# Heat transfer and pressure drop comparison between smooth and different sized rib-roughened rectangular divergent ducts

K.Sivakumar[#1], Dr. E.Natarajan[*2], Dr. N.Kulasekharan[#3]
[#]Assistant Professor, Department of Mechanical Engineering, Professor, Department of Mechanical Engineering
Valliammai Engineering College, Chennai
Saveetha Engineering College, Chennai
[1]tjksiva2k@yahoo.co.in
[3]sekarannk@gmail.com
[*]Professor, Institute for Energy Studies, Anna University, Chennai.
[2]enat123@gmail.com

**Abstract - the work reported in this paper is a systematic experimental heat transfer and pressure drop comparison between smooth and three different sized square ribbed divergent rectangular ducts. The heights of the rib turbulators (*e*) were 3, 6 and 9 mm. This yields an rib height (*e*) to mean hydraulic diameter of the duct ($D_m$) ratio of 0.035, 0.0697 and 0.1046 respectively with a fixed rib pitch (*p*) to test section inlet width (*w*) ratio of 0.6, and to maintained identical mass flow rate. The results obtained from the ribbed ducts were compared with that of the same parameter smooth (without ribs) divergent rectangular duct. The enhanced heat transfer rate for the 3 mm height rib divergent rectangular duct is more than 6, 9 mm rib height rectangular divergent duct and smooth duct. For pressure drop point of view 6 and 9 mm rib height is higher than 3 mm and smooth duct respectively.**

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

## I. INTRODUCTION

The application of ribs attaching in the cooling channel or channel heat exchangers is one of the commonly used enhanced heat transfer technique in single-phase internal flows. This heat transfer enhanced technique had been applied to various types of industrial applications such as air compressor heat exchanger (Shell and tube), cooling system of CPU and internal cooling systems of gas turbine blades. The location of ribs 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, the effect of impinges are the main reasons for heat transfer enhancement in such test section.

To obtain a higher heat transfer with a smaller or reasonable friction loss, there have been a significant amount of investigations in the last two or three decades reflecting this. These investigations, the effects of geometric parameters on both local and overall heat transfer coefficients have been discussed, such as channel aspect ratio, rib height, rib height-to-passage hydraulic diameter, rib angle of attack, rib pitch-to-height ratio and 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 mainly motivated to conduct the cooling method of a gas-turbine blade, therefore the following review the references mentioned are mainly related to this area. A large number of experimental studies are 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, angle and pitch-to-height ratio. They found that 45˚ ribs produced better heat transfer performance than 90˚ ribs 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 angle of attack from 30˚ to 45˚.

In a recent study, Majority of recent research papers are mainly focused on the geometry or geometries with broken transverse, inclined, V, Z and W – shaped ribs (see for instance, Refs. [3,4, 5, 6, 7, 14, 15, 16, 17, 18, 23, 23, 30]. The heat transfer in a rectangular channel with aspect ratio AR = 4:1 has been reported by Zhou et al.[8] for smooth surfaces and by Griffith et al. [9] for rib-roughened surfaces. Non-monotonic behavior with respect to the rotation number was observed for the smooth channel, while for the ribbed channel, the trailing surface heat transfer shows strong dependence on the rotation number. Data for 1:4 AR duct with rotation is similarly limited, with Agarwal et al. [10] recently reporting mass transfer data for smooth and ribbed ducts with rotation.

Most of the earlier computational studies on internal cooling passage of the blades have been restricted to three-dimensional steady RANS simulations *[11-12]*. The flow and heat transfer through a two pass smooth and 45˚ rib roughened rectangular duct with an aspect ratio of 2 has been reported by Al-Qahtani et al. [13] using a Reynolds stress turbulence model. They have been found reasonable match with experiments although in certain regions there are significant discrepancies. Ribbed convergent/divergent square ducts were introduced by Liang-Bi Wang et al. [14] and also comparing with this result to straight duct. The comparison shows that among the three ducts, the divergent duct has the highest heat transfer performance. Santosh B.Bopche and Madhukar S.Tandle [15] performed an experimental work has been carried out inverted U-shaped turbulators on the absorber surface of an rectangular duct similarly Jaurker et al. [16] was done on rib-grooved artificial roughness on one heated wall of a large aspect ratio duct to find heat transfer and friction characteristics. Web and Ramadhyani [17] have been done an experimentally by a constant property fluid flow laminarly through a parallel plate channel with staggered, transverse ribs and a constant heat flux along both walls. Thianpong et al. [18] and 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 using data acquisition system (fluke 2650B) and then recorded via a personal computer The uniform rib height performs better than the corresponding non-uniform one and Amro et al. [19] was done 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. Shyy woei chang et al. [20] studied influence of channel height on heat transfer in a rectangular channel with two opposite rob-roughened walls. Ben lu and Pei-xue jiang [21] has been compared 45˚ ribs on one wall rectangular channel by experimentally and numerically 90˚, 60˚, 45˚, 30˚, 20˚, 0˚. Finally the rectangular channel with 20˚ ribs had the best overall thermal performance. Abhishek Gupata et al. [22] presented 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 this three type's ribs, heat transfer enhancements caused by 60˚ V-broken ribs are higher than those of 90˚ continuous ribs. Chandra et al. [23] presented air flow in a square channel with transverse ribs on one, two, three and four walls with rib height to channel hydraulic diameter ratio was kept at 0.0625.

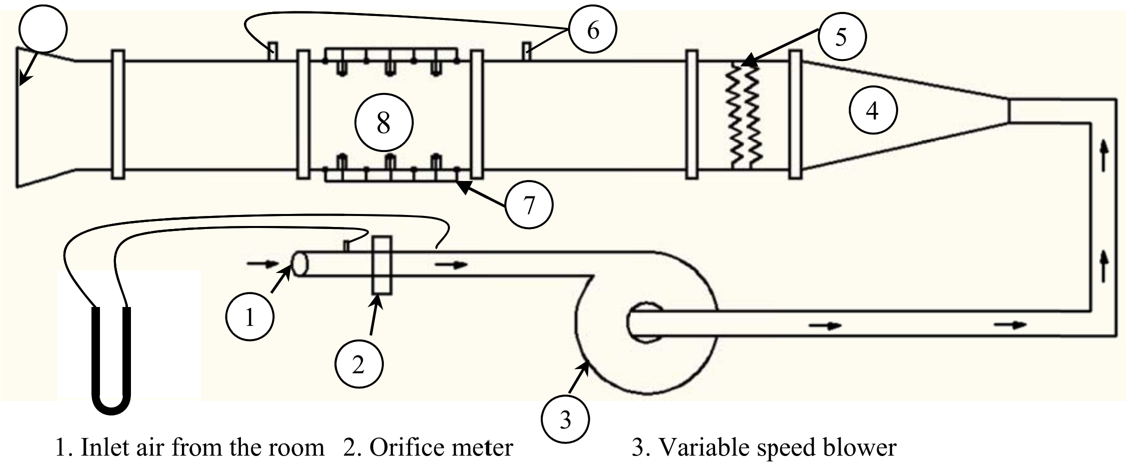
Carl olof Olsson and Bengt sunden [24] deals with a rectangular channel with various geometrics parameters such as cross ribs, parallel ribs, cross V-ribs, parallel V-ribs and multiple V-ribs the Reynolds number range from 500 to 15000. Ahn [25] comparison of fully developed heat transfer and friction factor characteristics has been made in rectangular ducts with ones roughened by five different shapes. The effect of rib shape geometries and Reynolds numbers are examined. Within five different shapes triangular type rib has a substantially higher heat transfer performance than any other ones. Xiufang Gao and Bengt Sunden [26] from this paper liquid crystal thermography were used 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. Jean-Marie Buchlin [27] steady-state heat transfer behind two dimensional perforated ribs immersed in a turbulent boundary layer are performed and to find the thermal distribution over the surface was measured by infrared thermography associated with the heated thin foil technique. Panigrahi and Acharya [28] have been presented experimentally 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 by Giovanni Tanda [29] for four rib pitch-to height ratios 6.66, 10.0,13.33 and 20.0. Wei Peng et al. [30]. Done an experimentally and numerical investigation of convection heat transfer in channels with different types of ribs and numerical simulation they are using SST k-ω turbulence model.

From the literature survey, no significant study was found that to deals with heat transfer in ribbed roughened rectangular divergent ducts with different sized square ribs. In this present study, an experimental measurement was carried out to find the developing heat transfer and friction factor of turbulent flow in ribbed divergent stationary rectangular duct with uniform heat flux boundary condition and compare with divergent smooth rectangular duct for same heat flux boundary condition.

## II. 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 divergent rectangular duct. A schematic of the experimental system is shown in Fig. 1. The flow system consists of an entry section, test section, exit section, 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 copper divergent rectangular duct made a taper of 1:100 mm inclinations along the x-direction. 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 $h \times 0.74w$ and exit section of $h \times 0.8w$ respectively. The thickness of copper duct is $0.05w$.

Two electrical heaters of size 280 x 90 sq.mm, fabricated by connecting series and parallel loops of heating wire on 5 mm asbestos sheet were placed at top and bottom of the test duct. A 100 mm glass wool was applied as insulation on the ambient side of the heater and wound around the test section. The heat flux can be 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. Fig. 2 shows the test section geometry details with ribs placed on the bottom divergent walls. It can be observed that the four ribs are placed at the bottom wall surface.




1. Inlet air from the room 2. Orifice meter 3. Variable speed blower
4. Diverging section 5. Flow stabilizer 6. Pressure drop across test section
7. Test section heaters 8. Test section 9. Exit of test section

Fig. 1 Experimental setup schematic of the divergent duct

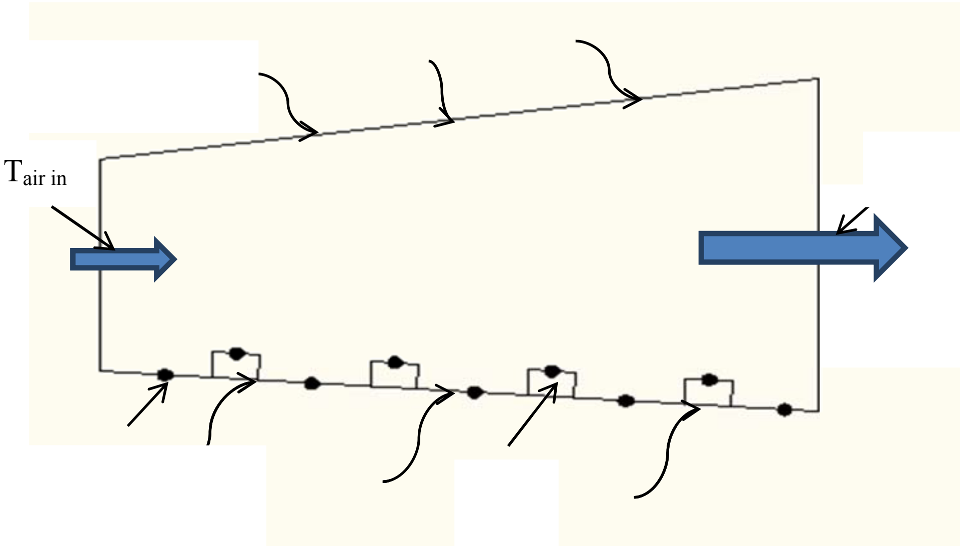



Fig. 2 Ribs and thermocouple arrangement

Measurements were carried out for a divergent rectangular channel with inlet aspect ratio of 1.35 and an exit aspect ratio of 1.25. The heights of the rib turbulators (*e*) tested were 3, 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.

The 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 of temperature as shown in Fig. 2. The pressure drop across the test section was measured by a U-tube manometer having a least count of 0.1 mm. 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. Two more thermocouple is inserted at inlet and outlet test section at the bottom of heating strip to measure the heat supplied to the test section.

## III. EXPERIMENTAL PROCEDURE

Before starting the experimental investigations, all the thermocouples were checked properly so that they indicate the room temperature. 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 at heating strip not change for about 50 seconds. When a change in the operating conditions is made, it takes about 2 to 3 hours to reach the steady state. Five values of flow rates were used for each set at a constant heat flux of the test. After each change of flow rate, the system is allowed to attain a steady state before the data were noted. The following parameters were measured:

(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 rib and bottom surfaces of the test section ($T_{01}$ to $T_{09}$).

(iv) Mass flow across the orifice plate by using U-tube manometer.

(v) Pressure difference across the test section by using U-tube manometer.

## IV. DATA REDUCTION

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

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

The local wall temperature used in Eq. (1) was read from the output of the thermocouple. The local bulk air temperature of air 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.

$$Nu_x = \frac{h_x D_m}{k} \tag{3}$$

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

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

$$D_m = \frac{D_{h,in} + D_{h,out}}{2} \tag{5}$$

$$T_m = \frac{T_{b,in} + T_{b,out}}{2} \tag{6}$$

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

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

$$\mathrm{Re}_m = \frac{U_m D_m}{\nu} \tag{8}$$

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

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

where $\Delta p$ 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 16. The average friction factor for the duct is defined by

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

where λ and the pressure recovery factors for viscous fluid and for ideal fluid respectively,

$$\lambda = 1 - \frac{C_p}{C_{p,i}} \tag{11}$$

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

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

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. In the present experiments, the Reynolds number varied from 20,000 to 60,000 and all geometric parameters were kept constant.

## V. RESULTS AND DISCUSSION

The present discussion are simplifies by the following symbols to present the types of ducts on which the experiments were conducted: (1) DD-0: Divergent duct without ribs (smooth divergent duct), (2) DD-3: Divergent duct with 3 mm square ribs (3) DD-6: Divergent duct with 6 mm square ribs and (4) DD-9: Divergent duct with 9 mm square ribs. Using the data obtained from experiments, the heat transfer and friction factor characteristics of duct are discussed as follows.

### V.1. LOCAL HEAT TRANSFER COEFFICIENT

The local heat transfer features are deeply affected by the flow structure originated by the repeated ribs at the bottom surface. The centerline Nusselt number as a function of position for Reynolds number various from 20,000 to 60,000 of the smooth duct, 3 mm, 6 mm ribbed and 9 mm ribbed divergent ducts namely (DD-0, DD-3, DD-6 and DD-9) as shown in Fig. 3-6.

Fig. 3 shows the variation of heat transfer coefficient for smooth divergent rectangular duct (DD-0) with a wide range of axial position and different Raynolds number. The value of heat transfer were found to increase with increasing Reynolds number in all case as expected. For the smooth duct with variable cross section from entry to exit, the heat transfer may be regarded as fully developed at exit of the test section, so the heat transfer coefficent varistion almost 30% to 50% upto Reynolds number 60000, the other Reynolds number; variation of heat transfer coefficent has to be 10% to 30%. The position of $x/D_m$ is 3.25 to reach the maximum heat transfer.

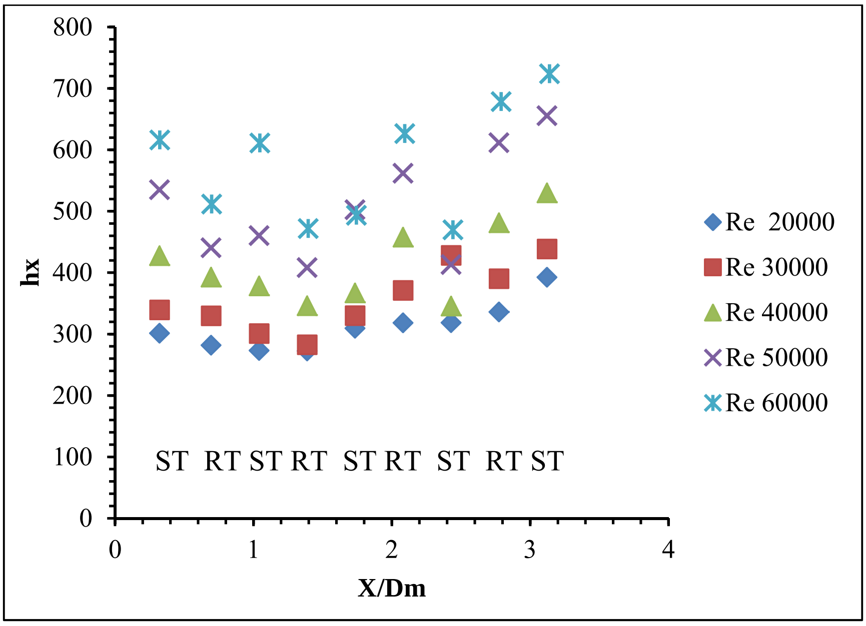



Fig. 3. Axial variation of heat transfer coefficent on the bottom wall for various Reynolds numbers (DD-0)

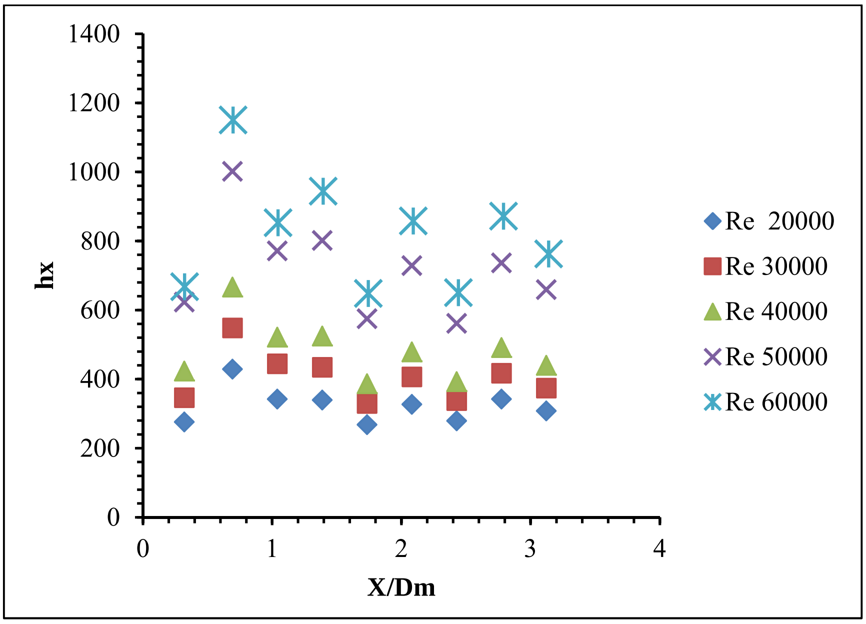



Fig. 4. Axial variation of heat transfer coefficient on the bottom wall for various Reynolds numbers (DD-3)

Fig. 4 bring out the effect of 3 mm rib height, within this, we can seen that the heat transfer decreses with increase in the rib position. The decrease in heat transfer 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 $x/D_m$ = 0.5 shows the maximum heat transfer at the Reynolds number of 60000. In rib-roughened divergent 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 increase 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 divergent duct the heat transfer found to decrease with Reynolds number continuously. For lower Reynolds number, the percentage decreasing heat transfer rate with for a given location is less and its value ranges between 5% to 10%. But in case of heigher Reynolds number, the percentage decrease in heat transfer is found to go upto 20% to 35%.

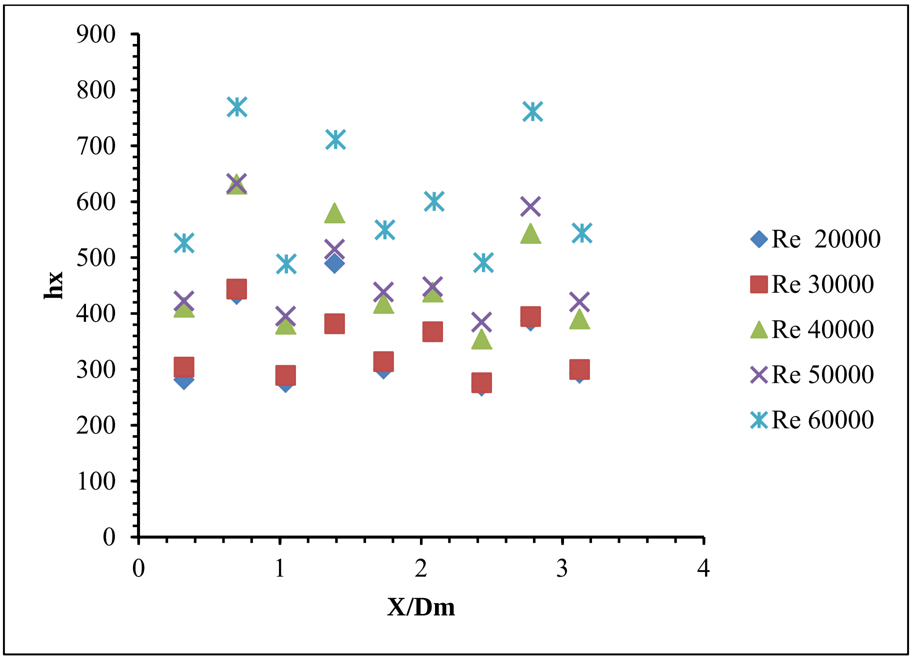



Fig. 5. Axial variation of heat transfer coefficent on the bottom wall for various Reynolds numbers (DD-6)

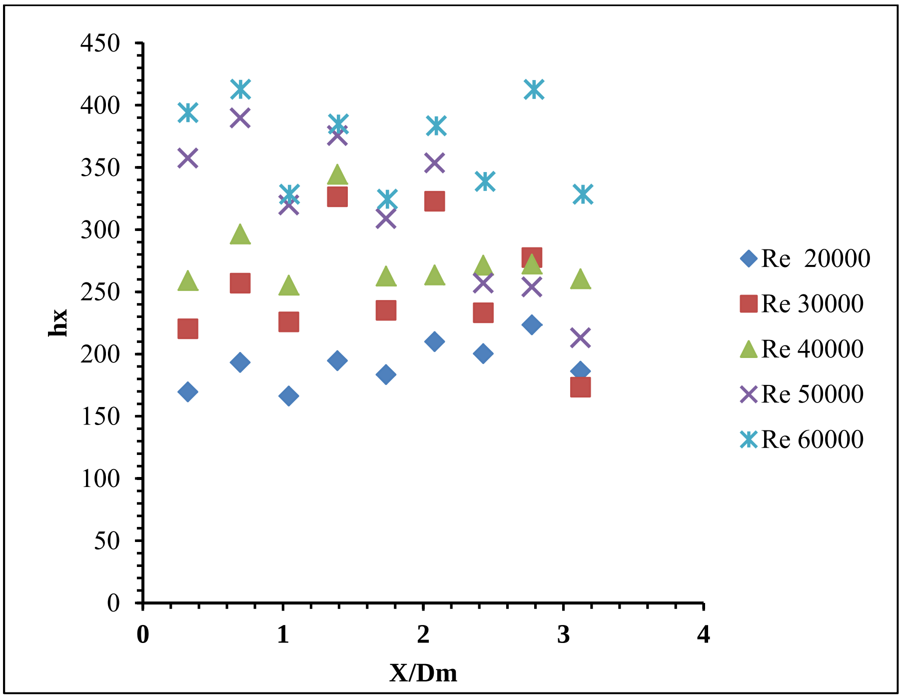



Fig. 6. Axial variation of heat transfer on the bottom wall for various Reynolds numbers (DD-9)

Fig. 5 and 6 shows that varition of heat transfer for 6 mm and 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 6 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 heat transfer coefficient values between the CD-6 and CD-9 channels taking any one particular thermocouple location for comparison.

## V.2. FULL – SURFACE AVERAGED HEAT TRANSFER COEFFICIENT FACTOR FACTOR

Since the heat transfer augmentation in rib-roughened channels is typically accompained by an increasein pressure drop, acomperhensive analysis of performance must include the impact of rib turbulators on channel pressure drop. Fanning friction factor of the ribbed channels, calculated according to Eq. 15, was normalized by the friction factor $f_o$ for fully developed turbulent flow in smooth rectangular channel proposed by Blasius:

$$f_o = 0.0791Re^{-0.25} \quad (15)$$

The friction factor ratio $f/f_o$ shown in fig. 7, decreses with increasing Reynolds number.For 3 mm rib height channel, the degree of enhancement of the friction factor, relative to the smooth wall channel. The other remaining channels 6 and 9, the friction factor ratio $f/f_o$ is greatly enhanced becoming about two times lower as compared with 3 mm rib height channel.

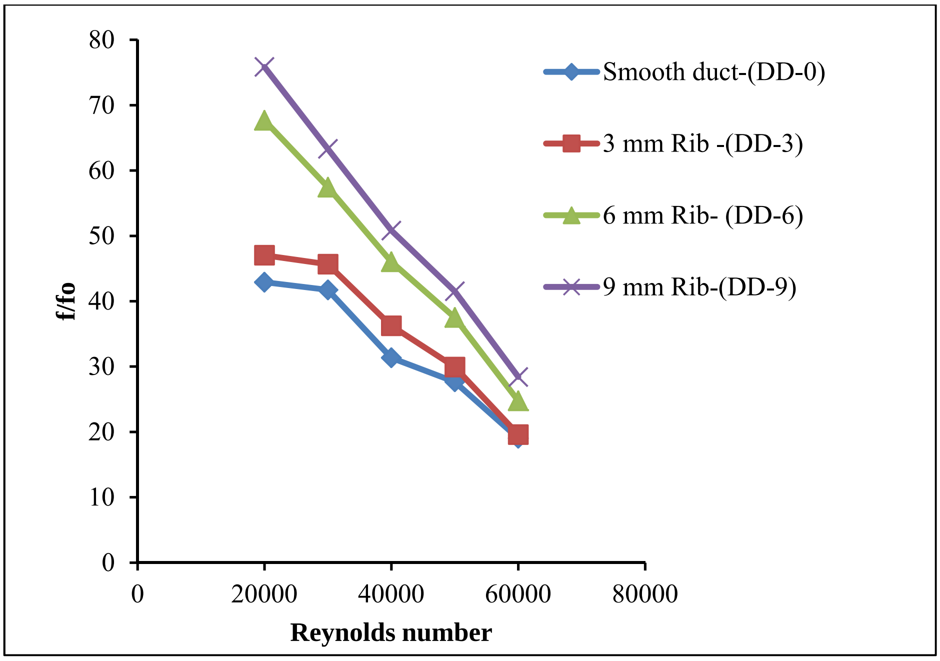



Fig. 7. Friction factor ratio versus the Reynolds number .

## VI. CONCLUSION

The work reported here is a systematic experimental study of heat transfer and friction factor of a divergent rectangular ducts with inclination angle of 1° in y-direction in three different height ribs (3, 6 mm rib height and 9 mm rib height). The Reynolds number variation range was 20 x$10^3$ to 60 x $10^3$. The heat transfer enhancement comparisons were made smooth, and 3, 6, 9 mm height ribbed divergent rectangular duct under the same mass flow rate. The 6 mm and 9 mm rib height divergent rectangular channel produced the highest friction factor then 3 mm rib height divergent rectangular channel slightly lower while the smooth divergent channel had been least value. For the entire ribbed divergent duct the heat transfer rate decreases from inlet region to outlet region. Under the constraints of comparison in identical mass flow rate the 3 mm ribbed divergent duct has the more heat transfer, and minimum friction factor compared to 6, 9 mm ribbed divergent duct.

## VII. NOMENCLATURES

| | |
|---|---|
| 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 K |
| $D_h$ | hydraulic diameter |
| $D_m$ | average hydraulic diameter, 0.086 m |
| e | rib height. m |
| f | friction factor |
| $f_o$ | fanning friction factor for the smooth duct |
| h | heat transfer coefficient, W/$m^2$K |
| k | thermal conductivity, W/m K |
| L | axial length of duct, m |
| m | mass flow rate, kg/s |
| Nu | Nusselt number |
| $Nu_s$ | Nusselt number of smooth side |
| $Nu_r$ | Nusselt number of ribbed side |
| $Nu_t$ | duct average Nusselt number |
| $\Delta p$ | pressure drop of duct, pa |
| 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, m/s

x streamwise direction

**Greek Letters:**

α: Orientation of the rib, degrees

λ: parameter defined by Eq. (11)

ρ: Density of the coolant, kg/m$^3$

ST surface thermocouple

RT rib thermocouple

**Subscripts:**

b bulk

loss heat loss

m mean

in inlet

out outlet

w wall temperature

x local

## VIII. REFERENCES


[1] Han.J.C, Glicksman. L.R., Rohsenow.W.M., 1978. An investigation of heat transfer and friction for rib-roughened surfaces, International journal Heat Mass Transfer Vol.21, Issue 7, pp.1143-1156.

[2] Han,J.C, Park.J.S., 1988. Developing heat transfer in rectangular channel with rib turbulators, International Journal of Heat Mass Transfer Vol.31, pp.183-195.

[3] Han.J.C., 1988. Heat transfer and friction characteristics in rectangular channels with rib turbulators, Journal of Heat Transfer Vol.110, pp.321-328.

[4] Wagner.J.H, Johnson.B.V, Graziani.R.A, YehF.C., 1992. Heat transfer in rotation serpectine passages with trips normal to the flow, Journal of Turbomachinery. 114, pp. 847-857.

[5] Johnson B.V, Wagner.J.H, Steduber.G.D: Yeh.F.C., 1994. Heat transfer in rotating serpentine passages with trips skewed to the flow, Journal of Turbomachinery.Vol.116, pp.113-123.

[6] Johnson.B.V, Wagner.J.H, Steuber.G.D 1993. Effect of rotation on coolant passage heat transfer, NASA contractor Report – 4396, vol. II,

[7] Chen.Y, Nikitopoulos.D.E, Hibbs. Acharya.R.S, Myrum.T.A., 2000. Detailed mass transfer distribution in a ribbed coolant passage, International Journal of Heat mass transfer Vol.43, pp.1479-1492.

[8] Griffith.T.S, Al-Hadhrami.L Han.J., 2002. Heat transfer in rotating rectangular cooling channels with angled ribs, Journal of Heat Transfer Vol.124, pp. 617-625.

[9] Agarwal.P, Acharya.S, Nikitopoulos.D.E., 2003. Heat transfer I 1:4 rectangular passages with rotation, Journal Turbo machinery, Vol.125, pp.726-733.

[10] Rigby.D.L, Steinthorsson.E, Ameri.A.A., 1997. Numerical prediction of heat transfer in a channel with ribs and bleed, ASME Paper 97-GT-431.

[11] Lacovides.T.Bo, Launder.B.E., 1995. Developing buoyancy modified turbulent flow in ducts rotating in orthogonal mode, Journal of Turbomachinery 117 pp.474-484.

[12] Lacovides.H., 1988. Computation of flow and heat transfer through rotating ribbed passage, International journal of Heat Fluid flow, Vol.19, pp.393-400.

[13] Qantani.M.Al, Chen.H.C, Han.J.C., 2003. A numerical study of flow and heat transfer in rotating rectangular channels (AR = 4) with 45˚ rib turbulators by Reynolds stress turbulence model, Journal of Heat Transfer Vol.125 (1), pp.19-26.

[14] Wang.L.H, Tao.W.Q, Wang.Q.W, Wong.T.T., 2001. Experimental study of developing turbulent flow and heat transfer in ribbed convergent/divergent square duct, International Journal of Heat and fluid flow, Vol. 22, pp. 603-613.

[15] Santosh B. Bopche., Madhykar S.Tandale., 2009. Experimental investigations on heat transfer and frictional characteristics of a turbulators roughened solar air heater duct, International journal of Heat and Mass Transfer, Vol.52, pp. 22834-2848.

[16] Jaurker.A.R., Saini.J.S., Gandhi.K.B.,2006. Heat transfer and friction characteristics of rectangular solar air heater duct using rib-grooved artificial roughness, Solar Energy, Vol.80, pp.895-907.

[17] Webb.B.W., Ramadhyani.S.,1985. Conjugate heat transfer in a channel with staggered ribs, International journal Heat and Mass Transfer, Vol.28, No.9, pp. 1679-1687.

[18] Thianpong.C., Chompookham.S.Skullong, Promvonge.P., 2009. Thermal characterization of turbulent flow in a channel with isosceles triangular ribs, International Communications in Heat and Mass Transfer Vol.37, pp.712-717.

[19] Amro.M., Weigand. B., Poser. R., Schnieder.M.,2007. An experimental investigation of the heat transfer in a ribbed triangular cooling channel, International Journal of Thermal Science, Vol.46, pp.491-500.

[20] Shyy Woei Chang., Tong-Miin Liou., Wei-Chen Juan., 2005. Influence of channel height on heat transfer augmentation in rectangular channels with two opposite rib-roughened walls, International Journal of Heat and Mass Transfer, Vol.48. pp. 2806-2813.

[21] Ben Lu., Peo-Xue Jiang., 2006. Experimental and numerical investigation of convection heat transfer in a rectangular channel with angled ribs, Experimental Thermal and Fluid Science, Vol.30, pp. 513-521.

[22] Abhishek Gupta., Sriharsha.V., Prabhu.S.V., Vedula. R.P., 2008. Local heat transfer distribution in a square channel with 90˚ continuous, 90˚ saw tooth profiled and 60˚ broken ribs, Experimental Thermal and fluid Science, Vol.32, pp. 997-1010.
[23] Chandra.P.R., Alexander. C.R., Han.J.C., 2003. Heat transfer and friction behaviours in rectangular channels with varying number of ribbed walls, International Journal of Heat and Mass Transfer, Vol. 46, pp. 481-495.
[24] Carl-olof Olsson, Bengt Sunden., 1998. Experimental study of flow and heat transfer in rib-roughened rectangular channels, Experimental Thermal and Fluid science Vol.16, pp.349-365.
[25] Ahn.S.W., 2001. The effect of roughness types on friction factors and heat transfer in roughened rectangular duct, International Communication Heat Mass Transfer, Vol.28.No.7, pp.933-942.
[26] Xiugang Gao., Bengt Sunden., 2001. Heat transfer and pressure drop measurements in rib-roughened rectangular ducts, Experimental Thermal and Fluid Science, Vol.24, pp.25-34.
[27] Jean-Marie Buchlin., 2002. Convective heat transfer in a channel with perforated ribs, International Journal of Thermal Science, Vol.41, pp. 332-340.
[28] Panigrahi. P.K., Acharya. S., 2005. Excited turbulent flow behind a square rib, Journal of Fluids and Structures, Vol.20, pp.235-253.
[29] Giovanni Tanda., 2011. Effect of rib spacing on heat transfer and friction in a rectangular channel with 45˚ angled rib turbulators on one/two walls, International Journal of Heat and Mass Transfer, Vol.54, Issues 5-6, pp.1081-1090.
[30] Wei Peng, Pei-Xue Jiang, Yang-Ping Wang, Bing-Yuan Wei., 2011. Experimental and numerical investigation of convection heat transfer in channels with different types of ribs, Applied Thermal Engineering, Volume 31, Issues 14-15, pp. 2702-2708.
[31] Bonhoff.B., Parneix.S.,Leusch.J., Johnson.B.V., Schabacker.J., Bolcs..A., 2001. Experimental and numerical study of developed flow and heat transfer in coolant with 45 degree angled ribs by Reynolds stress turbulence model, ASME J.Turbomach. 123,124-132.