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Numerical Investigation of Heat Transfer for Laminar and Turbulent Flow over Ribbed Walls

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Abstract*— A numerical investigation is conducted flows over heated ribbed walls under the effect of the Reynolds number and rib height. Laminar and turbulent flow with constant thermophysical properties is assumed for air at two values of the initial streamwise Reynolds number of $2.7x10^5$ and $3.4x10^6$. The finite- volume-method is employed to solve the governing equations, coupled with the k─ε turbulence model with near-wall treatment. The results indicate that the presence of the ribs can effectively enhance the heat transfer. The heat transfer enhancement increased rib height and become more pronounced in laminar than that of turbulent flows.

Index Terms*— Heat transfer enhancement, flow separation, ribbed wall, laminar flow, turbulent flow

I. INTRODUCTION

Heat transfer enhancement over ribbed walls at low and high Reynolds number has been a major subject of intensive research over the years. It has numerous applications such as cooling of electronic systems, internal cooling inside turbine blades, compact heat exchangers, biomedical devices, etc. Therefore, several investigators have studied heat transfer enhancement from heated ribs.

Tsay and Cheng [1] studied on the two-dimensional forced convection in a channel with heat generating blocks, and indicated that heat transfer increased with increasing block height. An experimental and numerical investigation of turbulent heat transfer in a channel with periodically arranged rib roughness elements was carried out by Braun et al. [2]. Ryu et al. [3] worked out the heat transfer characteristics of a turbulent flow in channels with two-dimensional ribs and three dimensional blocks in the context of surface roughness effects. Perng and Wu [4] investigated the turbulent flow field and heat transfer enhancement of mixed convection in a horizontal block-heated channel. A simulation for a backward-facing step flow and heat transfer inside a channel with ribs turbulators was studied by Mushatet [5] who reported that the Reynolds number and contraction ratio have a significant effect on the heat transfer. Iaccarino et al. [6] investigated the effect of thermal boundary conditions on numerical heat transfer predictions in rib-roughened passages. Luo et al. [7] carried out a numerical study on the flow and forced-convection characteristics of turbulent flow through parallel plates with periodic transverse ribs, and found that the standard k-ε model had superiority over the Reynolds stress mdel. Korichi and Oufer [8] conducted a numerical investigation of convective heat transfer between a fluid and three physical obstacles mounted on the lower wall and on the upper wall of a rectangular channel.

Computation of flow and heat transfer in two dimensional rib-roughened passages, using low Reynolds number turbulence models was investigated by Iacovides and Raisee [9]. Yuan [10] conducted a numerical study for the characteristics of the periodically fully developed turbulent flow and heat transfer in a channel with transverse opposite-positioned fins. A numerical investigation of forced convective cooling of an array of obstacles was performed to synthesize the effects of various pertinent parameters on the cooling performance by Zeng and Vafai [11]. Nakajima et al. [12] presented numerical results simulating a three-dimensional laminar separated flow and heat transfer around staggered surface-mounted rectangular blocks in a plane channel.

To better understand the thermal distribution over ribbed walls, numerical analyses of the forced-convection characteristics of the walls should be performed. The present study was conducted to fulfill this purpose and to investigate the effects of the Reynolds number and rib height on the heat transfer behavior.

II. METHOD AND APPROACH

A steady, two dimensional and incompressible flows was chosen to represent flow over the heated walls with sequence ribs and modeled with the FLUENT code based on the finite-volume method. The geometry of the walls and computational domain is depicted in Fig. 1. The inlet lengths of L, outlet lengths of 5L and domain height of 5L were used. The bottom plate was uniformly heated. The unheated plate length was fixed at 20L in laminar and 40L in turbulent flow. A flat with a 2.4L width and 7.5L length and three different ribbed walls were used in the investigations. The first ribbed wall has dimensionless rib height (h/H) of 0.03 and the streamwise wall length of 6L with an array of 9 rib, the second one with h/H=0.04, 6L length and 8 rib, and the last wall with h/H=0.05, 4.8L length and 7 rib. The ratio of the rib width to cavity width was fixed at w/s=1 and the walls were uniformly heated.

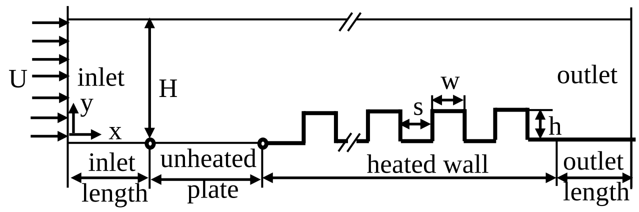

Fig. 1. Wall geometry and computational domain.

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The governing equations were the steady-state continuity, momentum and the energy equations for turbulent flow, can be written as follows:

$$\frac{\partial u_i}{\partial x_i} = 0 \tag{1}$$

$$\rho\frac{\partial u_i u_j}{x_j} = -\frac{\partial P}{\partial x_i} + \mu\frac{\partial}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)+\rho\frac{\partial}{\partial x_j}\left(-\overline{u_i'u'_j}\right) \tag{2}$$

$$\rho C_p\frac{\partial u_i T}{x_i} = \frac{\partial}{\partial x_i}\left(\lambda\frac{\partial T}{\partial x_i}-\rho\overline{u_i'T'}\right) \tag{3}$$

In the Eq. (2), the Reynolds stress term $\left(-\overline{u_i'u'_j}\right)$ is defined as,

$$-\overline{u_i'u'_j} = \mu_t\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right) \tag{4}$$

where μt is turbulent viscosity.

The standard k─ε model (Launder and Spalding, [13]) is a semiempirical linear eddy viscosity model based on the model transport equations for the turbulence kinetic energy (k) and its dissipation rate (ε). The model transport equation for k is derived from the exact equation, whereas that for ε is obtained from physical reasoning and has little resemblance to its mathematically exact counterpart. The Reynolds stress term is calculated according to the Boussinesq hypothesis, and the dynamic viscosity is replaced by an effective viscosity ($\mu_{eff} = \mu+\mu_t$ with, in which C_μ is a viscosity constant). Because of its simplicity and overall good performance properties, it had been chosen in the present study. The transport equations are defined as,

$$\rho\frac{\partial}{\partial x_i}\left(ku_i\right) = \frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+G_k-\rho\varepsilon \tag{5}$$

$$\rho\frac{\partial}{\partial x_i}\left(\varepsilon u_i\right) = \frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]+C_{1\varepsilon}\frac{\varepsilon}{k}G_k-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}+S_\varepsilon \tag{6}$$

to obtain these variables, where, $G_k = \mu_t S^2$, $S = \sqrt{2S_{ij}S_{ij}}$ and $S_{ij} = \frac{1}{2}\left(\frac{\partial u_j}{\partial x_i}+\frac{\partial u_i}{\partial x_j}\right)$. The model constants are $C_{1\varepsilon}=1.42$, $C_{2\varepsilon}=1.92$, $C_\mu=0.09$, $\sigma_k=1.0$ and $\sigma_\varepsilon=1.3$, and standard wall function is used for near wall treatment.

The SIMPLE-C algorithm and pressure staggering option was used for the solutions. The momentum equations were discretized by a second order interpolation scheme and a 2-D quadrilateral structured mesh was used, as shown in Fig. 2. A fine mesh with the minimum cell number of 38000 and the maximum cell number of 48000 was provided through the walls.

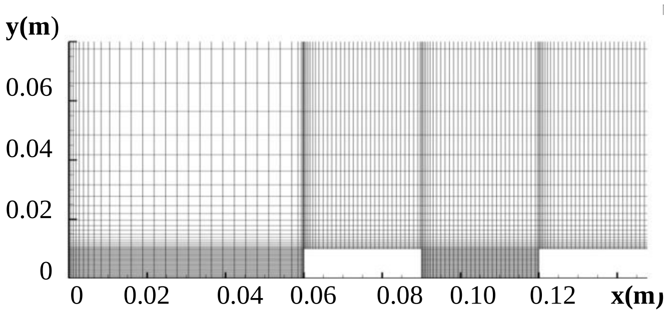


Fig. 2. Mesh structure.

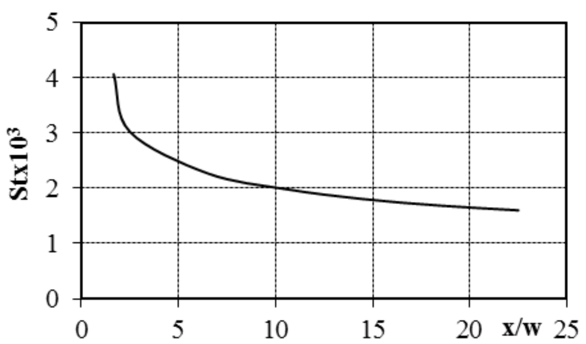
The boundary conditions were given as,

1. The air entered the channel at ambient temperature with a uniform velocity *($u=U$, $v=0$, $T=T_0$).*
2. The upper and lower plates were assumed to be insulated thermally ($\partial T/\partial y = 0$), except the heated section, which was provided with a uniform heat flux of 200-250 W/m^2 in laminar and 650-800 W/m^2 in turbulent flow.
3. No-slip boundary conditions were enforced at all walls and rib sides conditions *($u=0$ and $v=0$).*
4. Zero streamwise gradients of velocity components and temperature in the axial direction were applied at the exit plane of the channel ($\partial u/\partial x = 0$, $\partial v/\partial x = 0$, $\partial T/\partial x = 0$).

III. RESULTS AND DISCUSSION

The heat transfer characteristics of the ribbed walls were investigated numerically with the effects of Reynolds number and rib height, and all results were compared to those of the flat plate values.

The streamwise variations of Stanton numbers $\left(St = h/\rho UC_p\right)$ over flat plate at Rex=2.7x10^5 and 3.4x10^6 are given in Fig. 3 (a) and (b), respectively. The local Stanton numbers changed from 4.1x10 ^{3} to 1.6 x10^{-3} in laminar flow and 2.8x10^{-3} to 2.1x10^{-3} in turbulent flow, and decreased in the streamwise direction as expected.



(a) Re_x=2.7x10^5

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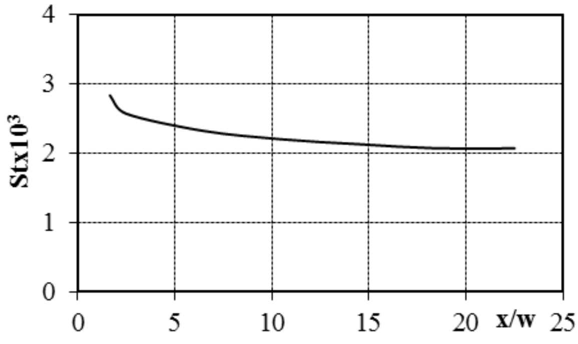

(b) $Re_x=3.4\text{x}10^6$

Fig. 3. Streamwise variation of Stanton numbers over flat plate.

The streamwise variations of Stanton numbers at $Re_x=2.7\text{x}10^5$ for h/H=0.03, 0.04 and 0.05 are presented in Fig. 4. The curves showed similar trends and the streamwise variation of Stanton numbers increased with dimensionless rib height. The Stanton numbers rose by 250, 240 and 230% on the first ribs and by 175, 200 and 215% on the last ribs compared to the flat plate values for h/H=0.03, 0.04 and 0.05, respectively. The maximum heat transfer were obtained at the inlet corners of the first rib and then decreased in the subsequent blocks, similar to Luo et al. [7] findings. The local Stanton numbers were bigger than those of flat surface 80, 50 and 25% on the middle of the first and 35, 20 and 15% on the middle of the last cavities at h/H=0.03, 0.04 and 0.05, respectively. The Stanton numbers on the cavities also increased with rib height and the lower Stanton number cavities between the ribs is a result of recirculation region and adverse pressure gradient.

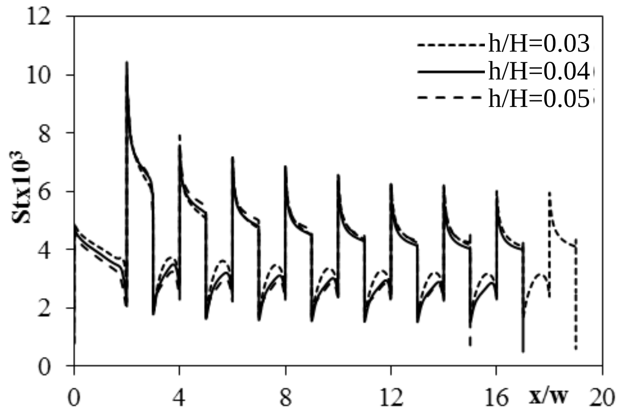

Fig. 4. Streamwise variation of Stanton numbers at $Re_x=2.7\text{x}10^5$.

The turbulent Stanton numbers for h/H of 0.03, 0.04 and 0.05 showed similar distributions to laminar flow, as given in Fig. 5. St values increased on the ribs, while decreased in the cavities due to negative effect of flow separation between the ribs. The enhancement of the heat transfer at $Re_x=3.4\text{x}10^6$ changed from 165, 150 and 140% on the first to 110, 115 and 130% on the last ribs, while the St was higher than that of flat surface by 20, 15, 10% on the first and 7, 5, 3% on the last cavities for h/H=0.03, 0.04 and 0.05 respectively. The heat transfer increased more remarkable in the laminar flow than the turbulent as the block heights increased, like Mohammed [14]. The heat transfer increased with rib height as laminar flow due to the augmentation in the turbulent level in the thermal boundary layer and the vortex with greater energy.

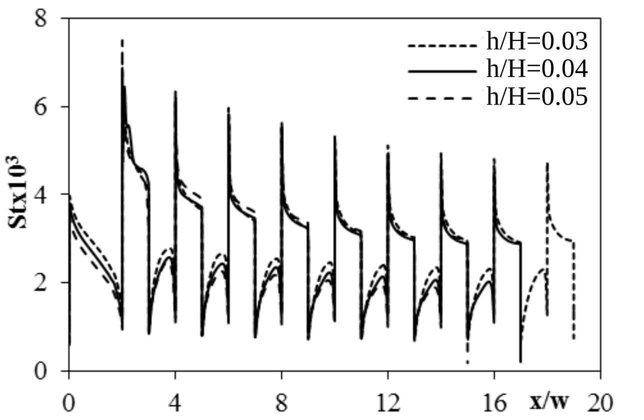

Fig. 5. Streamwise variation of Stanton numbers at $Re_x=3.4\text{x}10^6$.

A comparison of averaged Stanton numbers of ribbed walls to those of flat plate was presented in Fig. 6. The ribbed walls increased the average laminar Stanton numbers up to 2.05, 2.10 and 2.12 fold of flat plate value and up to 1.30, 1.31 and 1.34 in turbulent flow for h/H of 0.03, 0.04 and 0.05, respectively. These results showed that heat transfer increased with dimensionless rib height and the heat transfer enhancement become more pronounced in laminar than the turbulent flows, similar to Ryu et al. [9].

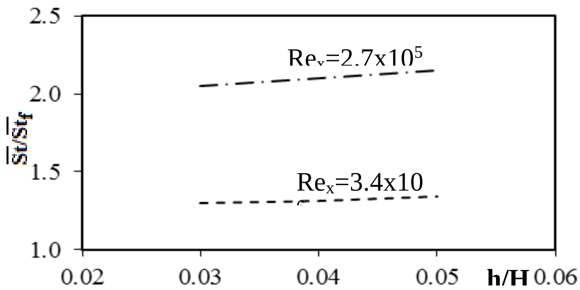

Fig. 6. Mean Stanton numbers variations with h/H and Re_x.

IV. CONCLUSION

Numerical simulations of fluid flow and heat transfer over uniformly spaced rectangular ribs were performed for $2.7\text{x}10^5<Re_x<3.4\text{x}10^6$ and 0.03<h/H<0.05. The presence of the ribs destabilized the flow and enhanced heat transfer. The average heat transfer enhancement between the fluid and the heated ribs of up to 103, 105, 112% in laminar flow and 30, 32, 35% in turbulent flow in the overall Stanton number was obtained for h/H of 0.03, 0.04 and 0.05 respectively with respect to the flat plate values. The maximum heat transfer were obtained at the beginning corner of the first rib and then decreased in the subsequent blocks and the minimum values were found between the last two ribs. The heat transfer increased with dimensionless rib height and became more remarkable in the laminar flow than the turbulent flows.

Nomenclature

C_p constant pressure specific heat

h rib height

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h convective heat transfer coefficient
H domain height
k turbulence kinetic energy
L inlet length
P pressure
Re_x streamwise distance Reynolds number
s rib spacing
St Stanton number
T temperature
u streamwise velocity
U mean free stream velocity
v pitchwise velocity
w rib width
x streamwise directions
y pitchwise directions
Greek symbols
ε dissipation rate of turbulence kinetic energy
λ thermal conductivity
μ dynamic viscosity
μ_t turbulent viscosity
ρ density
Superscript
— average

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