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# An Experimental Studies on Heat Transfer and Friction Factor in a Square Channel with Varying Number of Ribbed Walls

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**Abstract** : An experimental study on the heat transfer and friction characteristics of a fully developed turbulent air flow in a square channel with 45° inclined ribs on one, two, and four walls is reported. Tests were performed for Reynolds number ranging from 7,600 to 24,900. The pitch-to-rib height ratio, $p/e$, was kept at 8 and rib height-to-channel hydraulic diameter ratio, $e/D_h$, was kept at 0.0667. The heat transfer coefficient and friction factor values were enhanced with the increase in the number of ribbed walls. Results of this investigation could be used in various applications of internal channel turbulent flows involving different number of roughened walls.

**Key words** : Number of ribbed walls, Square channel, Friction factor, Nusselt number, Ribs of 45° attack angle

## Nomenclature

$A$ : Heat transfer area $[m^2]$
$AR$ : Aspect ratio, $W/H$
$C$ : Total area of channel walls $[m^2]$
$C_p$ : Specific heat [J/kg℃]
$D_h$ : Hydraulic diameter of channel $[m^2]$
$e$ : Height of rib [m]
$f$ : Friction factor, $\Delta p/[4(L/D_h)(\rho u_b{}^2)]$
$H$ : Height of test section [m]
$k$ : Thermal conductivity [W/m°C]
$L$ : Length of test section [m]
$\dot{m}$ : Mass flow rate [kg/s]
$Nu$ : Nusselt number, $h\,D_h/k$
$p$ : Roughness pitch [m], pressure $[N/m^2]$
$Q$ : Heat transfer [W]
$Re$ : Reynolds number, $\rho D_h u_b/\mu$
$St$ : Stanton number, $Nu/(Re\ Pr)$
$u$ : Bulk velocity [m/s]
$W$ : Width of test section [m]
$x$ : Distance from entrance of heated test section [m]

## Subscript

$1$ : Entrance
$2$ : Exit

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*b* : Bulk
*ra* : Average values of rough channel
*rs* : Rough wall
*sm* : Smooth wall
*ss* : Empirical equation for smooth channel
*w* : Wall

## 1. Introduction

It is well known that the ribs can break up the viscous sublayer of the flow and promote local wall turbulence that, in turn, increases the heat transfer from the rib and the smooth surfaces. In addition, a thermally conductive rib attached to the heated wall provides a greater surface area for heat transfer over that of ribless walls. In applications such as cooling of gas turbine airfoils, rib turbulators are cast mostly on two opposite sides of the cooling channels, since the heat transfer takes place from the inner walls of the pressure and the suction sides of the blade. However, in some cases, rib turbulators are cast on one side or four sides of the cooling channels. While turbine blade internal cooling has been widely studied in the past, other applications such as electronic equipment, heat exchangers, and nuclear reactors may utilize the results of enhanced internal cooling in channels with one, two, three, or all four rib-roughened walls.

Several publications have addressed the state-of-the art review of turbine blade cooling and the analysis of heat transfer and friction characteristics of the channel flow with two opposite ribbed walls[1]-[4]. The effects of flow Reynolds number and rib geometry (rib height, rib spacing, rib angle-of-attack, and rib configuration) on heat transfer and pressure drop in the fully developed region of uniformly heated square channels with two opposite ribbed walls have been investigated[1]-[2].

Further study of the combined effects of rib geometry and channel aspect ratio on the local values of heat transfer and pressure drop was also reported.[3-5] The results show that the angled ribs provide a better heat transfer performance than transverse ribs, and the lower aspect ratio (*AR*) channels perform better than the higher aspect ratio channels.

Choi et al.[6] investigated the convective heat/mass transfer characteristics and pressure drop in the square duct with Λ-and V-shaped rectangular ribs. They showed that the Λ-shaped ribs have higher heat/mass transfer coefficients than the V-shaped ribs. And Rhee et al.[7] dealt with the effect of the rib arrangements on an impingement/effusion cooling system with initial crossflow. The results represented that average heat transfer coefficients with rib turbulators are approximately 10% higher than that without ribs, and the higher values are obtained with smaller pitch of ribs.

Zhang et al.[8] investigated the effects of surface heating condition on heat transfer enhancement in square channels with hemi-circular wavy, hemi-triangular wavy tape, and twisted tape inserts, at which this study was performed about the channel with four ribbed walls.

An objective of this study is to investigate the effect of the number of ribbed walls on wall friction and the heat transfer coefficient in the square channel. In the past, most of experiments have been related to turbine blade cooling channels with two opposite ribbed walls, there is a need to find out the effect of number of ribbed walls in the rectangular channel for application of various heat exchanging system. The present study extends its base to provide experimental data for the square channels with one, two, or four ribbed walls. The square channel (3.5 cm×3.5 cm cross-section) is roughened with ribs of 45° attack angle only on the bottom wall. The flow Reynolds number for this study ranges from 7,600 to 28,000. The channel length-to-hydraulic diameter ratio, $e/D_h$, is kept at 0.0667, and the rib pitch-to-height ratio, $p/e$, is kept at 8.

## 2. Experimental Apparatus and Procedure

An experimental facility was constructed to test the augmentation technique and to provide the smooth duct reference data. Fig. 1 shows a schematic of the test rig. A 195W blower forced air through a 10.16-cm-dia flexible tube at room temperature. At the end of the heated test section, the air was exhausted into the atmosphere. The test duct, consisting of four heated parallel aluminum plates, 0.5 cm thick, as shown in Fig. 2, is of 30cm×30cm cross-section and is 270 cm long. The aluminum plates are separated by 0.08 cm thick fiber to reduce both the streamwise and circumferential heat conduction and to obtain the regional average values of the heat transfer coefficient. Square sharp- edged aluminum ribs with a height of 0.2cm($e/D_h$=0.067) and equally spaced at 1.6cm($p/e$=8) as shown in Fig. 2 are glued with silicone adhesive onto the walls of the channel. The ribs have the attack angle of 45° as shown in Fig. 3. The entire test section is enclosed by an additional 5 cm thick wood insulation. The channel walls are heated individually with electric woven heaters, which are embedded and flatly placed between the aluminum and wood plates to insure good contact. Heaters are independently controlled by a transformer and a constant heat flux to the channel walls.

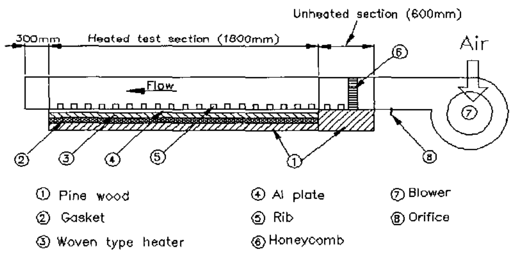


**Fig. 1 Schematic diagram of experimental setup**

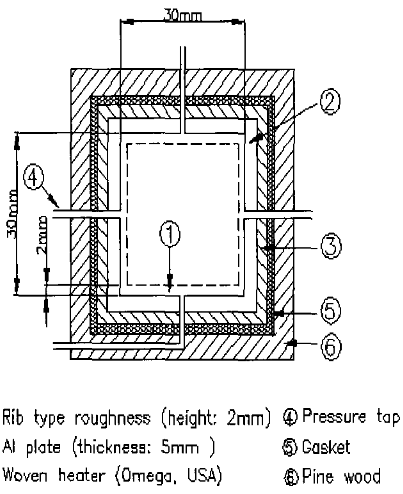


**Fig. 2 Cross-section of test section**

The thermocouples embedded in the centerline of the walls are used to measure the rib spanwise surface temperature at one downstream location in the test section for examining the temperature uniformity along the rib surface. The thermocouple beads were carefully embedded into the wall and then ground flat to ensure that they were flush with the surface. Measurement of the channel wall regional temperature is accomplished by 40 copper constantan thermocouples, which are installed along the axial centerlines of the four walls of the channel. A streamwise temperature distribution is recorded for all the ribbed and smooth walls. Temperatures of the air entering and leaving the test channel are also recorded. Five pressure taps are drilled along the axial centerline of the walls to determine the pressure drop.

A micro-manometer is used to measure the pressure differential over the specified channel length. Air flow is adjusted to obtain the desired Reynolds number. Power to the heaters is adjusted so that the temperatures of four walls as a cross-sectional location are about the same and the difference between the last section wall temperature and the exit air temperature is about 15°C. When thermal steady state is reached, air inlet and exit temperatures, atmospheric temperature, and temperatures along the channel walls are recorded. Upstream pressure and the pressure drop across the orifice, atmospheric pressure, and the pressure drop over a specified channel length are also recorded. The uniform flow in the channel is done through the honeycomb. The same procedure is repeated for the range of the Reynolds numbers and for all the rib configurations.

## 3. Results and discussion

The friction factors can be defined in terms of the pressure drop and the air mass velocity and can be calculated for a fully developed flow from

$$f = \Delta p / [4(L/D_h)(\rho u_b^2/2)] \tag{1}$$

The semi-empirical correlations developed in the past were based on the assumption that the friction factor and the rib-sided heat transfer coefficients in a channel with two smooth and two ribbed walls can be calculated from geometrically similar channels with four smooth walls and four-ribbed walls[1]-[4]. The values from four-sided smooth and four-sided ribbed walls cases are weighed proportionally to the cross-sectional widths of the smooth and the ribbed walls. There was no experimental verification of this assumption. The ribbed channel average friction factor, $f_{ra}$, is the weighted average of the four-sided smooth channel friction factor, $f_{sm}$, and the four-sided ribbed channel friction factor, $f_{rs}$.

These friction factors are weighted by the total smooth wall cross-sectional width, $C_{sm}$, and the total ribbed wall cross-sectional width, $C_{rs}$. Friction factor in a ribbed channel can be expressed as sum of:

$$f_{ra} = f_{sm}(C_{sm}/C) + f_{rs}(C_{rs}/C) \tag{2}$$

The friction factors are normalized by

the friction factors for the fully developed turbulent flow in a smooth circular tube proposed by Blasius[9] as:

$$f_{ra}/f_o = f_{ra}/(0.079Re^{-1/5}) \quad (3)$$

The local heat transfer coefficient is calculated from the local net heat transfer rate per unit area to the cooling air, the local wall temperature, and the local bulk mean air temperature as:

$$h = Q/[A(T_w - T_b)] \quad (4)$$

where heat transfer area($A$) used was always that of the smooth wall(i.e., the projected area). The heat transfer rate($Q$) was defined as:

$$Q = \dot{m}C_p(T_{b2} - T_{b1}) \quad (5)$$

Local Nusselt numbers are normalized by the Nusselt number for fully developed turbulent flow in a smooth circular tube correlated by Sieder and Tate[10] as given in Eq. (6).

$$\frac{Nu}{Nu_{ss}} = \frac{(h\,D_h\,/\,k)}{\{0.027\,Re^{0.8}\,\Pr^{1/3}\,(\mu/\mu_w)^{0.14}\}} \quad (6)$$

The fully developed heat transfer coefficient values are correlated with $Re$, as shown in Fig. 3. The correlation of previous investigators are also included. It can be seen from this figure that there is good agreement between existing correlations and the experimental results for present smooth duct. The channel average Stanton number, $St_{ra}$, is from experimental $Nu_{ra}$ data as:

$$St_{ra} = Nu_{ra}/(RePr) \quad (7)$$

The experimental uncertainties were calculated according to the procedure outlined by Kline & McClintock[11] Maximum errors in the primary measurements of $\dot{m}$, $\Delta p$, $\Delta T$, and $Q$ at the Reynolds number of 19,100 were 4.1%, 6.0%, 4.2%, and 7.8%, respectively. Based on this precision of error analysis, the maximum uncertainties in $Re$, $f$, and $Nu$ were 4.2%, 7.2% and 10.4%, respectively. Before initiating experiments with the rib-roughened surfaces, the friction factor and heat transfer coefficient were determined for smooth ducts. With the help of the pressure measurements, it was found that the developing length (i.e, the entrance section~$36D_h$) for smooth duct was long enough to guarantee the present smooth wall test duct reaching a hydro-dynamically fully developed flow, as witnessed by the measured linearity of the axial pressure distribution.

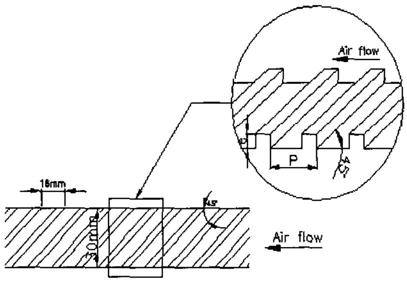


**Fig. 3 Attack angle of ribs on the wall**

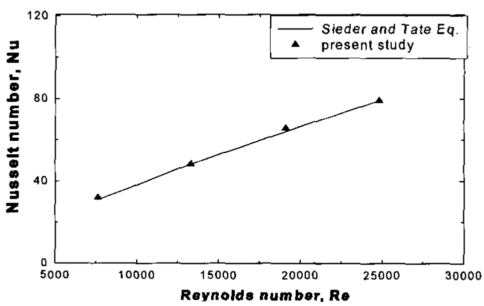


**Fig. 4 Average Nusselt numbers for smooth channel**

Fig. 4 shows the channel averaged Nusselt numbers at the smooth channel with four uniformly heated walls for the fully developed regime. For a comparison, The empirical correlation of Seider and Tade[10]was included. It can be seen from this figure there is good agreement between existing correlations[10] and the experimental results for the present smooth duct. Figs. 5~9 represent regionally averaged heat transfer coefficients, which are computed and plotted as normalized Nusselt number ratio versus normalized axial distance from the channel entrance. In these figures, rib placement on specific channel walls is shown in the frame. Letter B, L, R, and T represent bottom, left, right, and top walls, respectively. Fig. 5 represents the regionally averaged Nusselt number distributions for the a heated smooth test section. The Nusselt number ratio exhibit the highest value at the beginning of test section and decreasing monotonically with increasing downstream distance, eventually reaching a constant value in the fully developed region for a given Reynolds number. This is due to the entrance region having higher thermal boundary slope.

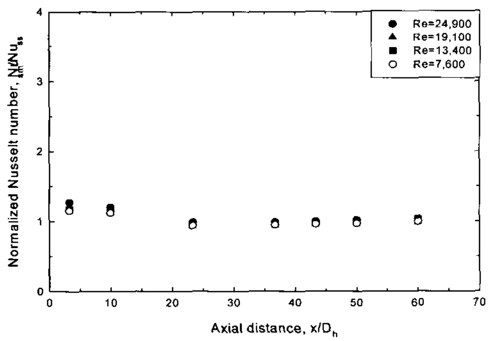


**Fig. 5 Normalized local Nusselt number for smooth channel**

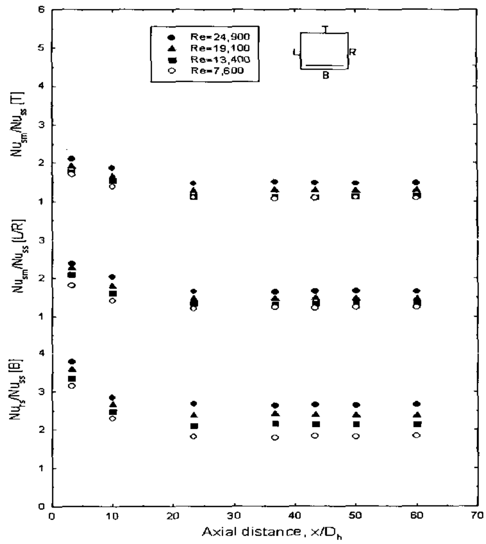


**Fig. 6 Normalized heat transfer distributions in one-ribbed wall channel**

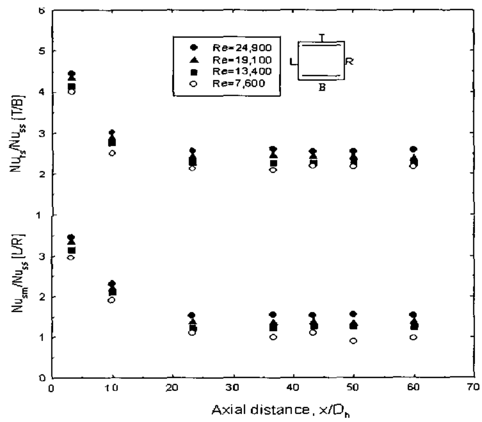


**Fig. 7 Normalized heat transfer distributions in two-ribbed wall channel**

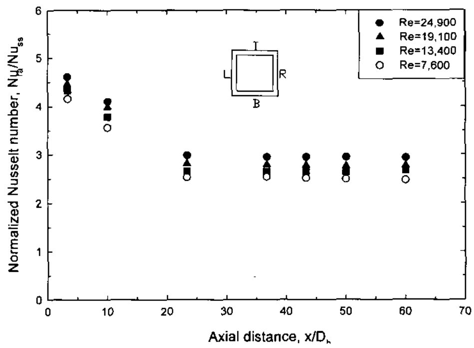


**Fig. 8 Normalized heat transfer distributions in four-ribbed wall channel**

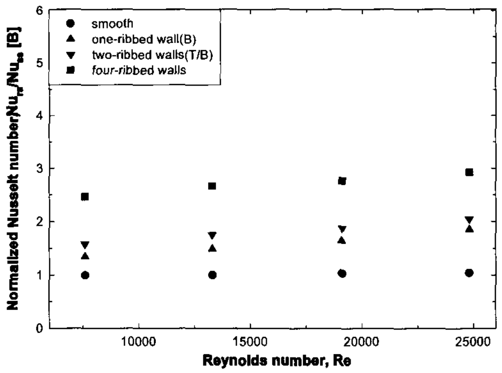


**Fig. 9 Normalized average Nusselt number rate against Reynolds number**

Figs. 6-9 are the results of the cases with one, two, and four ribbed channel walls. These figures show the regionally averaged ribbed-side and smooth-side Nusselt number distributions for $45^{\circ}$ inclined ribs versus normalized axial distance.

The results shows that for all cases, local Nusselt number ratios on ribbed as well as on smooth walls increase with increasing Reynolds number. The trend of the Nusselt number ratios is reversed to Chandra et al.[12]dealing with the $90^{\circ}$ transverse ribbed walls of channel. The discrepancy might be attributed to a difference in the attack angle of rib.

In Fig. 6, Nusselt number ratios on ribbed bottom walls of $Nu_{rs}/Nu_{ss}$[B] show a heat transfer enhancement of 1.8 to 3.8 times over the range of Reynolds numbers from 7,600 to 24,000. Nusselt number ratios on adjacent smooth sides[L/R] and opposite smooth sides[T] are in the range of 1.2 to 2.5(about 66 percent decrease from that of the ribbed side[B]) and 1.02 to 2.1(about 56 percent decrease from that of the ribbed side[B]), respectively. This trend of heat transfer enhancement with closer region to ribbed wall[B] is due to the increase in the level of turbulence generated from the wall. Fig. 7 shows Nusselt number ratios in the channel with two opposite ribbed walls. Nusselt number ratios on the ribbed walls[T/B] are higher than those on the smooth walls[L/R]. It is attributed to the same phenomenon in Fig. 6. Fig. 8 shows normalized heat transfer distributions on four-ribbed wall channel. Ribs on all four walls of the channel create maximum Nusselt number ratios that about 2.5 to 4.6 times. This trend of heat transfer enhancement with increase in the number of ribbed walls is apparently due to the increase in the level of turbulence generated in the channel. Fig. 9 represents the effects of number of ribbed walls on normalized channel averaged Nusselt numbers for the fully developed region. The heat transfer coefficients encounter greater turbulent mixing with each additional ribbed wall and thus experience higher normalized Nusselt numbers.

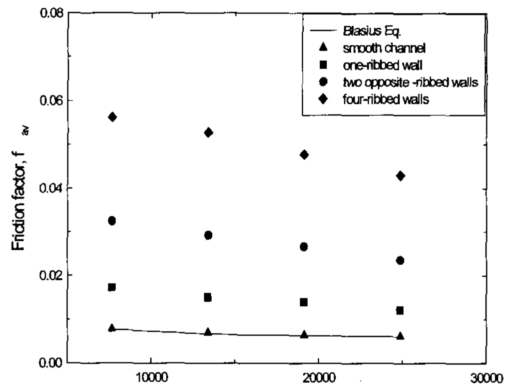


**Fig. 10 Centerlined friction factors**

Fig. 10 shows the channel averaged friction factors in the channel varying the number of ribbed walls. The channel averaged friction factors were calculated from Eq. (5) using static pressure differences measured at the pressure tap centerlined in the walls. The calculation of friction factor takes into account the temperature variation in the channel. Blasius equation[9] for smooth circular pipe is included. It can be seen from this figure that there is fairly good agreement between Blasius equation and the experimental results for the present smooth duct. The deviation is within 2.5%. The results show that the friction factors decrease with increasing Reynolds number. It is because a increase in the square of fluid velocity is higher than that of wall shear stress with increasing Reynolds number. The channel average friction factor increases with increasing the number of ribbed walls and ribs on all four walls of the channel create maximum friction factor that is about 7.0 times $f_{ss}$ for a Reynolds number of 24,900. The flow generates greater resistance with each additional ribbed wall and thus leads to higher friction.

## 4. Conclusions

The effects of number of ribbed walls on the distribution of the local heat transfer efficiency and on the friction factors in square ribbed channel were investigated for Reynolds number ranging from 7,600 to 24,900. The following conclusions are drawn:

(1) The Nusselt numbers in the channels of one, two, and four ribbed walls were 1.4 to 1.8 times, 1.6 to 2.1 times, and 2.5 to 2.9 times, respectively higher than those of fully developed turbulent flow in smooth ducts.

(2) The heat transfer coefficients of the channel with 45° inclined ribbed walls are very different from those of the channel with 90° transverse ribbed walls, exhibiting a steady decreasing behaviour.

(3) The channel average friction factor increases with increasing the number of ribbed walls and ribs on all four walls of the channel create maximum friction factor that is about 7.0 times $f_{ss}$ for a Reynolds number of 24,900.

## Acknowledgement

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