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Numerical Investigation on Heat Transfer Enhancement of Traverse Ribs in 3D Turbulent Duct Flow

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Abstract: In this study, investigation on hydrodynamic and forced convection heat transfer in a rectangular horizontal duct have been studied. Heat sources were cross rectangular ribs with small aspect ratio and uniform heat flux under turbulent regime. The purpose of this study is application of a passive method to increase rate of heat transfer from the ribs. Geometry and the physics of the problem are similar to cooling of electronic boards. Therefore three rectangular ribs established along the width of the channel with specified distance from each other. Between ribs some vortexes were appeared which in general were acted as heat traps and thus reduced heat transfer rate. These thermal resistances should be neutralized by applying heat transfer enhancement methods. Due to low pressure of these areas in comparison with their surrounding environment, establishing holes between the ribs is an appropriate method for vanishing the heat traps. Actually by applying holes between the ribs, some distortions in the vortex patterns are made and an enhancement in the heat transfer due to existence of secondary flows, are made subsequently. These phenomena are occurred without any outsource energies. This method classified as passive method. Numerical simulation for assumed geometry is performed by solving governing equations in finite volume with Phoenics software. The obtained simulated results indicate good agreements with experimental investigations which have been reported by other researchers. Comparison between plain and passive cases shows that performance evaluation criteria (PEC) is highly dependant on the holes geometric parameters, specially their numbers and arrangements. Nine different arrangements for holes with same number of holes were studied comprehendly in this research.

Key words: Heat transfer enhancement, forced convection in rectangular duct, rectangular ribs, passive cooling

INTRODUCTION

Cooling of hot electronic boards surfaces with air flow is one of the common methods in practical. All of the electronic boards are constructed from small and tiny elements which generate considerable amount of heat. The main cooling systems for these equipments is flowing of the air over them through channels and reducing of generated heat by forced convection. Now a days, due to using a lot of different elements in boards and enhancement in the compactions of these elements in one board to reduce the sizes, the cooling has became a vital and inevitable process for these equipments. Thus using of optimum active and/or passive cooling systems are preferred to enhance performance evaluation criteria (PEC) of boards (Zimparov *et al*., 2006; Peterson and Ortega, 1990). Passive methods are well known for their ability to work without any outsource energies. The present study is based on a special passive method.

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Abdel-Rahman *et al*. (1992) investigated experimental and numerical investigations on flow and heat transfer in a rectangular duct with injection of secondary flow. They investigated the effects of air injection on heat transfer of rectangular ducts through a porous media.

Sultan (2000) reported the effects of holes arrangements between rectangular ribs in channels. He indicated an enhancement in heat transfer by inducing distributions in vortexes due to existence of holes behind the ribs. But in analysis of fluid motion in wake region he considered the effect of density variation and buoyancy forces. According to the negligible Richardson number of flow buoyancy forces contribute minor role in this phenomenon. In addition in definition of critical Reynolds a specified characteristic length should be established which is not appropriate with geometry and physical model of this case. Applying repeated ribs and their distances from each other in channel or in external flow on plain surfaces can influence pattern of flow field.

Webb *et al*. (1971, 2000) investigated on some experimental studies on cross barriers and their obtained flow patterns for distances between ribs. They showed that if the ratio of ribs distance on ribs height (S/b) is less than 8, produced vortexes will fill the whole area between ribs. They claimed that by increasing the distance, reattachment of flow occurred in the space between the ribs and then heat transfer coefficient in the vicinity of reattachment point reached to its maximum value.

Arman and Rabas (1991, 1992) performed a numerical study on turbulent flow with 2D rectangular ribs in vicinity of wall using k-e and two layer models (Chen and Patel, 1988). Computations were made for zones near wall and core of flow. They predicted reattachment position with high accuracy. Their results were in good agreement with experimental results made by other scientist for Nusselt and friction coefficients.

Liou *et al*. (1993, 2002) investigated the effects of cross ribs on heat transfer via numerical simulation. They also consider the effects of a rectangular duct flow with ribs containing holes. They claimed that using punctured ribs will deduce more PEC values in comparison with the plain ribs without holes.

Leung *et al*. (1999) applied passive method for increasing heat transfer rate on a plain surface with cross ribs established in a wind tunnel as an experimental approach. Due to instability of recirculation zones, with the punched holes between ribs heat transfer rate increased. The Reynolds number was changed from 500 to 20000 by varying channel height from top of the testing ribs to the roof of the channel and discharge rate. They achieved a correlation between obtained heat transfer and friction coefficients which depended to these Reynolds numbers. They focused on variation of Reynolds number and holes diameter. The measurements were estimated for inlet and outlet of the channel and no information was given for fluid behavior inside the channel.

Sara *et al*. (2001) reported momentum and heat transfer enhancement over a flat surface in a channel flow due to perforated rectangular cross-sectional ribs attached on its surface. They developed correlation equations for the average Nusselt number and friction factors. They were able to recovery energy lost by perforations opened in the ribs by which they achieved energy gains up to 40%.

Buchlin (2002) reported an experimental study on convective heat transfer in a channel with perforated ribs. Effect of the rib design on the heat transfer was investigated the influence of the rib-to-rib spacing the open area factor and channel Reynolds number on the enhancement have been emphasized. Compared to solid rib, a local thermal enhancement factor of 3 can be expected by this passive method.

Moon and Lau (2003) experimentally investigated the hydrodynamics and heat transfer between two ribs with holes for turbulent flow in a rectangular duct. For different staggered arrays of holes in the ribs and Reynolds numbers up to 30000. They obtained average heat transfer coefficient, local heat transfer distribution and overall pressure drop across the ribs. They concluded the local heat transfer distribution was strongly dependent on the configuration of the holes array in the ribs.

Chandra *et al*. (2003) made an experimental study of hydrodynamics and heat transfer of a fully developed turbulent air flow in a square duct with traverse ribs on one, two, three and four walls. The results of the investigation could be used to predict the friction factor and heat transfer coefficient in a rectangular duct with varying number of ribbed walls. In this study pitch to ribs height ratio, S/b, was kept at 8 and rib-height to channel hydraulic diameter ratio, b/D_h, was kept at 0.0625. It seems this passive method is concentrated on roughened surface and the traverse ribs have the dimension around 0.5 mm in height and thickness, so it could not possible to consider them as large ribs for analyzing the behavior of air flow in relation to the hydrodynamics and heat transfer phenomena. Their correlations also could be used only in the domain of their experimental data. State of the art of this study in comparison with others is the effects of secondary flow caused by drilling holes in floor of channel between ribs to disturb the recirculation zones. It is emphasized again, the effect of pressure gradients induced by drilling holes between ribs causes to increase heat transfer rate. For a two-dimensional rib on a flat plate the experimental data of researchers show that the heat transfer coefficient attains its maximum value near the reattachment point between successive similar ribs. In our study the flow patterns according the dimensionless rib spacing (S/b) near unit contain only one stable recirculation zone. Therefore, this zone appears as trap of heats and resists to the heat transfer.

Lau *et al*. (2003) conducted experimental study of heat (mass) transfer enhancement by blockages with staggered round and square holes for turbulent air flows through a wide rectangular channel. Their results showed that the blockages enhanced the average heat (mass) transfer on the channel walls by 4.7-6.3 times than for fully developed turbulent flow through a smooth channel. Obviously, the dissimilarity between heat (mass) and momentum transfer is high in this passive enhancement method.

Duttaa and Hossainb (2004) investigated the local heat transfer and hydrodynamics phenomena in a rectangular duct with inclined solid and perforated ribs. They also showed that the local Nusselt number distribution is strongly depended on geometry parameters of ribs.

Changa *et al*. (2008) reported an experimental investigation about hydrodynamics and heat transfer in rectangular channel fitted a novel heat transfer enhancement (HTE) roughness with V-shaped ribs and deepened scales. They used both forward and backward flows in the Re range with laminar and turbulent flows. Relative to the smooth-walled duct, HTE ratios of these authors with forward and backward flows, respectively, reach 9.5-13.6 and 9-12.3 for laminar flows and 6.3-6.8 and 4.3-5.7 for turbulent flows.

Frequently, in thermal systems increase in heat transfer rate causes decrease in momentum transfer. This phenomenon literally named dissimilarity. At first by applying ribs in channel, local pressure and momentum both drop. Therefore for a specified inlet velocity situation more pumping power is needed. On the other hand due to local effects such as disturbance and extending of contact surface, heat transfer increase. So it is necessary to modify a convenient similarity between these two opposite phenomena. Punching of holes in suction channel can be used to diminish the dissimilarity drastically.

Drilling same holes in the space between the ribs on the bottom of the channel results a pressure gradient from environment toward channel which itself makes a secondary cross flow to the main channel flow. Therefore the set up of holes between ribs will change the flow pattern. By using suitable arrangements for them on the channel the heat transfer can be enhanced.

In the present study for avoiding from immoderate changes in physical properties of fluid which causes uncoupled solving of momentum and energy equations, the temperature differences between fluid and rib was restricted. Also we examined 9 different arrangements of holes between ribs and the best arrangement is distinguished based on computation results (Fig. 1).

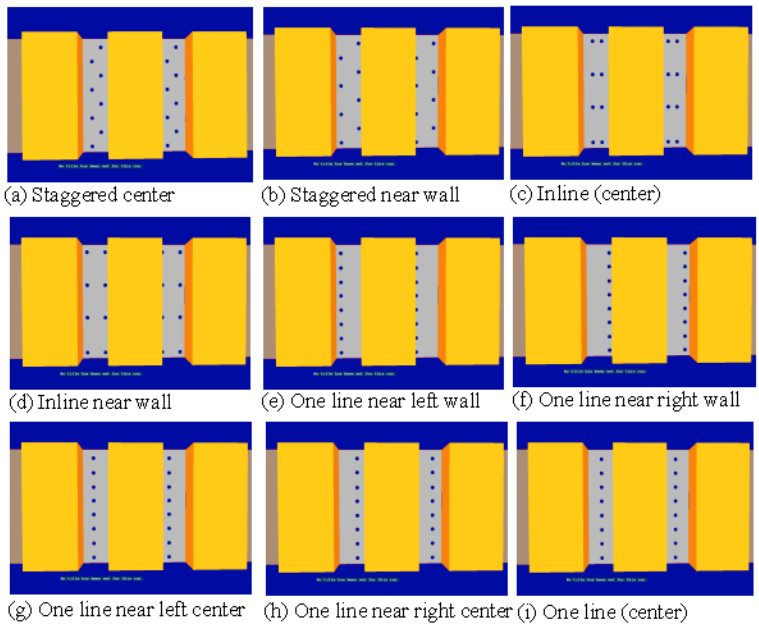

Fig. 1: Types of holes arrangements

THEORETICAL APPROACH

A schematic layout of physical model is shown in Fig. 2, where investigation region is a rectangular duct with 3 rectangular traverse ribs located on the floor with uniform heat flux and small aspect ratio. Channel is 570.47 mm in length and 120×120 mm in its cross section. The ribs assumed to be 16×27×120 mm and located 15 mm from each other and heated with uniform flux. Air flow enters the channel from right side by a suction producing a turbulent flow. The efficiency is a function of diameter, number, suction angle (90°in this study) and arrangement of the holes. Meanwhile, this passive method is presented with parameter ß which is the area of holes to the side surface area of ribs. So it is defines as:

$$\beta = (n\pi d^2/4)/(Wb) \tag{1}$$

where, n and d are number and diameter of holes, W is channel width and b is the height of the ribs. In this study for 2 mm diameter of holes $\beta = 0.02618$. In Fig. 2a the plain case of duct with ribs is shown and in Fig. 2b the test section is presented for passive case.

The pitch to rib height ratio was kept at 0.9375 and rib height to channel hydraulic diameter ratio, $b/D_h = 0.13$ and duct length to hydraulic diameter ratio, L/D_h was 4.75.

Before introduction the basic conservation equations of problem it is required to introduce some of parametric correlations:

$$\theta = f(Re_L, H/L, b/L, S/L) = f(Re_{D_h}, b/D_h, S/D_h, L/D_h) \tag{2}$$

Where:

$$\theta = [(T_i - T_0)/(q_W L/k)] \tag{3}$$

$$Re_L = \frac{u_o L}{\upsilon} \text{ or } Re_{D_h} = \frac{u_o D_h}{\upsilon} \tag{4}$$

Reynolds numbers based on width of ribs, Re_L and hydraulic diameter of duct Re_{Dh}.

$$Gr = g\beta_1 q_W (L + 2b)L^3/(K\upsilon^2) \text{ (Grashof No)}, \; Ri = \frac{Gr}{Re_L^2} \text{ (Richardson No)} \tag{5}$$

where, T_o, u_o, K, υ, β_1, q_w and T_i are fluid inlet temperature, velocity inlet, coefficient of conductivity for air, kinematics viscosity of air, thermal expansion coefficient of air, constant heat flux over ribs and local surface temperature in floor of duct respectively. Re_L and Re_{Dh} represented Reynolds numbers for length of ribs and hydraulic diameter of channel. In this study Reynolds number assumed to be $376 \le Re_L \le 6170$ or $1670 \le Re_{Dh} \le 27404$ and Richardson number variation is restricted to $1.733\times10^{-10} \le Ri \le 6.436\times10^{-10}$. By considering the fact that $Ri << 1$ in this situation, therefore the heat is only transferred by forced convection.

Local Nusselt number, Nu_{hx}, along the surface of ribs defines in terms of wall surface temperature T_i and wall heat flux q_w as follow:

$$Nu_{hx)Passive} = hL/K = q_W L/(T_i - T_o)K \tag{6}$$

Average Nusselt number defined by average rectangular ribs surface temperature:

$$Nu_{hx)Plain} = q_W L/(\overline{T}_i - T_o)K \tag{7}$$

$Nu_{hx)passive}$ and $Nu_{hx)plain}$ are related to passive and plain cases respectively. The ratio of heat transfer enhancement expresses as:

$$\%Enhancement = Nu_{hx)Passive}/Nu_{hx)Plain} \tag{8}$$

The efficiency index of enhancement, η_e, is the heat transfer enhancement defined as:

$$\eta_e = \frac{j/j_s}{f/f_s} = j/j_s \times f_s/f \tag{9}$$

where, the subscript s indicate the plain case, f skin friction factor and j Colburn heat transfer coefficient and the relationship between j and f is provided by Reynolds analogy as:

$$St.Pr^{2/3} = j = f/2 \tag{10}$$

For the steady state of fan flow, instability of flow is negligible. So:

$$\frac{f_s}{f} = \frac{\Delta P_s}{\Delta P} = \frac{P_{OWS}}{P_{OW}} \tag{11}$$

where, P_{ow} is the power of sucked flow. Finally the efficiency index as a performance evaluation criteria (PEC) becomes (Webb, 2005):

$$\eta_e = \frac{Nu}{Nu_s} \times \frac{P_{OWS}}{P_{OW}} \tag{12}$$

CONSERVATION EQUATIONS AND BOUNDARY CONDITIONS

Fundamental equations for three dimensional, steady, incompressible and turbulent flow are as follow:

Continuity Eq.

$$\frac{\partial \bar{u}_j}{\partial x_j} = 0 \tag{13}$$

Momentum Eq.

$$\rho(\bar{u}_j \frac{\partial \bar{u}_i}{\partial x_j}) = -\frac{1}{\rho}\frac{\partial \bar{P}}{\partial x_i} + \frac{\partial}{\partial x_j}(\mu \frac{\partial \bar{u}_i}{\partial x_j} - \rho \bar{u}_i' \bar{u}_j') \tag{14}$$

Energy Eq.

$$(\bar{u}_j \frac{\partial \bar{T}_j}{\partial x_i}) = \alpha \frac{\partial^2 \bar{T}_i}{\partial x_i^2} - \frac{\partial}{\partial x_i}(u'T') \tag{15}$$

Turbulent transport equations (Warsi, 2006):

k Eq.

$$\bar{u}_j \frac{\partial k}{\partial x_j} = P - \varepsilon + \frac{\partial}{\partial x_j}(\frac{\nu_T}{\sigma_k}\frac{\partial k}{\partial x_j}) + \nu \nabla^2 k \tag{16}$$

e Eq.

$$\bar{u}_j \frac{\partial \varepsilon}{\partial x_k}(C_e \frac{k}{\varepsilon}\bar{\tau}_{kl}\frac{\partial \varepsilon}{\partial x_l}) + C_{e1}\frac{\varepsilon P}{k} = -C_{e2} f_e(R_T)\frac{\varepsilon}{k}\left[\varepsilon - 2\nu(\frac{\partial \sqrt{k}}{\partial n})^2\right] + \nu C_{e3}\bar{\tau}_{ij}\frac{k}{\varepsilon}\frac{\partial^2 \bar{u}_k}{\partial x_i \partial x_l}\frac{\partial^2 \bar{u}_k}{\partial x_j \partial x_l} + \nu \nabla^2 \varepsilon \tag{17}$$

In which:

$$f_e(R_T) = 1 - \frac{2}{9}\exp(-R_T^2/36)$$
$$R_T = k^2 \nu \varepsilon$$

For numerical study, as shown in Fig. 2, some boundary conditions were assumed including:

- The pressure inlet boundary condition was considered for right side
- Fan boundary condition was considered for left side
- The walls of the channel, except rectangular ribs are assumed to be adiabatic and insulated
- A uniform heat flux was applied for the surfaces of the ribs
- On the middle of width of the channel a symmetric condition was applied
- For holes in passive case, pressure inlet boundary condition assumed
- The operation conditions are 25°C and 100 kPa

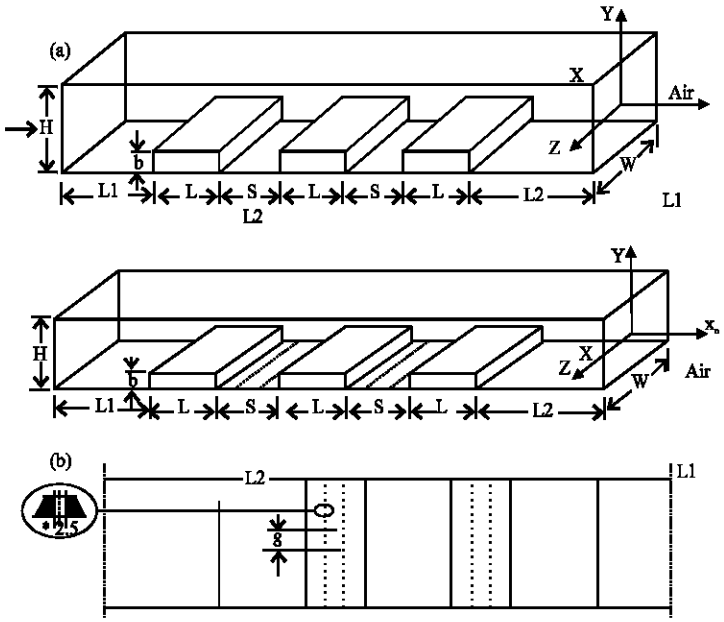

Fig. 2: Schematic geometry of main problem (a) Plain case and (b) Passive case

SOLUTION PROCEDURE

Numerical simulation was carried out for both plain and passive cases. In passive case, the holes between successive ribs have been established for heat transfer enhancement. Angle of holes from main direction of duct, diameter of holes, number of holes and their geometry arrangements on surface have strong effects on heat transfer enhancement. In this study the diameter and the number of holes between ribs has been taken constant (2 mm and 16) and the holes have right angle. Therefore the geometry arrangement of holes has been considered variables in these investigations. For this purpose nine different arrangements were undertaken for numerical computations. As it is shown the nine geometry arrangements are named a to i and have been selected for simulation.

The conservation equations of fluid flow with 2 equations of turbulent transport models were numerically solved with finite volume method. The computational grids are staggered with regular topology (Fig. 3a, b). The numerical simulation for three dimensional mean steady turbulent flow for our cases has been supplied by means of the PHOENICS software.

The continuity, momentums, energy equations and the modified k-ε turbulent transport equations were discritized by hybrid scheme. Then these equations were solved with TDMA and SIMPLE algorithms.

The Reynolds numbers based on ribs width have been considered in the range of $376 \leq Re_L \leq 6170$. In channel flow, it is normal to choose Reynolds number with hydraulic diameter as the length scale. By assuming this Reynolds numbers for inlet flow the regimes become turbulent in the flow field for all of the calculations. Therefore, the modified k-ε turbulent transport model was introduced in the main equations.

The numerical simulation is based on suitable convergence for marching procedure. By selecting suitable initial values for this problem, the solution can be led to more accuracy and convergence. With suitable iteration numbers, the accuracy was provided, so the independence of solution for cell numbers has been achieved adequately. In Fig. 4 the non-dimensional temperature distribution, θ, over surface with ribs was shown. In this Fig. 4 the plain case of study was taken for demonstration of independence of solution in cell number. This independence has been achieved up to 65450.

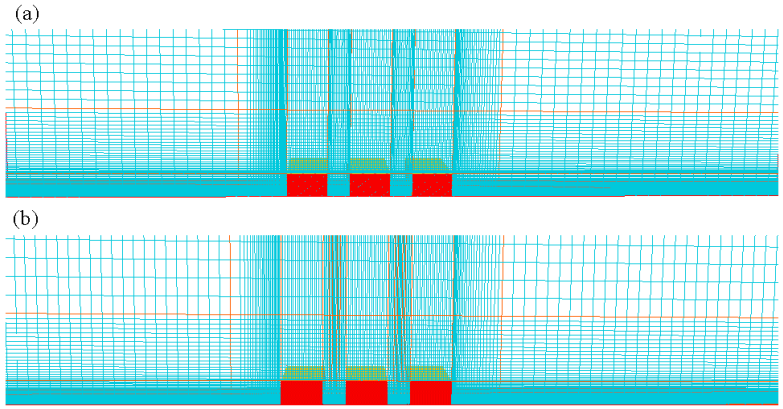

Fig. 3: Grid generation for main problem (a) Plain case and (b) Passive case

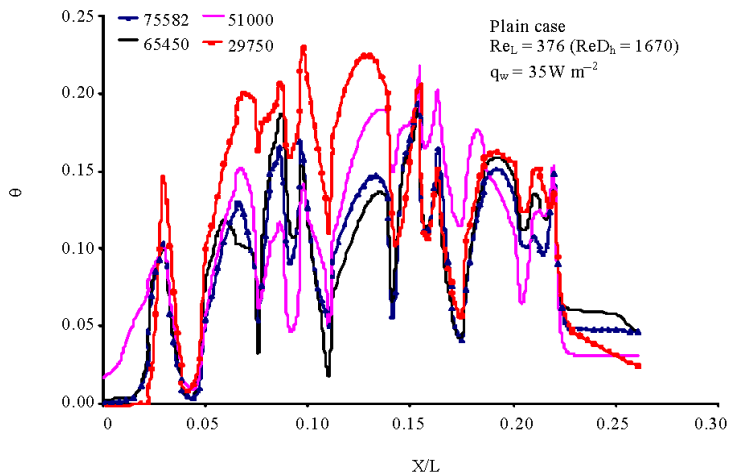

Fig. 4: Independence of solution to the grid size

RESULTS AND DISCUSSION

At first, based on validation of our numerical solution, the geometry of experimental research of Sultan (2000) has been adopted for this study. The plain and passive cases are compared for both works. The Reynolds numbers based on rib width is calculated by means of inlet velocity and physical properties data of air with $T_0 = 25°C$ and $P_0 = 100$ KPa. For internal duct flow in inlet section, the values of k_0 and ε_0 were taken as $k_0 = 1.4\times10^{-5} J\ kg^{-1}$, $\varepsilon_0 = 4\times10^{-7}\ J\ kg^{-1}$ sec. In all cases, the upper, lower and side surfaces of channel were chosen adiabatic. Only over surface of ribs, a constant heat flux about 35 w m^{-2} was exerted.

Figure 5 shows the pressure contours for plain case around the ribs with $Re_L = 376$ ($Re_{Dh} = 1670$). The geometry parameter of S/b (pith to height rib ratio) for pattern of the flow between ribs is most important. As mentioned by Webb *et al.* (2005) if the geometry parameter of S/b become less than 8 in the space between two successive ribs, the near wall flow field consists only one stable recirculation zone. In this study the value of S/b is 0.94. From this Fig. 5 it is shown that between the ribs the constant pressure could demonstrate the existence of a stable recirculation zone.

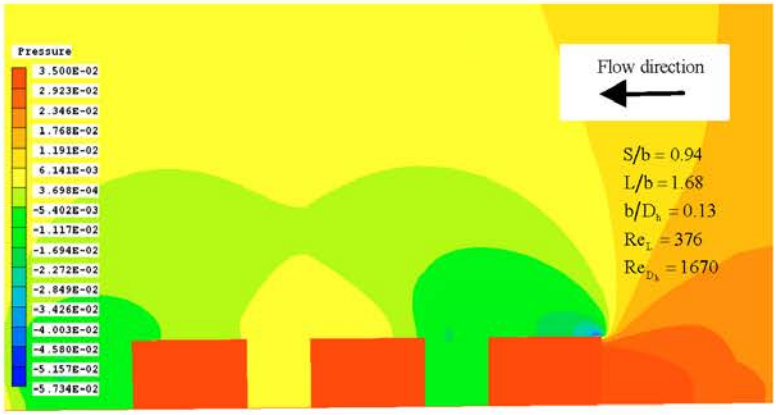

Fig. 5: Pressure contours for plain case

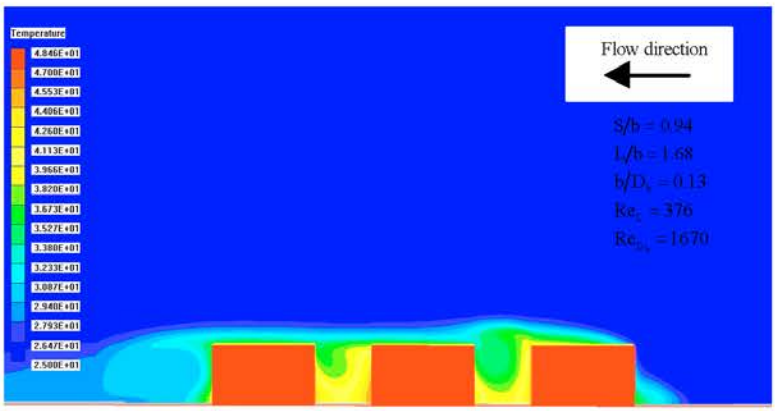

Fig. 6: Temperature contours for plain case

In the streamwise direction of main flow the values of constant pressure between ribs in the reason of pressure drop decrease so the stable recirculation zone also diminish. As explained earlier, the recirculation zone between two successive ribs behaves as resistance to the heat transfer from ribs. Between ribs the strong temperature gradients towards the wall causes the resistance to the heat transfer from the ribs to the main flow. This physical phenomenon becomes weaker in streamwise direction (Fig. 6). For clarification our physical. The flow before arriving to the first rib a small recirculation zone is formed and after separation of flow from top edge of the first rib a separated bubble is generated (Fig. 7). Between first and second ribs, the single recirculation zone is affected by this separated bubble. But for the second circulation zone, this phenomenon is not significant and it becomes stable and smaller than the first one. After the last rib, the recirculation zone becomes large and its S/b.

Tends to infinity, therefore the main flow enforces this zone to reattach to the wall. The reattachment point causes the local sharp variation of shear stress and heat transfer.

In passive method, we used 9 geometry arrangements of holes for our calculations. But for comparison of our results with experimental data of Sultan (2000) only the case of a (staggered array in two rows with holes established symmetrical from the center traverse line) has been considered.

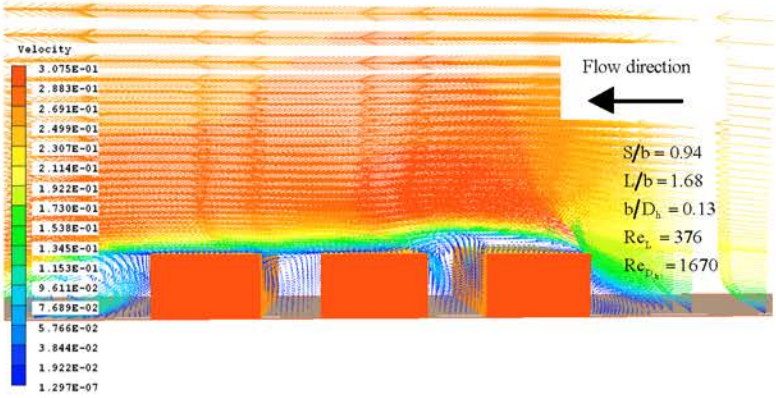

Fig. 7: Velocity vectors for plain case

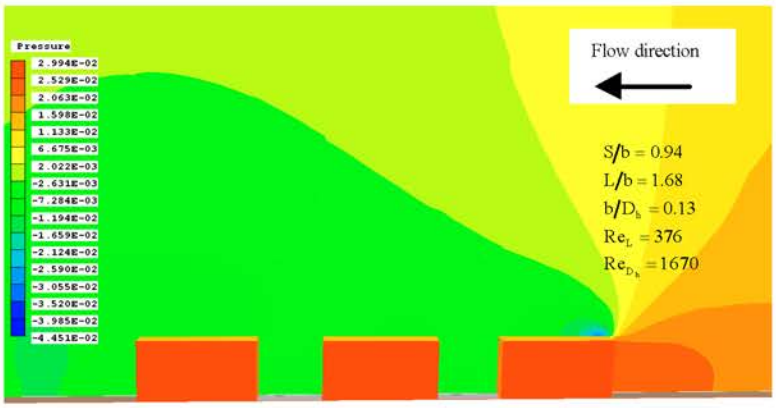

Fig. 8: Pressure contours for passive case

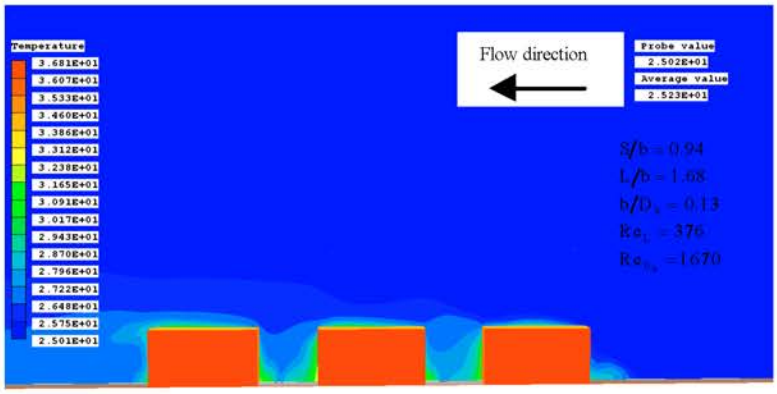

Fig. 9: Temperature contours for passive case

In Fig. 8-10 the pressure contours, the temperature contours and velocity vectors are presented respectively. In comparison of Fig. 5 and 8 it is proven that in passive method the pressure gradients between ribs was established to the direction of main flow. The pressure of ambient is a little

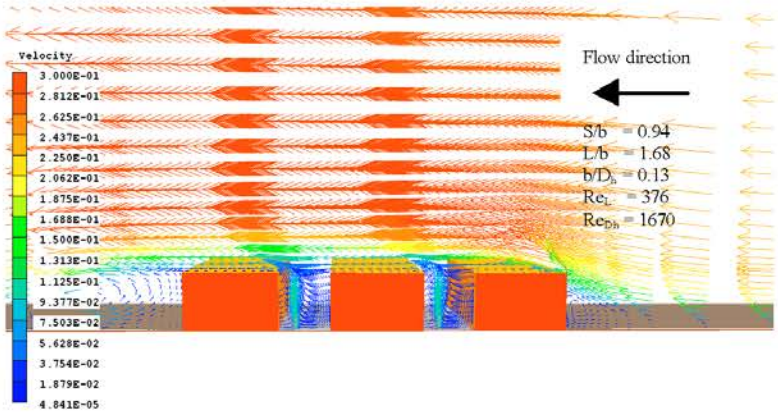

Fig. 10: Velocity vectors for passive case in staggered arrangement (a)

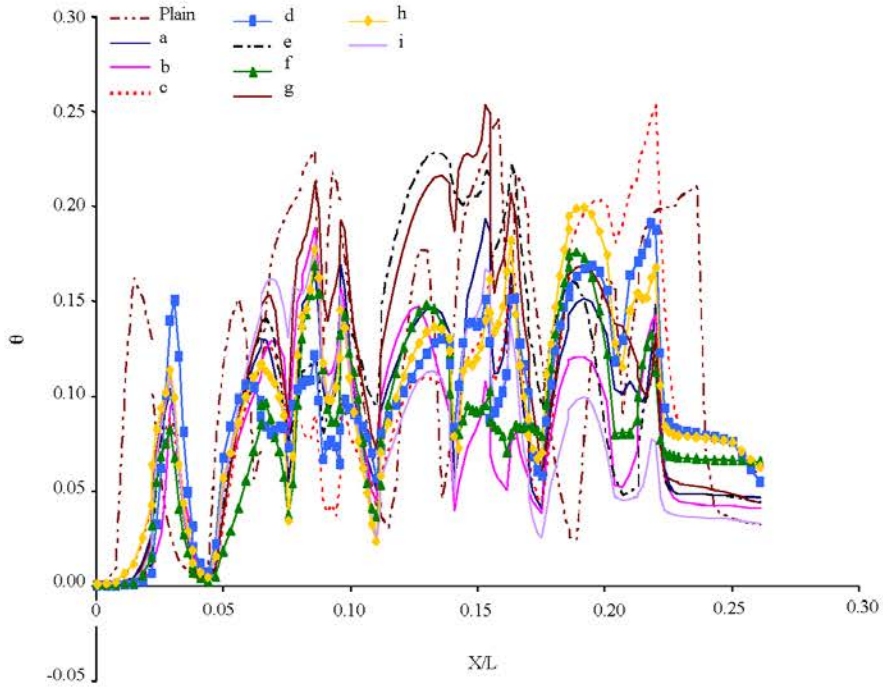

Fig. 11: Dimensionless temperature distribution along the ribs surface for different arrangement ($Re_L = 376$)

larger than that of duct flow. So, the air is sucked from the holes towards the duct transversely. This behavior causes a secondary flow from the holes to the main flow and disturbs the stability of recirculation zones and finally enhances the local heat transfer rate in comparison with plain case. The same interpretation of results is obtained when we compare the Fig. 7 and 10. In Fig. 10 the direction of secondary flow and the distortion of recirculation zones of Fig. 7 were clearly pointed out from the velocity vectors.

Comparison of the temperature contours of plain and passive cases (a geometry arrangement of holes) from the Fig. 6 and 9 demonstrate the significant affects of the passive method use in present study. As it can be shown, in plain case the temperature gradient is weak towards the main flow but in passive case this phenomenon becomes very important which can transport more heat from the ribs to the flow. This is the aim of this study for enhancement of cooling heat transfer from the ribs to the main flow.

As mentioned before, the arrangements of holes have significant effects on heat transfer enhancement. For the nine cases, the air flow Re_L 376 and Re_L 6170 were compared with plain case. In Fig. 11, non-dimensional temperature, θ, versus x/L presented for all cases. The Reynolds number in this case is Re_L 376 (Re_{Dh} = 6170) and the ribs are heated by constant flux q_w 35 w m^{-2}. From these results, the best geometry arrangement of holes between ribs related to the b case which are two rows with staggered near wall holes.

In Table 1 the numerical results for all arrangements and their percentage enhancement of heat transfer for Re_L 376 and Re_L 6170 are shown. From this table it is illustrated that high Reynolds numbers causes more enhancement of heat transfer. The reason of this fact is dependent to the decrease of pressure in duct for high blowing rate of fan. This diminution of pressure in duct causes high gradient pressure from holes inward main flow and provides strong secondary flow.

The enhancement techniques in most of the methods causes dissimilarity between momentum and heat transfer. This dissimilarity must be moderate for any type of enhancement method. In present study the passive method used could be able to moderate enough this phenomenon. The dissimilarity of this method based on secondary flow in comparison with other passive methods is convenient enough. In Table 2 for all of arrangements, momentum transfer and heat transfer performance evaluation criteria is shown by calculation of effectiveness cooling factor η_e for Re_L 376 and Re_L 6170. From the data of this table, it is proven that for high Re_L, Nu/Nu_s becomes greater than unit but $\Delta P_s/\Delta P$ remains close to one.

Table 1: The percent of heat transfer enhancement for different hole arrangements

	Heat transfer rate (%)	
Type of arrangement	Re = 376	Re = 6170
a	110.95	117.845
b	119.237	126.646
c	108.76	115.437
d	113.04	120.650
e	102.73	108.814
f	113.60	120.658
g	102.18	108.212
h	105.47	111.8245
i	118.90	126.300

Table 2: The Efficiency index (η_e) for different hole arrangements

	Re_L = 376			Re_L = 6170		
Type of arrangement	$\Delta P_s/\Delta P$	Nu/Nu_s	η_e	$\Delta P_s/\Delta P$	Nu/Nu_s	η_e
a	0.7398	1.1095	0.8208	0.89700	1.17845	1.05707
b	0.7395	1.19237	0.88175	0.89558	1.26646	1.13420
c	0.7442	1.0876	0.8100	0.88090	1.15437	1.01688
d	0.7456	1.1304	0.8428	0.88242	1.20650	1.06460
e	0.7618	1.0273	0.7826	0.89800	1.08814	0.97715
f	0.6928	1.1360	0.7870	0.92730	1.20658	1.11886
g	0.76125	1.0218	0.7778	0.89323	1.08212	0.96658
h	0.7687	1.0547	0.8107	0.89716	1.118245	1.00324
i	0.7601	1.1890	0.9037	0.89245	1.26300	1.12700

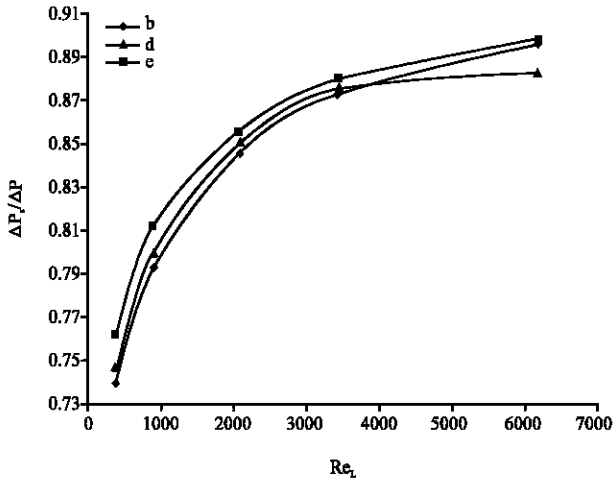

Fig. 12: The ratio of pressure loss for different Reynolds numbers and arrangements

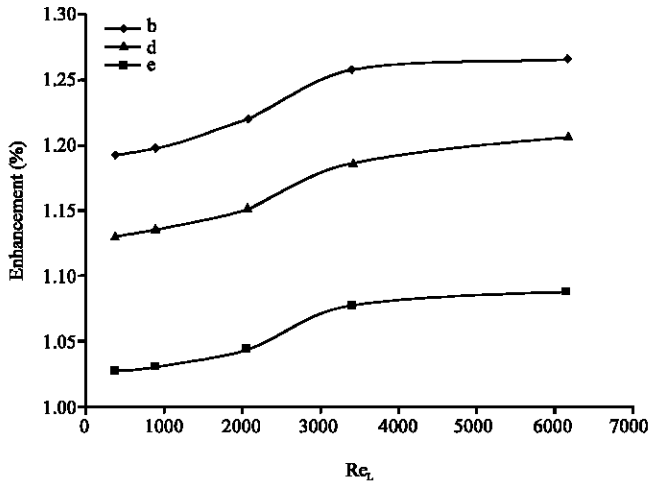

Fig. 13: The heat transfer enhancement in different Reynolds numbers and arrangements

In Fig. 12-15 for geometry arrangements of b, d and e; the momentum transfer $\Delta P_s/\Delta P$, the heat transfer Nu/Nu_s and the effectiveness cooling factor η_e were plotted for different Reynolds numbers. As all of the Fig. 12-15 show, increasing Reynolds numbers causes more effectiveness in the passive technique and b arrangement of holes become more effective than others.

Numerical correlations are used for f_s/f, Nu/Nu_s and η_e for the ranges of $376 \leq Re_L \leq 6170$ and $S/b \leq 8$ for Pr = 0.71 From the results the best fit was correlated to the form of $Y = AX^B$. In the power law relation Y denotes parameters of transport phenomena (f_s/f and Nu/Nu_s) and effectiveness (η_e), X depicts Reynolds number, A and B depend on hole geometry parameters: diameter, number, one row, 2 row and inline or staggered arrangements. For the arrangements of b, d and e we present:

- Two rows, staggered near wall (b case)

$$\frac{f_S}{f} = 0.491Re^{0.07}, \frac{Nu}{Nu_S} = 1.027Re^{0.024}, \eta_e = 0.505Re^{0.094} \tag{18}$$

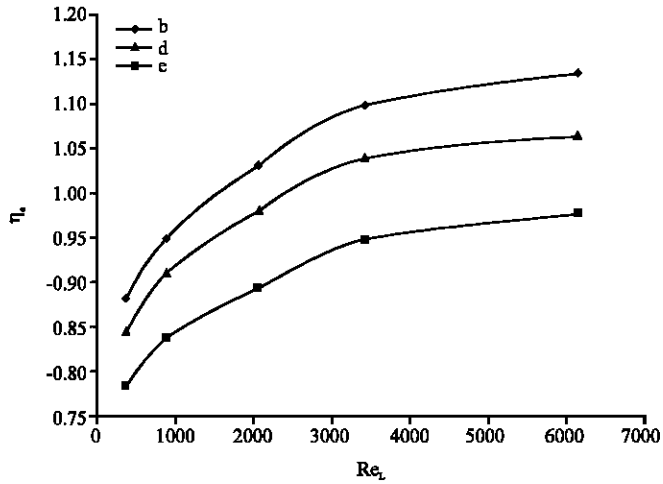

Fig. 14: The efficiency index for different Reynolds numbers and arrangements

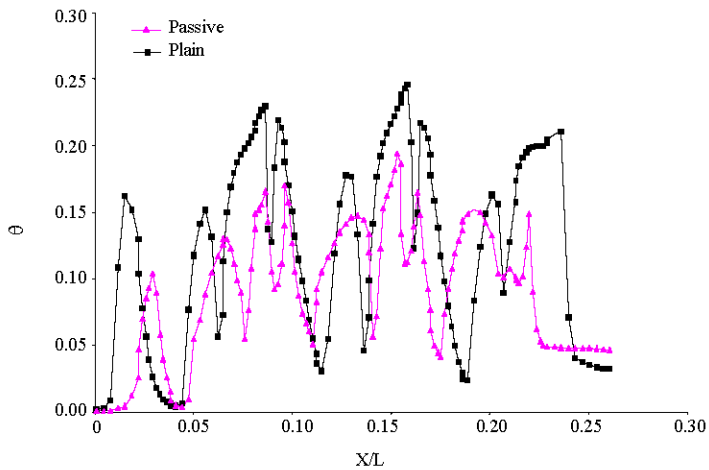

Fig. 15: Dimensionless Comparison of θ distribution for both cases with (Re_L = 376) for a arrangement

- Two rows, inline near wall (d case)

$$\frac{f_S}{f} = 0.519Re^{0.063}, \frac{Nu}{Nu_S} = 0.971Re^{0.024}, \eta_e = 0.504Re^{0.087} \tag{19}$$

- One row, inline near left wall (e case)

$$\frac{f_S}{f} = 0.539Re^{0.059}, \frac{Nu}{Nu_S} = 0.893Re^{0.022}, \eta_e = 0.481Re^{0.082} \tag{20}$$

In Fig. 15 and 16 for arrangement of a case, the comparison of results for θ and Nu variations along channel in plain and passive cases are presented for Re_L = 376.

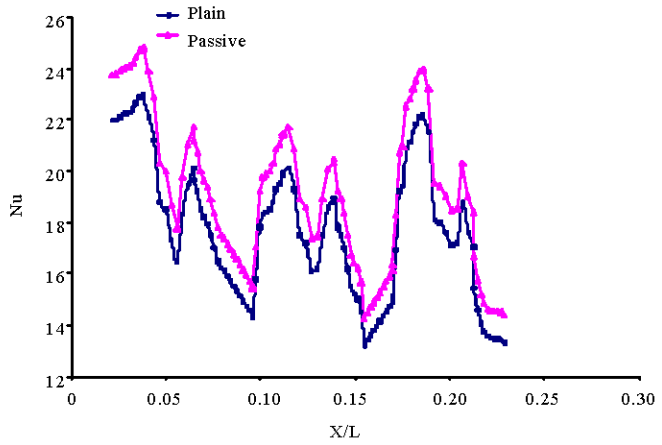

Fig. 16: Comparison of Nusselt number distribution along the ribs for plain and passive cases ($Re_L = 376$)

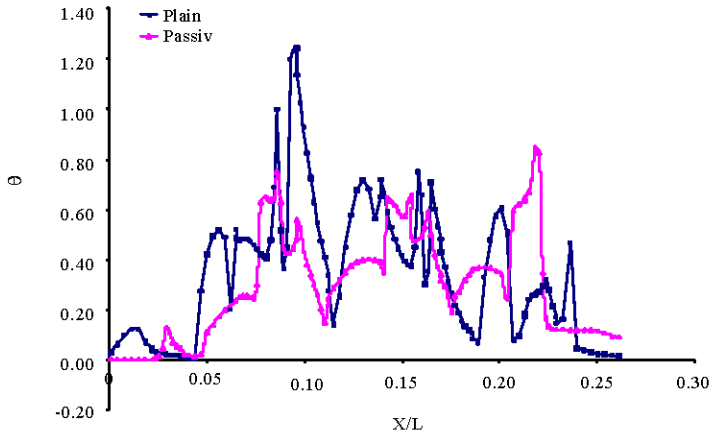

Fig. 17: Dimensionless temperature distribution along the ribs surface ($Re_L = 6170$)

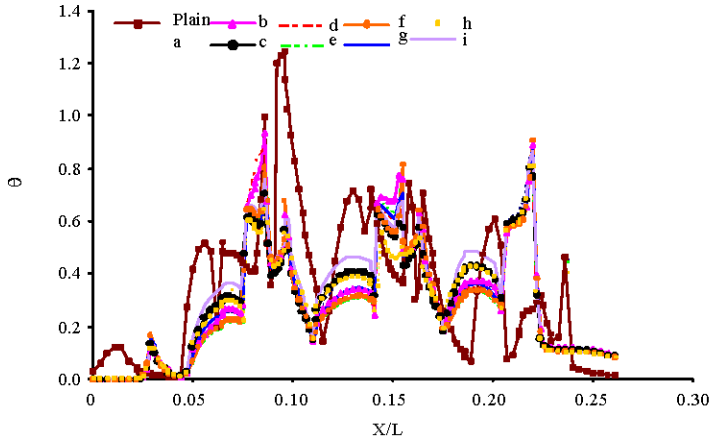

Fig. 18: Dimensionless temperature distribution along the ribs surface for different arrangement ($Re_L = 6170$)

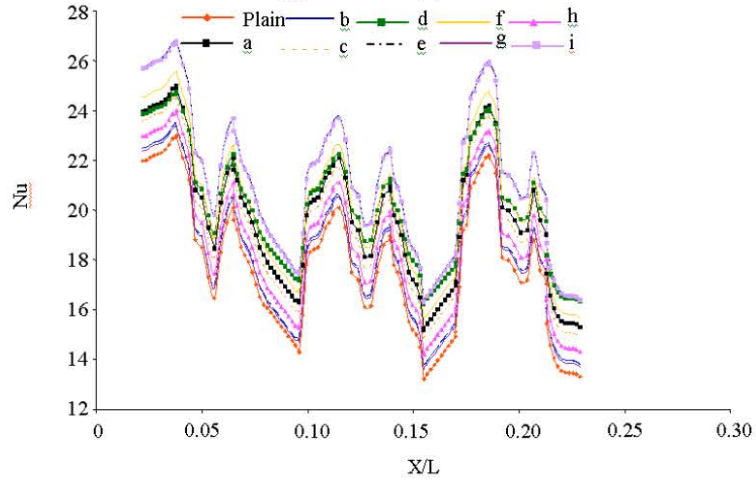

Fig. 19: The Nusselt number distribution along the ribs surface for different arrangements ($Re_L = 376$)

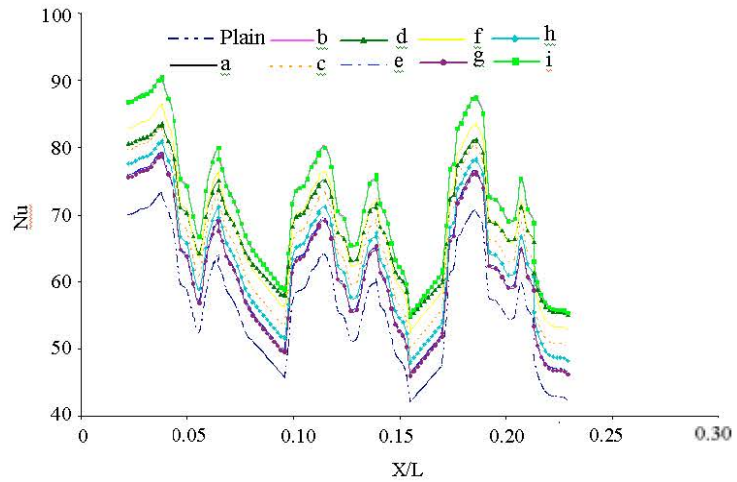

Fig. 20: Nusselt number distribution along the ribs surface for different arrangements ($Re_L = 6170$)

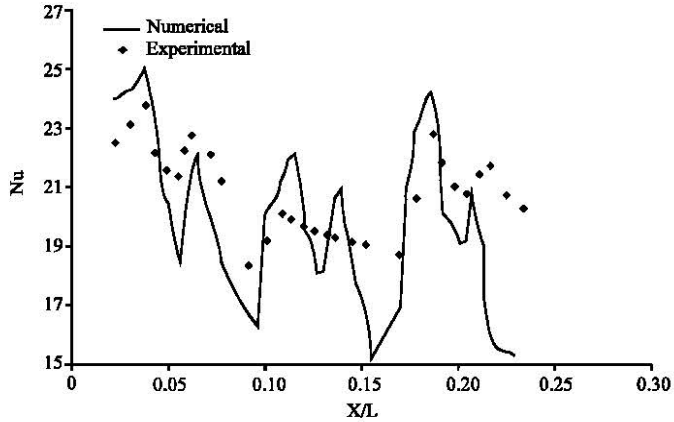

Fig. 21: Comparison of Nusselt number distribution between numerical and experimental results for passive case ($Re_L = 376$)

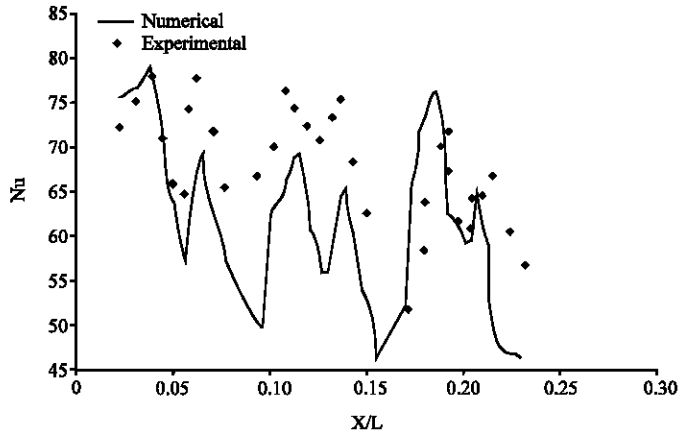

Fig. 22: Comparison of Nusselt number distribution between numerical and experimental results for passive case ($Re_L = 6170$)

The same behavior for $Re_L = 6170$ also is shown in Fig. 17 and 18. It can be seen that for θ in all of cases significant diminution from plain case achieved. Therefore it caused the local increasing of Nusselt Number.

In Fig. 19 and 20 distributions of Nu along channel at $Re_L = 376$ and $Re_L = 6170$ are shown for all of hole geometry arrangements. From these results the importance of hole arrangements are clearly illustrated to the heat transfer enhancement.

In Fig. 21 and 22 comparison of Nusselt numbers between our calculations and experimental data of Sultan (2000) in the arrangement of a case are shown for $Re_L = 376$ and $Re_L = 6170$. As it cab be proven the computational results nearly have good agreement with experimental data and average rms errors of results on Nusselt numbers become around 10%.

CONCLUSION

The numerical simulation used in this study has ability to predict the efficiency enhancement of transport phenomena even in complex geometry. In present geometry in cavities between transverse ribs using holes as a passive technique can distort stable recirculation zone and enhance the heat transfer rate. The geometry and arrangement of holes has significant effects on enhancement. For 9 arrangements used in this work, the case of two rows in staggered configuration near the walls is more effective than others. Anyhow increasing Reynolds number causes more enhancements. From our numerical simulation 19 and 27% enhancement of heat transfer were gained for $Re_L = 376$ and $Re_L = 6170$, respectively in comparison with plain case for b geometry arrangement. For the geometry arrangement of a these values become 11 and 18%.

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