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EXPERIMENAL STUDIES ON TURBULENT FLOW IN RIBBED RECTANGULAR CONVERGENT DUCTS WITH DIFFERENT RIB SIZES

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ABSTRACT

Experimental investigations on Nusselt number and friction loss behaviors of airflow through a rectangular convergent duct fitted with square ribs of different sizes are presented here. The test section has two sloped surfaces and two parallel vertical surfaces. The ribs are placed only on the sloped surfaces of the test section. A specified heat-flux is applied to the ribbed sides of the test section and the ribs are arranged in the form of a staggered array. Measurements were carried out for a convergent rectangular channel with inlet aspect ratio of 1.25 and an exit aspect ratio of 1.35. The heights of the rib turbulators (*e*) were 6 and 9 mm. This yields a rib height (*e*) to mean hydraulic diameter of the duct (D_m) ratio of 0.0697 and 0.1046 respectively with a fixed rib pitch (*p*) to test section width (*w*) ratio of 0.6. Reynolds number based on the mean hydraulic diameter (D_m) of the channel were kept in a range of 20,000 to 60,000. The results from the ribbed ducts are compared with that of the smooth convergent duct, to estimate the heat transfer enhancements from the ribbed ducts. The comparison shows that among the tested smooth, 6 mm and 9 mm ribbed ducts, the heat transfer and pressure drop increases with the increased rib height. The local variation of thermo hydraulic performance ratio of 6mm ribbed ducts at different Reynolds numbers showed that the ratio is greater than 3 for all Reynolds numbers with CD-6 duct, which indicates that the ribbed ducts with 6 mm ribs are enhancing the heat transfer by 3 times more than the corresponding smooth ducts. The corresponding value for CD-9 ducts is close to 1.2 only. The peak values of THPR are found to be close to 9 for CD-6 duct at $X/D_m = 2.8$ for Re=20,000, whereas for the CD-9 duct it is close to 6 at $X/D_m = 2.0$ for the same Reynolds number. The values of THPR decreases with increasing Reynolds number, which indicates that the pressure drop penalty increases with increase in Reynolds number.

Keywords: Rib turbulators, Turbulent flow, Reynolds number, heat transfer, frictional factor.

1. INTRODUCTION

The application of ribs attached to in the cooling channel or a channel in heat exchanger is one of the commonly used enhanced heat transfer technique in single-phase internal flows. Several researchers are working on the fluid flow and heat transfer in ribbed channels. This heat transfer enhancement technique had been applied to various types of industrial applications such as air compressor heat exchanger, cooling system of CPU and internal cooling systems of gas turbine blades. The ribs located in the test section interrupts the hydrodynamic and thermal boundary layers. Downstream of each rib the flow separates, recirculates and impinges on the channel surfaces, which is one of main reason for heat transfer enhancement in such test section.

To obtain a higher heat transfer with a reasonable friction loss, there have been a significant amount of investigations in the last two or three decades and are reported in literature. These investigations include the effects of various geometric parameters on both local and overall heat transfer coefficients, such as channel aspect ratio, rib height, rib height-to-passage hydraulic diameter ratio, rib angle, rib pitch-to-height ratio and rib shape, the manner in which the ribs are positioned (in-line, staggered, oblique, one-wall opposite two-walls), and so on. The present experiment was conceived with an objective study the internal cooling passage of a typical gas-turbine blade. Hence the following review the references mentioned below will be mainly related to this area.

Large number of experimental studies is reported in the literature on internal cooling in turbine blade passages; particularly for square coolant passages. Han *et al.* [1] investigated the effects of rib shape, rib orientation and pitch-to-height ratio. They found that 45° ribs orientation produced better heat transfer performance than 90° rib orientation for the same friction power. Han and Park [2] varied the channel aspect ratio and concluded that the best heat transfer performance was obtained using a square channel with rib turbulators angled at 30° to 45° with respect to the flow direction.

Majority of recent research papers are mainly focused on the geometry or geometries with broken transverse, inclined, V, Z and W –shaped ribs, refer [3], [14]-[18]. Ribbed convergent/divergent square ducts were introduced by Wang *et al.* [4] and when comparing the results with that of straight ducts, showed that among these ducts the divergent duct has the highest heat transfer enhancement. Santosh and Madhukar [5] performed an experimental work on inverted U-shaped turbulators on the absorber surface of a rectangular duct used in solar heating.

Jaurker et al. [6] conducted experiments on rib-grooved artificial roughness on one heated wall of a large aspect ratio duct to find the heat transfer and friction characteristics. Web and Ramadhyani [7] conducted experiments on a channel with a constant property fluid flowing laminarly through a parallel plate channel with staggered, transverse ribs and a constant heat flux along both walls. Thianpong *et al.* [8] reported that the heat transfer and friction loss behaviors of different heights of triangular ribs with three e/H ratios (e/H =0.13, 0.2 and 0.26) and aspect ratio 10. They found that the uniform rib height performs better than the corresponding non-uniform one. Amro *et al.* [9] investigated experimentally a triangular channel with rounded edge as a model of a leading edge cooling channel for a gas turbine engine to measure heat transfer by liquid crystal method.

Chang *et al.* [10] studied the influence of channel height on heat transfer in a rectangular channel with two opposite rob-roughened walls. Ben and Jiang [11] have been compared 45° ribs on one wall rectangular channel experimentally and other rib orientation likes 90°, 60°, 45°, 30°, 20°, 0° numerically. The rectangular channel with 20° ribs has been found to be giving the best overall thermal performance.
Sivakumar et al [12] conducted a convergent channel with three different sized ribs namely 3, 6, and 9mm with Reynolds range 20 $x10^3$ to $60x10^3$. Gupta *et al.* [13] presented the local heat transfer distribution in a square channel with 90° continuous, 90° saw tooth profiled and 60° broken ribs for various pitch distant to height ratio for different position. Among these three type of ribs, heat transfer enhancements caused by 60° V-broken ribs are higher than those of 90° continuous ribs.

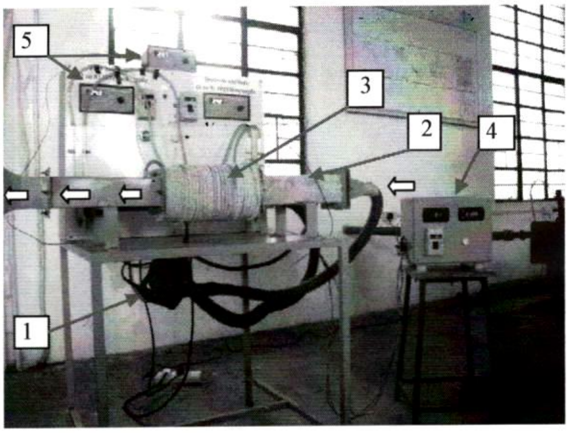
Chandra *et al.* [14] reported air flow in a square channel with transverse ribs on one, two, three and four walls, with rib height to channel hydraulic diameter ratio kept at 0.0625. Olsson and Sunden [15] reported experiments on a rectangular channel with various geometrics parameters such as cross ribs, parallel ribs, cross V-ribs, parallel V-ribs and multiple V-ribs the Reynolds number range from 500 to 15000. Ahn [16] compared of fully developed heat transfer and friction factor characteristics in rectangular ducts with ribs roughened by five different shapes. The effect of rib shape and Reynolds numbers are examined. Within five different shapes triangular type rib has a substantially higher heat transfer performance than any other ones.

Gao and Sunden [17] used liquid crystal thermography to find temperature distribution between a pair of ribs on the rib surfaces in three configurations were parallel ribs and V-shaped ribs pointing upstream or downstream of the main flow direction. Suneeta et al [18] conducted an experimental analysis of natural convection heat transfer from horizontal rectangular notched fin arrays. The fins are arranged by staggered as per this paper. Panigrahi and Acharya [19] presented experimentally the reattachment shear layer developing behind a surface-mounted rib with and without an external imposed excitation. The effect of rib spacing on heat transfer and friction in a rectangular channel with 45° angled rib turbulators on one/two walls was reported by Tanda [20] for four rib pitch-to height ratios 6.66, 10.0, 13.33 and 20.0. Tariq et al [21] numerically investigated heat transfer dissipation from a rectangular fin embedded with rectangular perforations of various aspect ratios under natural convection.

From the literature survey, it was found that not many works were reported that deals with heat transfer in ribbed roughened rectangular ducts with channel convergence and with different sized ribs. It is thus the objective of the present study to discern the flow and heat transfer characteristics in a ribbed convergent rectangular duct with two different rib sizes (6mm, 9 mm). In this present study, an experimental measurement was carried out to find the developing heat transfer and friction factor of turbulent flow in ribbed convergent stationary rectangular duct with a uniform heat flux on the ribbed walls and compare with convergent smooth rectangular duct for the thermal configuration.

2. EXPERIMENTAL SETUP

An experimental setup has been designed and fabricated to study the effect of ribs height on the heat transfer and fluid characteristics of air flow in the convergent rectangular duct. A photographic view of the experimental system as shown in Figure 1.

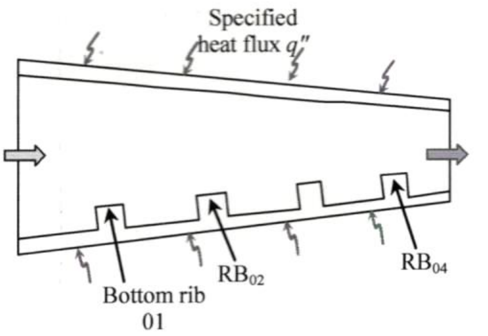


1. Variable speed blower 2. Flow stabilizer
3. Test section 4. Voltage Regulator
5. Temperature indicator

Figure 1 shows the photographic of the experimental setup

The flow system consists of an entry section, test section, exit section; It has a flow measuring orifice plate with u-tube manometer, pressure measuring u-tube manometer and a centrifugal blower with a variable regulator to control the blower speed. The convergent rectangular duct is made of copper plates with parallel vertical plates and two tapering test section surfaces, one in the top and one in the bottom with a slope of 1:100 along the length wise direction. These dimensions are similar to [4] where the experiments were conducted on a square shaped duct. The total length (l) of the test duct is three times the test section inlet width (w). The dimension of rectangular duct at entrance section is $w \times 0.8w$ and exit section of $w \times 0.74w$ respectively. The thickness of copper plates used for the duct construction is $0.05w$.

Two electrical heaters of size 280 x 90 sq.mm, fabricated by combining series and parallel loops of heating wire on 5 mm asbestos sheet were placedoutside of the ribbed top and bottom of the test duct. A 100 mm thick glass wool was applied as insulation on the ambient side of the heater and kept in place by an insulating rope wound around the test section. The heat flux was varied from 0 to 500 W/m^2 by a variac transformer. The mass flow rate of air is measured by means of a calibrated orifice plate connected with U-tube manometer using water as manometer fluid and flow is controlled by the voltage variable regulator to control the blower speed. Figure 2(a) shows the test section geometry details with ribs placed on the top and bottom converging walls. It can be observed that the four ribs in the bottom wall (RB_{01} to RB_{04}) of the test section.



(a) Rib Location – Staggered arrangement

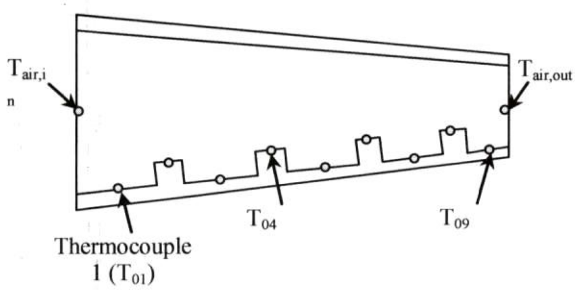


Figure 2 (b) Thermocouples location

These dimensions give the convergent rectangular channel as inlet aspect ratio of 1.25 and an exit aspect ratio of 1.35. The heights of the rib turbulators (*e*) tested were 6 and 9 mm. This yields a rib height to mean hydraulic diameter of the duct (e/D_m) ratio of 0.0697 and 0.1046 respectively. A fixed rib pitch to test section inlet height (p/h) ratio of 0.6, along the sloped surface of the test section is maintained. Reynolds number based on the mean hydraulic diameter (D_m) of the channel was kept in a range of 20,000 to 60,000.

K-type thermocouples were used to measure the temperature of air and the heated plate at different locations. A digital temperature indicator is used to display the output of the thermocouples. It is calibrated to measure the temperature within ±0.1˚C. The locations of the thermocouples used in the test section to measure the local variations are shown in Figure 2(b). The pressure drop across the test section was measured by a U-tube manometer having a least count of 0.1mm. The air is sucked through the rectangular duct by means of a blower driven by a 1-phase, 240 V, 820 W AC motor.

To get a detailed distribution of the local heat transfer coefficient, a total of 9 thermocouples are attached to the bottom surface of the test section and the ribs along its centerline to measure the surface temperature. Similarly the top surface temperature where measured by thermocouples attached to the top surface and ribs. The location of the thermocouples is dictated by the rib locations and the thermocouples at the walls, upstream and downstream of the ribs are positioned at the mid length of the corresponding walls. For the smooth duct the thermocouples are positioned at intervals of 30mm from the inlet to the outlet on the tapering wall surfaces facing the air.

3. EXPERIMENTAL PROCEDURES

Before starting the experimental investigations, all the thermocouples were checked properly so that they indicate the room temperature correctly. The test runs were conducted under steady state conditions to collect relevant heat transfer and flow friction data. The steady state condition was assumed to have been reached when the temperature do not change for about a minute. When any change in the operating conditions is made, it takes about 2 to 3 hours to reach the steady state again. Five values of flow rates were used for each fixed heat flux of the test section. After each change of flow rate, the system is allowed to attain a steady state before the data were noted again. The following parameters were measured for each testing:

(i) Temperature of the heating strip.
(ii) Temperature of air at inlet ($T_{air,in}$) and outlet ($T_{air,out}$) of the test section.
(iii) Temperatures of the ribbed bottom surfaces of the test section (T_{01} to T_9).
(iv) Mass flow across the orifice plate by using U-tube manometer.
(v) Pressure difference across the test section by using U-tube manometer.

4. DATA REDUCTION

Steady state values of the heating strip, air temperature at entrance and exit test section duct, surface and ribs temperature at the top and bottom surfaces were used to determine the useful parameters. The heat transfer coefficient was calculated from the total net heat transfer rate and the different of the local wall temperature. The friction factor, Reynolds number and Nusselt number calculated as defined below:

$$h_x = \frac{(Q - Q_{loss})}{A(T_{w,x} - T_{b,x})} \tag{1}$$

The local wall temperature $T_{w,x}$ used in Eq. (1) was read from the output of the thermocouple. The local bulk air temperature of air $T_{b,x}$ was calculated by the following equation:

$$T_{b,x} = T_{in} + \frac{(Q - Q_{loss})A(x)}{Amc_p} \tag{2}$$

Where, A(x) is the heat transfer surface area from the duct inlet to the position where the local heat transfer coefficient was determined. The local Nusselt number is given as

$$Nu_x = \frac{h_x D_m}{k} \quad (3)$$

The duct average Nusselt numbers were defined by:

$$Nu = \frac{(Q - Q_{loss}) D_m}{Ak(T_w - T_m)} \quad (4)$$

The characteristic length, the reference temperature and the average wall temperature were determined by.

$$D_m = \frac{D_{h,in} + D_{h,out}}{2} \quad (5)$$

$$T_m = \frac{T_{b,in} + T_{b,out}}{2} \quad (6)$$

$$T_w = \frac{1}{A}\int_0^A T_{w,x} dA \quad (7)$$

For most of the cases internal convective heat transfer, the fluid properties are calculated at the mean temperature of the fluid in the duct. The Reynolds number was defined by

$$\mathrm{Re}_m = \frac{U_m D_m}{\nu} \quad (8)$$

The friction factor across the entire duct of the uniform cross-section was defined by

$$f = [(\frac{\Delta p}{L}) D_m] / (\frac{\rho U_m^2}{2}) \quad (9)$$

where Δp is the pressure drop of the entire test duct. As for the convergent or divergent duct, the term of pressure loss should be complicated one; it takes the effects of acceleration or deceleration into account, refer equation 10. The average friction factor for the duct is defined by

$$f = \frac{U_{in}^2}{U_m^2} \frac{D_m}{L} \lambda \left[1 - \left[\frac{A_{in}}{A_{out}}\right]^2\right] \quad (10)$$

where λ is given below C_p, $C_{p,i}$ are the pressure recovery factors for viscous fluid and for ideal fluid respectively,

$$\lambda = 1 - \frac{C_p}{C_{p,i}} \quad (11)$$

$$C_p = \frac{P_{out} - P_{in}}{\rho U_m^2 / 2} \quad (12)$$

$$C_{p,i} = 1 - \left[\frac{A_{in}}{A_{out}}\right]^2 \quad (13)$$

This definition of C_p can be applied for $C_{p,x}$ by replacing p_{out}, p_{in} with $p_{x+\Delta x/2}$, $p_{x-\Delta x/2}$, where Δx is the distance between two neighboring pressure taps, if there are more number of wall static prssure tapes. In the present experiments, the Reynolds number varied from 20,000 to 60,000 and all other geometric parameters were kept constant. A thermo hydraulic performance ratio is estimated to find the duct performace in terms of increased heat transfer with the convergent ribbed ducts, against the corresponding increase in the pressure drop penalty. The heat transfer is adjudged by the Nu/Nu_o ratio and the pressure drop by the f/f_o ratio.

5. EXPERIMENTAL UNCERTAINTY

The method described by Moffat [22] was used to estimate the experimental uncertainty. In this study, local Nusselt numbers were determined using Eq. (4). The maximum error of the wall temperature T_w by the present method was $\pm 0.1^\circ$C, and the error of the bulk air temperature was $\pm 0.1^\circ$C. The heat flux related uncertainty is within ± 9%. Based on these values, the uncertainty of the Nusselt number was estimated to be within ± 9.2%. In the pressure drop test rig, U-tube manometer has an error of less than 1%. The least square fit process yields less than 5% standard deviation in the slope 4 (fRe), according to Box *et al.*[23] If the error induced by the thermal properties and duct dimensions are taken into account in the calculation of the Reynolds number, the friction factor uncertainty was estimated to be within ±8%.

6. RESULTS AND DISCUSSION

The result presented in this section are made simple by the symbols specified below to present the types of ducts on which the experiments were conducted: (1) CD-0: Convergent duct without ribs (smooth duct), (2) CD-6: convergent duct with 6 mm square ribs and (3) CD-9: convergent duct with 9mm square ribs. Using the data obtained from experiments, the heat transfer and friction factor characteristics of these duct are discussed below.

6.1 HEAT TRANSFER

The wall centerline Nusselt number as a function of position for Reynolds number varying between 20,000 to 60,000 of the smooth duct, 6mm ribbed and 9mm ribbed convergent ducts (CD-0, CD-6 and CD-9) are shown in Figs. 3 – 5.

Figure 3 shows the axial variation of local Nusselt number values for smooth convergent rectangular duct (CD-0) with different Raynolds number. The value of Nusselt number were found to increase with increasing Reynolds number in all cases as one can expect. For the smooth duct with variable cross section from entry to exit, the heat transfer may be regarded as fully developed at the entry of the test section, so the Nusselt number varistion is almost 10% to 20% upto Reynolds number 40000, more than that Reynolds number variation of Nusselt number was found toto be 30% to 40%.

The position of x/Dm is 1.75 where the maximum Nusselt number is reached.

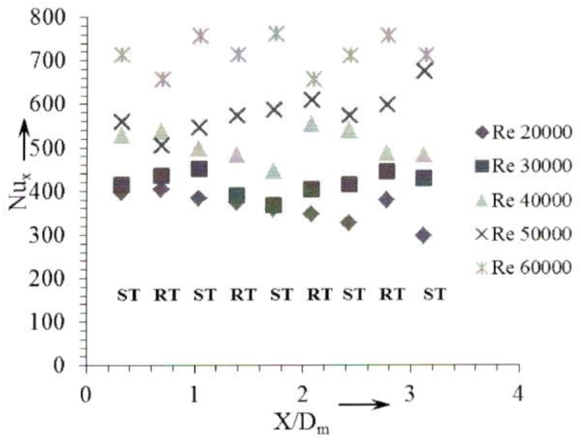


Figure 3 Axial variation of Nusselt number on the bottom wall with Reynolds number (CD-0)

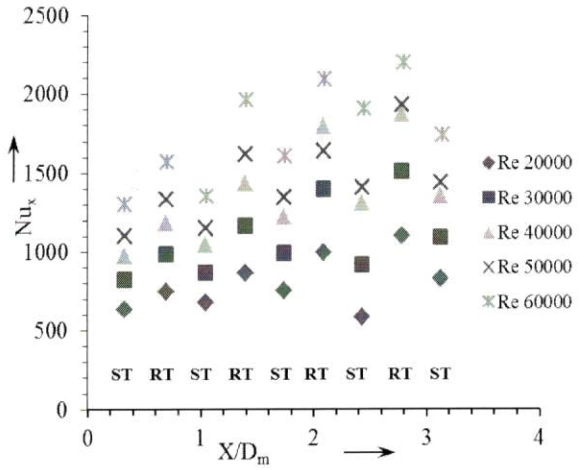


Figure 4 Variation of Nusselt number on the bottom wall with Reynolds number (CD-6)

Figure 4 bring out the effect of rib height, wherein it can be seen that the Nusselt number increase with increase in Reynolds number at the rib position. The figure shows that the local Nusselt number values along the bottom wall center line of the CD-6 channel. The increase in Nusselt number value depends on the location of the rib on the duct and reattachment of flow over the surface along the streamwise flow direction. The location of x/Dm = 2 shows the maximum Nusselt number at the Reynolds number of 60000. In rib-roughened convergent section, the heat transfer downstream of ribs was found to be lower than the values at ribs. This may be due to the situation where the flow may be going past the ribs without reattaching to the walls in between the ribs. This may be duce to the increased flow velocities in the duct ribbed section which effectively decreases the flow area.

The enhancement of the heat transfer and corresponding increased pressure drop in the ribbed ducts can be attributed to the thermal and hydrodynamic boundary layer tripping by the ribs on the walls. For convergent duct the Nusselt number found to increase with Reynolds number continuously. For

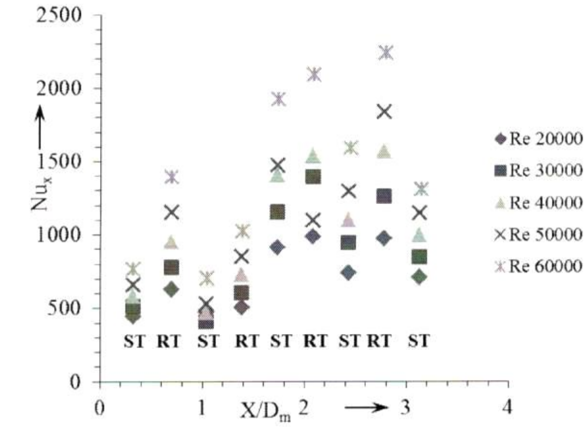


Figure 6 Variation of Nusselt number on the bottom wall with Reynolds nubmer (CD-9)

lower Reynolds number, the percentage increasing rate of Nusselt number with Reynolds number for a given location is less and its value ranges between 5% to 10%. But in case of heigher Reynolds number, the percentage increasing of Nusselt number is found to go upto 20% to 35%.

Figure 5 shows that varition of Nusselt number for 9 mm rib height as a function of position of rib inside the duct for various Reynolds number. The surface to rib heat transfer rate is suddenly changed due to more heat transfer area of the rib surface. With increased rib height of 9 mm, the effective flow area decreased further, which may causes a further increase in flow velocity in the ribbed section of the channel. This effect is profoundly observed from the Nusselt number values between the CD-9 and CD-6 channels taking any one particular thermocouple location for comparison.

6.2 FRICTION FACTOR

The Reynolds number dependences of the friction for the three types of duct as shown in Figure 6 Smooth convergent rectangular duct without rib (CD-0) showed a decrease in friction factor with the increase in Reynolds number. However, for the convergent duct CD-6 and CD-9, the friction factor increases with Reynolds number. This may be due to the decreasing cross sectional area in the converent along the flow direction, leading to an ever-increasing blockage effect of the rib on the fluid flow. It can be observed that for CD-6 channel friction factor is found to increase from 0.1 to 0.25, whereas for CD-9 duct, these values are found to be 50% to 100% higher than the corresponding values in CD-6 duct.

6.3 THERMO HYDRAULIC PERFORMANCE RATIO (THPR)

Figure 7 shows the local variation of thermo hydraulic performance ratio $(Nu/Nu_o) / (f/fo)^{0.33}$ of 6mm ribbed ducts at different Reynolds numbers. It can be observed that the ratio is greater than 3 for all Reynolds numbers with CD-6 duct, which indicates that the ribbed ducts with 6 mm ribs are enhancing the heat transfer by 3 times more than the corresponding smooth ducts, overcoming the increased pressure drop penalty created by the ribs.

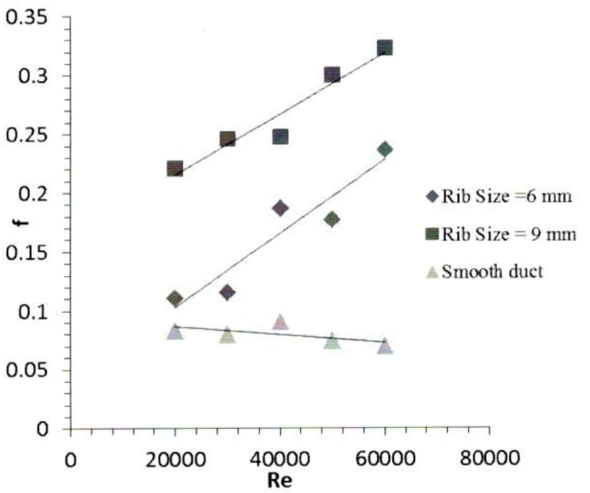


Figure 6 Variation of friction factor with Reynolds number

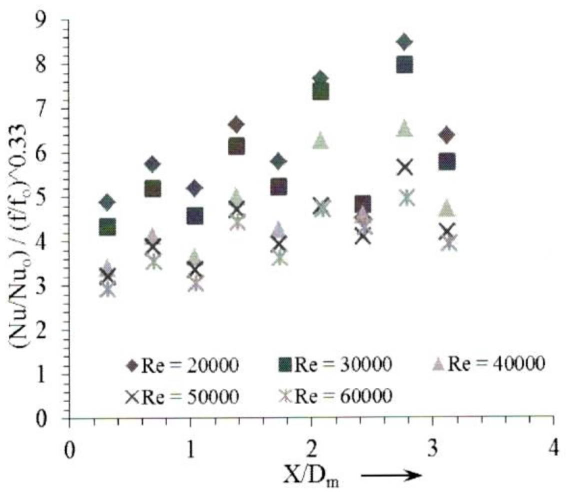


Figure 7 Thermo hydraulic performance ratio of 6mm ribbed ducts at different Reynolds number

Hence it can be confidently said that the ribbed convergent ducts are definitely advantageous in terms of thermal performance than the smoothed counterparts. The peak values of THPR are found to be close to 9 at X/D_m = 2.8 for Re=20,000. It is interesting to observe that the CD-6 ducts with lower Reynolds numbers showed higher values of THPR than the cases with higher Reynolds numbers. It is clear that the values of THPR decreases with increasing Reynolds number, which indicates that the denominator (f/fo)^0.33 increases with Reynolds number increase. This indicates that the pressure drop penalty dominates more than the increased heat transfer benefit, at higher Reynolds numbers.

Figure 8 shows the similar plot of local variation of THPR for CD-9 ducts at various Reynolds numbers. Compared to the CD-6 ducts, the CD-9 ducts showed a lower minimum values of THPR around 1.2, which indicates that the increased rib height decreases the thermal benefit obtained correspondingly. The peak values of THPR are found to be close to 6 at X/D_m = 2.0 for Re=20,000.

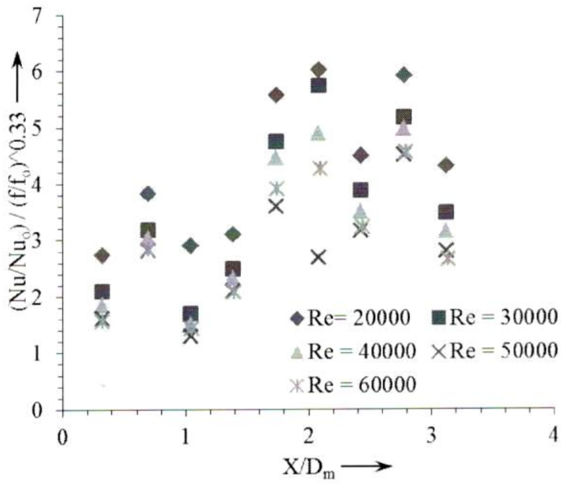


Figure 8 Thermo hydraulic performance ratio of 9 mm ribbed ducts at different Reynolds number

7. CONCLUSIONS

The work reported here is a systematic experimental study of three convergent rectangular ducts namely; smooth duct (CD-0), duct with 6 mm rib height (CD-6), and duct with 9 mm rib height (CD-9).

Under identical mass flow rate the CD-6 duct shows the large Nusselt number values and lower friction factor values compared to CD-9 ducts. The main reason for this phenomenon may be attributed to the streamwise acceleration (convergent duct) which changes the relative thickness of the thermal boundary layer and the hydrodynamic layer. Another aspect may also be due to the additional flow blockage offered by the increased rib height in CD-9 Channel.

The local variation of thermo hydraulic performance ratio (Nu/Nu_o) / $(f/f_o)^{0.33}$ of 6mm ribbed ducts at different Reynolds numbers showed that the ratio is greater than 3 for all Reynolds numbers with CD-6 duct, which indicates that the ribbed ducts with 6 mm ribs are enhancing the heat transfer by 3 times more than the corresponding smooth ducts.

The corresponding value for CD-9 ducts is close to 1.2 only. The peak values of THPR are found to be close to 9 for CD-6 duct at X/D_m = 2.8 for Re=20,000, whereas for the CD-9 duct it is close to 6 at X/D_m = 2.0 for the same Reynolds number. The values of THPR decreases with increasing Reynolds number, which indicates that the denominator $(f/fo)^{0.33}$ increases with Reynolds number increase

8. NOMENCLATURE

A surface area. m^2
A(x) surface area from inlet to the position of x, m^2
C_p pressure recovery factor
c_p heat capacity, [J Kg^{-1} K^{-1}]
D_h hydraulic diameter. m^2
D_m average hydraulic diameter, 0.086 m
e rib height
f friction factor

f_o fanning friction factor for the smooth duct
h heat transfer coefficient, $[Wm^{-2}K^{-1}]$
k thermal conductivity, $[Wm^{-1}K^{-1}]$
L axial length of duct, m
m mass flow rate, $[kgs^{-1}]$
Nu Nusselt number
$\overline{Nu}$ Average Nusselt number
Δp pressure drop of duct, $[Nm^{-2}]$
Q heat transfer rated, W
Q_{loss} heat loss to the environment, W
Re Reynolds number
Re_m Reynolds number based on D_m
T temperature, K
T_w wall temperature, K
T_b Local bulk temperature of air.
U_m Cross-section average streamwise velocity, $[ms^{-1}]$
x streamwise direction

Greek Letters:

α: Orientation of the rib, degrees
λ: parameter defined by Eq. (11)
ρ: Density of the coolant, $[kgm^{-3}]$
ST surface thermocouple
RT rib thermocouple

Subscripts:

b bulk
loss heat loss
m mean
in inlet
out outlet
w wall temperature
x local

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