the result, it is observed that average heat transfer coefficient of air in the rib duct increases by 6.24 percent than that of the smooth duct in the same range of Reynolds number.

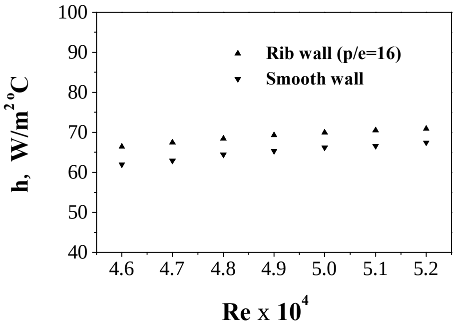

Fig.11 Comparisons on heat transfer coefficients between Two ducts.

(v) Primary flow Velocity and Temperature Distributions at X/D=3, 33 and 57:

Figure 12-13 show the primary flow velocity and temperature distributions between rib and smooth duct at X/D=33. The investigations have been carried out across Y-Z cross sections and at four different levels like; Y/B = 0, -0.25, -0.50 and –0.75. Velocities and temperatures are measured at 83 different points for every level as shown in Fig.6. Mean temperature distributions of air at X/D=33 is shown in Fig.12 Temperature distributions for both rib and smooth walls have minimum at the centre line of the duct. They increase gradually towards the side walls and having maximum at the nearest point of the walls. The locus of the mean temperature

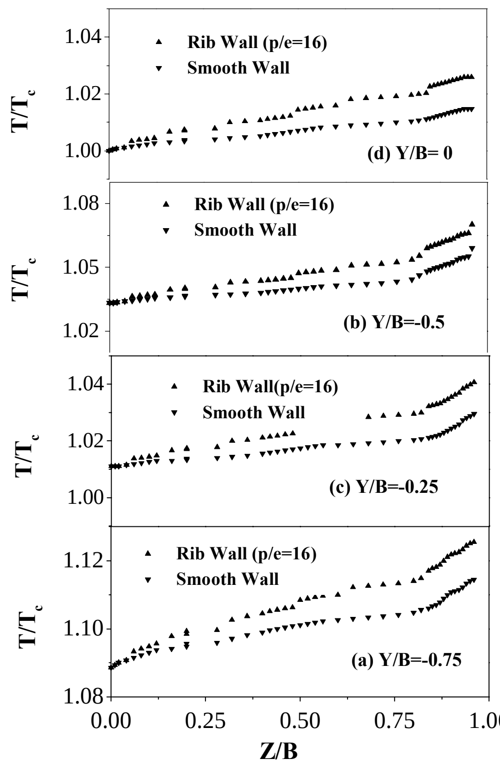

Fig.12 Primary flow Temperature distributions between ribbed and smooth walls at X/D=33.

distributions for smooth duct are almost regular curve profiles but due to rib surface the locus of the mean temperature distribution for rib wall having little bit zig-zag profiles.

Average air temperature at four different levels for both smooth and ribbed duct at X/D=3, 33 and 57are calculated and then percentage of increasing air temperatures are evaluated on the basis of the centre line temperature of air at inlet of the smooth duct. In the smooth duct at X/D=3, it is seen that at Y/B = 0, -0.25, -0.50 and -0.75 the percentage of increasing temperatures are 1.04, 1.49, 1.96 and 2.65 respectively, while in ribbed duct the corresponding percentage are 1.1, 1.75, 2.58 and 4.49. At X/D=33 in the smooth duct the percentages of enhancing temperatures are 2.28, 3.36,

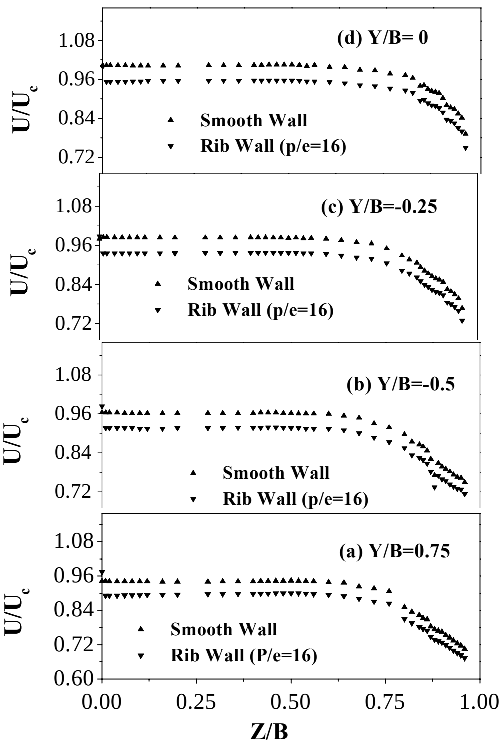

Fig.13 Primary flw velocity distributions between rib and smooth walls at X/D=33.

5.81 and 11.82 respectively, but in ribbed duct the corresponding percentages are 2.98, 4.06, 6.52 and 12.52. In the analysis at X/D=57, it is observed that, in smooth duct the percentages of enhancing air temperatures are 9.14, 12.04, 15.12 and 18.86 respectively, on the other hand in the ribbed duct the corresponding percentages are 9.51, 12.80, 616.10 and 20.20 respectively.

Primary flow velocity distributions of air for four different levels at X/D=33 are shown in Fig.13. It is seen that the velocity distributions for rib wall are little bit apart, more zig-zag, more flattened and having less value at every corresponding point than that of the smooth duct. This is due to the presence of rib wall

(vi) Development of Semi-analytical Correlation's:

Figure 14 and figure 15 exhibit the curve fittings to the data obtained from the experiments of ribbed duct. The curve fittings have been developed two semi-analytical correlation's related to the friction factor and Nusselt number. These equations which will be used by heat transfer designers.for the determination of forced convection heat transfer in the similar types of ribbed-duct in developing flow, are given in the following equations.

$$f = 0.58\,\mathrm{Re}^{-0.25}\left(\frac{p}{e}\right)^{-0.0025} \quad (1)$$

$$Nu = 0.0262\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.33}\left(\frac{p}{e}\right)^{-0.0025} \quad (2)$$

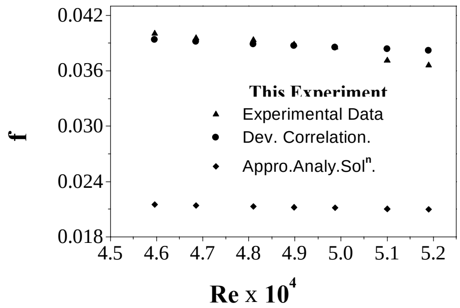

Fig.14 Developed correlation of friction factor from experimental data and other researcher's.

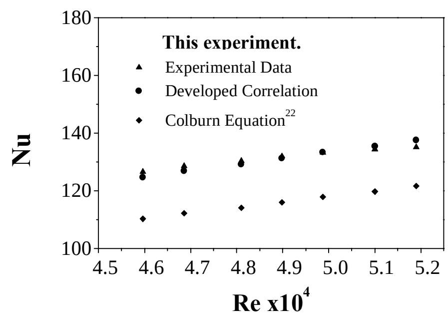

Fig.15 Developed correlation of Nu based on expt. data & compared with other researcher's.

CONCLUSIONS

The effects of rib turbulators on primary flow characteristics and heat transfer coefficient in square ducts with a ribbed and a smooth surface have been compared. The main findings of the study are;

(i) Pressure drops increase with an increase of Reynolds number and in the result it is seen that the average pressure drop in ribbed duct is 3.32 percent higher than that of the smooth duct at Re=5.0×10^4

(ii) The temperature difference of air between the outlet and inlet of ribbed duct shows 5.10 percent higher over the smooth duct in the same range of Reynolds number.

(iii) Friction factors for both ducts decrease with an increase of Reynolds number, and it is seen that the average friction factor in rib wall is 3.38 percent more than that of the smooth wall.

(iv) Heat transfer coefficients for both walls increase with an increase of Reynolds number and in the result it is concluded that the average heat transfer coefficient in rib wall is 6.24 percent higher over the smooth wall.

(v) Primary flow temperature distribution for both duct have the lower value at the centre line of the duct and it increases gradually towards the side walls and have the higher values at the nearest point of the adjacent walls.

(vi) Primary flow velocity distribution for both duct have the lower value at the nearest point of the adjacent walls and it increases gradually to the middle point between the centre line and the wall and then have almost the same values up to the centre line of the duct.

(v) Two developed semi-analytical equations in the rib wall will be used for the determination of friction factor and the Nusselt number in forced convection heat transfer in ribbed duct.

ACKNOWLEDGEMENT

The authors wish to express their appreciation to all the staffs of different workshops and laboratories of Bangladesh University of Engineering and Technology, Dhaka Babgladesh for their helping at various stages of the project.

NOMENCLATURE

f	Friction factor
h	Heat transfer coefficient, W/m^2 ^{o}C.
L	Length of developing duct, 3 m
Nu	Nusselt number
p/e	Rib pitch to height ratio.
ΔP	Pressure difference N/m^2.
Pr	Prandtl number
Re	Reynolds number
St	Stanton number
ΔT	Temp. difference bet^n outlet and inlet of the duct, ^{o}C.
T	Time average temperature,. ^{o}C
T_c	Centre line temperature, ^{o}C
U_c	Centre line velocity, m/s
U	time average velocity of air, .m/s
X,Y,Z	Three axes of the co-ordinate system.

REFERENCE

Akhanda. M. A. R., “Enhancement Heat Transfer in Forced Convective Boiling,” Ph.D Thesis, University of Manchester Institute of Science and Technology, U.K, 1985.

Ali T.A.M, “Flow Through Square Duct with Rough Ribs.” Ph.D Thesis, Imperial College, University of London, U.K, 1978.

Colburn. A. P., "A Method of Correlating Forced Convection Heat Transfer Data and a Comparison with Liquid Frictions." Trans. AIChE, Vol.29, PP.174- 210, 1933.

Dalle Donne. M., and Mayer. L., "Turbulent Convective Heat Transfer from Rough Surfaces with Two Dimensional Rectangular Ribs." International Journal of Heat and Mass Transfer, Vol.20, pp.582-620, 1977.

Hirota. M., Fujita. H., and Yokosawa. H., “Experimental Study on Convective Heat Transfer for Turbulent Flow in a Square Duct With a Ribbed Rough Wall (Characteristics of Mean Temperature Field).” Journal of Heat Transfer, Vol.116, pp. 332-340, May 1994.

Han. J. C., Park. J. S., and Lie. C. K., "Heat Transfer Enhancement in Channels with Turbulent Promoters." ASME, Journal of Engineering for Gas Turbines and Power, Vol.107,

Han. J. C., Zhang. Y. M., and GU. W. Z., “Heat Transfer and Friction in Rectangular Channels with Ribbed or Ribbed Grooved Walls.” Journal of Heat Transfer, Vol. 116, pp.58-65, February 1994.

Han. J. C., "Heat transfer and Friction Characteristics in Rectangular Channels with Rib Turbulators." Journal of Heat Transfer, Vol.110, pp.321-328, May 1988.

Han. J. C., “Heat Transfer and Friction in Channels with Two Opposite Rib-Roughened Walls”. Journal of Heat Transfer, Vol. 106, pp. 774-781, November 1984.

Hanna. J. C., “Heat Transfer and Friction Characteristics in Rectangular Channels with Rib Turbulators.” Journal of Heat Transfer, Vol. 110, pp. 321-328, May 1988.

Hishida. M., "Turbulent Heat Transfer and Temperature Distribution in the Thermal Entry Region of a Circular Pipe." Bulletin of the JSME , Vol.10, No.37, pp.113-123, 1967.

Hong. S. W., and A. E. Bergles, "Laminar Flow Heat Transfer in the Entrance Region of Semi-Circular Tubes with Uniform Heat Flux." Internal Journal of Heat and Mass Transfer, Vol.19, pp123-124, 1976.

James, D. D,“Forced convection Heat Transfer in ducts of Non-circular section.” Ph.D Thesis, Imperial College, University of London, U.K, 1967.

Monohar. R., "Analysis of Laminar Flow Heat Transfer in the Entrance Region of Circular Tubes," Internal Journal of Heat and Mass Transfer, Vol.12, pp.15-22., 1969.

Sparrow. E. M., "Analysis of Laminar Forced Convection Heat Transfer in Entrance Region of Flat Rectangular Ducts," NACA Tech Notes, TN 3331, 1955. pp.628-635, 1985.

Ulrichson. D. L., and R. A. Schmitz., "Laminar Flow Heat Transfer in the Entrance Region of Circular Tubes," Internal Journal of Heat and Mass Transfer, Vol.8, pp253-258, 1965.

Von Karman., "The Analogy Between Fluid Friction and Heat Transfer," Trans, ASME, Vol.61, PP.705-711, 1939.