Friction Factor and Heat Transfer in Equilateral Triangular Ducts with Surface Roughness

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Experimental investigations were conducted to study forced convection of fully developed turbulent flows in horizontal equilateral triangular ducts with different surface roughness pitch ratios (P/e) of 4, 8, and 16 on one side. The duct's bottom wall was heated uniformly and the other surfaces were thermally insulated. To understand heat transfer enhancement mechanism, heat transfer rates were measured. Smooth triangular ducts were also tested for benchmark purposes. The results were compared with previous results for similarly configured channels, at which they were roughened by regularly spaced transverse ribs in the rectangular and circular channels.

Key Words : Ridge Type Two Dimensional Roughness, Equilateral Triangular Duct, Friction Factor, Enhanced Heat Transfer, Efficiency Index

1. Introduction

The application of rib-type turbulators on high heat flux surfaces has attracted much attention for their significant enhancement of heat transfer (Webb, 1995). The use of ribs has been explored for applications such as nuclear reactors, electronic cooling devices, and heat exchangers. The ribs can break up the viscous sublayer of the flow and promote local wall turbulence that, in turn, increases heat transfer. In addition, a thermally conductive rib attached to a heated wall provides a greater surface area for heat transfer than a ribless wall. Various investigations (Han et al., 1978; Han, 1988; Kwon et al., 2000) examined the effects of geometric parameters such as the duct aspect ratio, duct angle, rib height, rib angle-of-attack, rib shape, rib shape ratio, and relative arrangement of the rib.

Webb et al. (1971) performed experiments on a tube with internal ribs. Webb et al. covered a wide range of rib height to hydraulic diameter ratios, but only pitch to height ratios greater than ten were used, and the ribs were aligned normal to the main stream direction. In the nuclear reactor area, considerable data exist for repeated-rib roughness in an annular flow geometry (Wilkie, 1966; Lee et al., 1990 and Ahn, 1999a). According to Hall (1962), when friction data, taken for an annulus with one rough wall and one smooth wall, were based on an equivalent diameter, they should agree with the results obtained in pipe flows with geometrically similar roughness. Wilkie(1966), using a simplification to the Hall transformation, found that a surface has the highest Stanton number and friction factor when the pitch to height ratio is between seven and eight.

The majority of these studies were mostly conducted with tubular, annular, square, and rectangular cross-sections. Althongh heat exchangers with triangular cross-sectional passages have high ratio of heat transfer area to core volume (Kays and London, 1984), and considerably lower fabrication costs than those for the shell-and-tube heat exchangers, data on the effects of surface roughness of triaugular duct's

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(Manuscript **Received** July 3, 2000 ; **Revised** February 20, 2001)

wall on forced convection are still lacking.

Therefore, this work focused on the heat transfer characteristics of equilateral triangular ducts with one wall only roughened by ribs. Equilateral triangular duct was chosen because it gives the best convective heat transfer performance among triangular configurations (Shah and London, 1978). Forced convection of fully-developed laminar flows in triangular ducts with high thermal conductivity was analyzed by Kays and Crawford (1993). Experiments in a sharp-cornered equilateral triangular duct were performed by Altemani and Sparrow (1980), and heat transfer characteristics in the entrance region and fully developed region were determined. The heating arrangement had two walls heated with uniform heat input per unit axial length, and the third wall was thermally insulated. Nevertheless, these investigations (Altemani and Sparrow, 1980; Kays and Crawford, 1993) did not focuse on the effect of surface roughness in triangular ducts.

Recently, Kang et al. (1998) conducted experiments on the forced convection of fully developed turbulent flows in an equilateral triangular duct with very small surface roughness values of 1.2 μm, 3.0 μm, and 11.5 μm. The entire inner wall of duct was heated uniformly, and the outer surface was thermally insulated. They concluded that ducts with high surface roughness can enhance convective heat transfer performance.

The objective of present experimental study is to examine the effects of roughness pitch to height ratio on the heat transfer and friction characteristics in an equilateral triangular duct with one uniformly heated wall roughened by square ribs of 2 mm, and two smooth insulated walls.

2. Experimental Apparatus and Test Procedure

Measurements were carried out in an open-circuit suction type wind tunnel. A schematic diagram of the experimental apparatus is shown in Fig. 1. A 0.86-kW blower forced air through the test channel. A long transition section was used to insure that the air entering the test section had a uniform velocity distribution. At the end of the test section, the air was exhausted into the atmosphere.

The test channels were constructed by using fir plates (10 mm thickness) and aluminum plates (5 mm thickness). A 75 mm×75 mm×75 mm equilateral triangular duct with a 3600mm length was fabricated. Aluminium square ribs (2 mm×2 mm) were glued onto the bottom plates. As given in Fig. 1, the lower horizontal wall of the test section is heated and remaining two duct walls are thermally insulated. Upstream of the test duct is an unheated roughened entrance section. This unheated duct provides a hydrodynamically fully developed condition at the entrance of the heated test channel. The heated surface, shown schematically in Fig. 1, is constructed using aluminum segments. The aluminum plate and rib surfaces are polished to minimize emissivity and radiative losses. To study the effects of roughness pitch ratio (P/e), the square ridge type ribs were sequentially installed on the plate in order of P/e=16, 8 and 4.

In channel, only 1800 mm of the bottom plate was heated by a woven heater. The heater was independently controlled by a variac transformer, which provided a controllable constant heat flux for the test plate. Woven heaters embedded in silicone rubber were clamped uniformly between a bakelite panel and the aluminum plate to insure good contact. The heater could provide a constant heat flux for the entire test surface using a watt meter. The blower was capable of providing a wide range of air velocities so that the Reynolds number based on the hydraulic diameter could be varied between 10, 000 and 70, 000. The pressure drop across the test section was measured by a micro-manometer (FCO-12) calibrated by an inclined manometer. A total of 16 pressure taps were installed along the spanwise centerline of the bottom aluminium plate for local static wall pressure measurements. Each tap was connected to a micro-differential transducer, an amplifer, and a digital readout. The time to reach steady state was less than 60 minutes.

Over the range of test conditions, the wall-to-

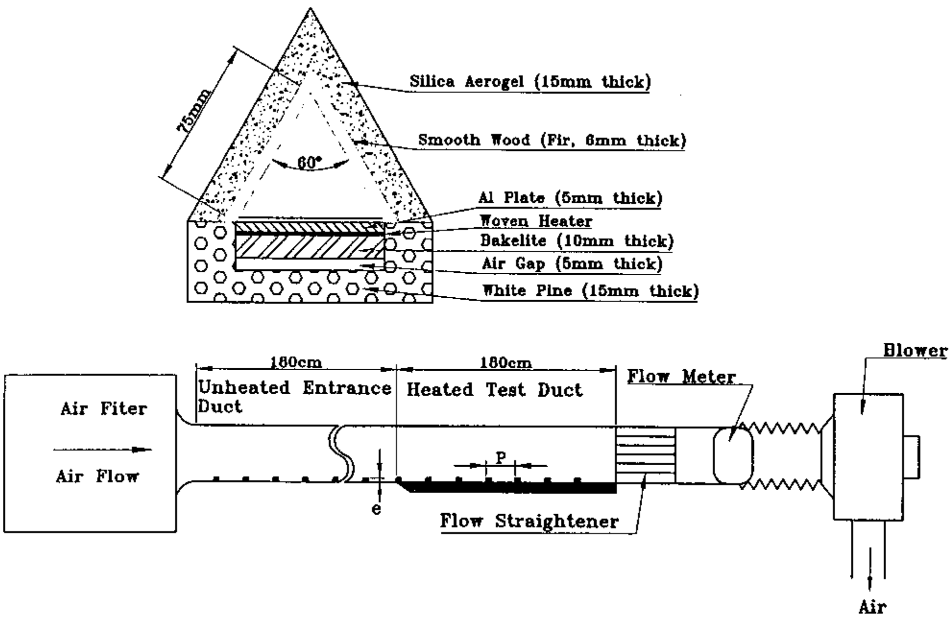

Fig. 1 Schematics of experiment setup

bulk fluid temperature difference varied between 10℃ and 35℃. Since the turbulence generated by ribs dominated over the secondary flow generated by buoyancy forces, the influence of free convection on heat transfer was believed to be negligible. To minimize conductive heat loss, the backside surface of the bakelite board was insulated using a 15-mm-thick section of pine wood with a 5-mm-thick clearance (air gap) between them (Fig. 1). The floor surface was instrumented with eighteen thermocouples. The junction-beads of the thermocouples were embedded into the wall and then ground flat to ensure that they were flush with the surfaces. In addition, a thermocouple probe and a thermocouple rake were placed at the test duct inlet and exit, respectively to measure air temperature. The temperature signals were transferred to a hybrid recorder, and subsequently sent to a computer.

The fully developed average heat transfer coefficients for triangular ducts were determined by the ratio of the floor heat flux to the difference between the floor and air bulk mean temperatures.

$$h = Q_{net}/[A(T_w - T_b)] \tag{1}$$

where the heat transfer area A is always that of the smooth bottom wall. Q_{net} is the net power defined by the following energy balance equation

$$Q_{net} = Q_t - Q_c - Q_r \tag{2}$$

where Q_t is the total power input to the test section, Q_c is the conduction heat loss to the environment, and Q_r is the radiative heat loss from the roughened surface to its surroundings. Heat loss from the outside of the duct wall through the bakelite to pine wood was quite low (about 0.5 percent for the maximum). The value of Q_r was evaluated using a diffusive gray-surface network (Siegel and Howell, 1981), and the loss due to radiation was less than 1.5 percent of the total power input. The bulk temperature T_b was measured from the following equation:

$$T_b = \frac{Q_{net}}{A_c u_b C_p \rho_{air}} + T_{air} \tag{3}$$

where A_c and u_b are the channel cross sectional area and bulk velocity, respectively. The ambient temperature (T_{air}) was about 23℃. From Eq. (1), the Nusselt number (Nu) is given by:

$$Nu = \frac{hDe}{K} \tag{4}$$

where k is the thermal conductivity. The friction

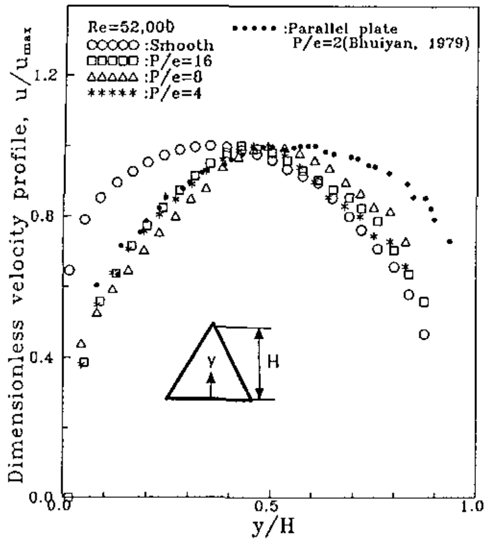

Fig. 2 Velocity profiles

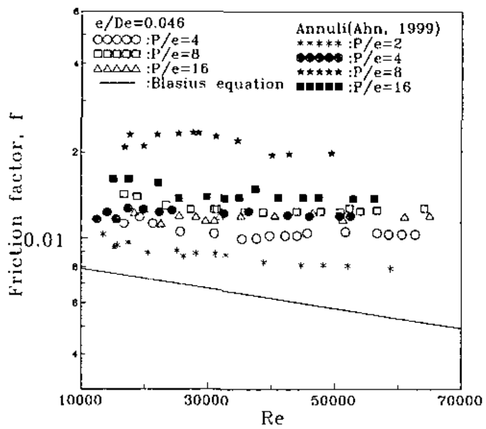

Fig. 3 Centerline friction factor for triangular duct

factor was calculated from the static pressure drop across the flow channel (ΔP) and the mass flow rate of the air (G) as

$$f=\Delta P/[4(L/De)G^2/2\rho] \tag{5}$$

A conservative estimate of the accuracy of the temperature measurement is ± 1 ℃. The heat flux was measured by a watt meter with a maximum uncertainty of ± 3.2 %. The uncertainty associated with the length scale used in data reduction was ± 2 mm. It was found that for the minimum flow rate, the worst case, for Reynolds number and friction factor uncertainties were ± 6.5 and $\pm$ 8.5 %, respectively. The uncertainty estimate procedure was described by Kline and McClintock (1953).

3. Results and Discussion

Before initiating experiments with surface roughness, the friction factor and heat transfer rates for smooth ducts were measured and compared with literature values. From the pressure drop measurements, the developing length was found to be sufficiently long to guarantee a hydrodynamically fully developed flow, as witnessed by the measured linearity of the axial pressure distribution.

Figure 2 represents the streamwise velocity distributions measured with a pitot tube at the channel centerline in the fully developed region. The test section was situated at 71 equivalent diameter lengths (De) away from the entrance. Smooth-wall experiments (open circular symbols) are also shown for comparison. Due to the turbulence mixing caused by ribs, the velocity gradients near the walls for the rib geometries are higher than those for smooth walls. The higher slope of velocity gradient enhanced the convective heat transfer rate by enhancing momentum and energy transfer. For the case of P/e=8 the position of maximum velocity moved to the highest place at the range studied. This phenomenon is in a line with Wilkie's results (1966) showing that the highest wall frictional resistance occurs when the roughness pitch and flow reattachment distances are equal.

The data for parallel plates with ridge type roughness on one side (Bhyuiyan, 1977) are also skown in Fig. 2. The maximum velocity is much higher than that from the present study. It may be attributed to different test section geometries. To examine the effects of roughness on the friction characteristics, static wall pressure measurements were undertaken. The calculated friction factors (Eq. (5)) for the bottom wall of triangular ducts are shown versus the Reynolds number in Fig. 3. The friction factors are the highest at P/e=8. The friction factors for the smooth triangular duct are almost in line with Blasius' solution for smooth circular tubes. The deviations between Blasius'

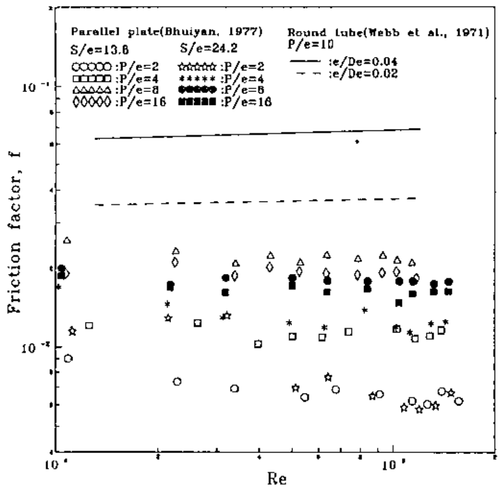

Fig. 4 Friction factors for similarly configured channels(Bhyuyan, 1977; Webb et al., 1971)

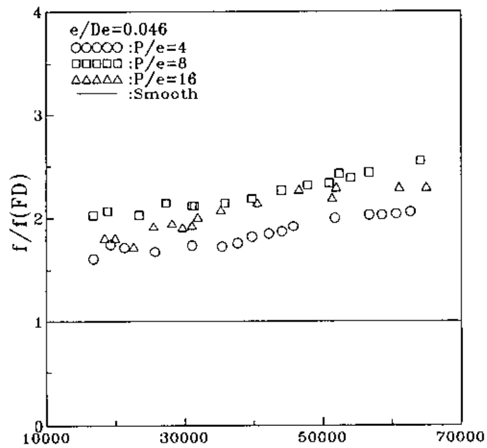

Fig. 5 Normalized friction factor

solution and the present data are within 6 %. For comparison, the concentric annuli of radius ratio (α)=0.4 having surface roughness elements on the inner tube only (Ahn, 1999b) is included in Fig. 3. The friction factors in the concentric annuli showed wider scatter than in the present work depending on roughness pitch ratio. It is supposed that the kinetic energy and fluid element size along radial direction distance due to the convex curvature effect are more strongly varied in the concentric annuli. Figure 4 illustrates the friction factors of previous investigations for similarly configured channels (i.e., the channel

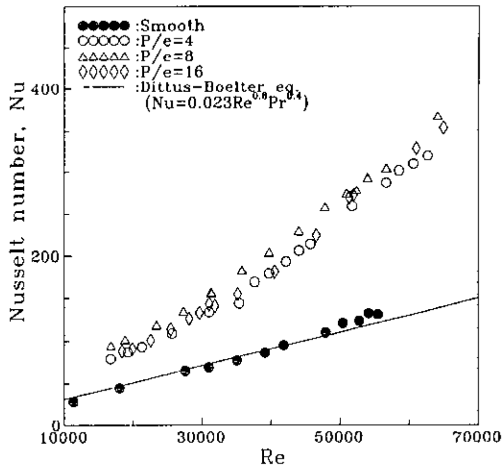

Fig. 6 Nusselt number

between the parallel plates with square ribbed roughness on the one side only (Bhyuiyan, 1977)) and circular ducts with square ribbed roughness on the entire wall (Webb et al., 1971). The friction factors for S/e=13.8 are a little higher than in S/e=24.2 in the channel between the parallel plates. Figure 4 shows that the roughness height ratio has less influence on the friction factor than the roughness pitch ratio. Note that an additional experiment has been conducted to examine the conductivity effect on the friction factor under heated wall conditions. No difference in the friction factor, beyond experimental uncertainties, is observed between thermally active and nonactive channels. The normalized friction factors are illustrated in Fig. 5, in which f(FD) stands for friction factors for smooth ducts. The normalized friction factor increases for P/e=4, 8, and 16. The normalized friction factor in a square duct with a pair of opposite rib-roughened walls is higher than that in a triangular duct with one rib-roughened wall duct to the difference in flow deflection and impingement. The Nusselt numbers (Nu) obtained from Eq. (4) are illustrated in Fig. 6. The results for P/e=8 were the highest as in Fig. 3 because the highest radial direction kinetic energy occurs when the most active interaction between the recirculating fluid flow in the groove and the entire fluid flow over the roughness in the channel takes a place. And Dittus and Boelter's equation was almost in line

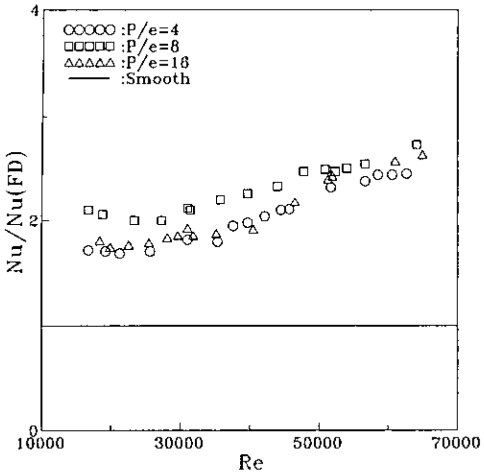

Fig. 7 Normalized Nusselt number

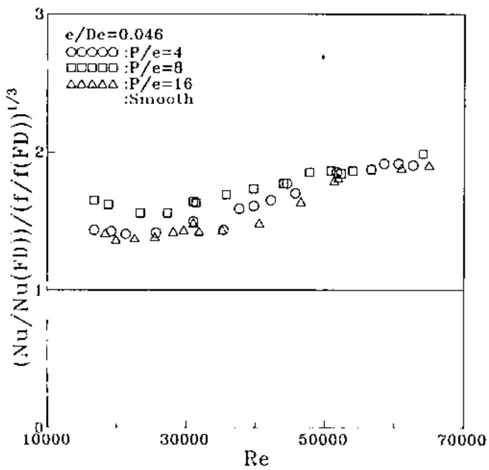

Fig. 8 Efficiency index

with the smooth triangular duct. The heat transfer coefficients on rib-roughened walls (from Eq. (1)) are shown by the average Nusselt number ratio in Fig. 7. Nu(FD) is the Nusselt number for a smooth triangular duct. $Nu/Nu(FD) > 1$ means that the Nusselt numbers in ribbed ducts are higher than those in smooth ducts. The normalized Nusselt number increases slightly with increasing Reynolds number, indicating that the increase in heat transfer in the rib-roughened channels is more significant at higher Reynolds numbers.

One criterion for evaluating the performance of an enhanced surface is to compare the heat transfer of a ribbed duct to that of a smooth-walled duct under constant pumping power constraint. According to Webb and Eckert (1972) and Han et al. (1985), an efficiency index $(Nu/Nu(FD))/(f/f(FD))^{1/3}$ can decide whether or not a given ribbed surface is potentially gainful. Fig. 8, where $(Nu/Nu(FD))/(f/f(FD))^{1/3}$ is plotted against Re, shows that all ribbed ducts show the same trend values of $(Nu/Nu(FD))/(f/f(FD))^{1/3}$ are lower at lower Re than those at higher Re. This behaviour was mainly due to the smooth duct friction factor (f(FD)), which is a function of Reynolds number. All of the values of $(Nu/Nu(FD))/(f/f(FD))^{1/3}$ are higher than 1, as shown in Fig. 8. Thus, the roughened channels have better heat transfer performance than smooth ducts. Furthermore, of P/e=8 has the best performance in the range we studied.

4. Conclusions

Experiments were conducted or heat transfer augmentation mechanisms in equilateral triangular ducts with one-wall roughened by a ridge type rib. The rib height-to-channel hydraulic diameter ratio was fixed at e/De=0.046. The flow Reynolds number varied from Re=10,000 to 70,000, while the rib pitch-to-height ratio was varied between P/e=4 and P/e=16. Velocity and friction factor measurements provide a better understanding of the heat transfer enhancement mechanism in ribbed ducts. The main findings based on these experiments are summarized as follows:

(1) The increases in friction factors and heat transfer can be seen in order of P/e=4, 16, and 8 for the three cases studied.

(2) For P/e=8, the position of maximum velocity moved to the highest place in the range studied.

(3) Pump-power performance comparisons reveal that the ribbed duct with P/e=8 has the best performance.

Acknowledgement

This work was supported by the Brain Korea

21 Project.

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