NUMERICAL INVESTIGATION OF TURBULENT FORCED CONVECTION IN A RECTANGULAR CHANNEL WITH TRANSVERSE RIBS

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ABSTRACT

A numerical investigation of turbulent forced convection and heat transfer characteristics in a two-dimensional rectangular channel with transverse ribs is carried out in the present work. The bottom wall formed by heated ribs of rectangular section is subject to a uniform heat flux condition, whereas the upper wall is thermally insulated. The effect of turbulence models is studied by utilizing three models: the standard k-ε, the Renormalized Group (RNG) k– ε , and shear stress transport (SST) k-ω. The numerical approach is based on the finite volume method and the SIMPLE algorithm has been adopted for the discretization of the pressure–velocity terms. The Fluent CFD code is used in this work. The fluid flow and heat transfer characteristics are presented for Reynolds number fixed at 9.4×10^4. The effects of geometry parameters, as well as characteristics of the turbulent forced convection flow through the ribs and in to the rectangular channel, are analyzed in detail. The predicted results from using three turbulence models reveal that the standard k−ε turbulence model generally provides better agreement as compared to the others.

Keywords: *forced convection, turbulent flow, ribs, fluent.*

NOMENCLATURE

Symbols:

D_h	hydraulicdiameter , m
L_t	total length of the channel, m
L_1	entrace length of the channel, m
H	height of the channel, m
w	rib length, m
e	rib height, m
T	fluid temperature, K
T_0	ambient temperature, K
q	heat flux, W/m^2
U	mean air velocity in the channel, m/s
U_i	velocity component in the i direction, m/s
U_j	velocity component in the j direction, m/s
k_s	thermal conductivity of the solid wall, W/mK
k_f	thermal conductivity of the air flow, W/mK
k	turbulent kinetic energy, m^2/s^2

Greek Letters:

μ	Dynamic viscosity , [Pa.s]
μ_t	turbulent viscosity , [kg/m s]
ρ	density of the air , [kg/m^3]
ν	kinematic viscosity , [m^2/s]
ε	Turbulent dissipation rate, [m^2/s^3]

Dimensionless numbers

Re_D	Reynolds number, [UD/ν]
Pr	Prandtl number, [ν/α]

1. INTRODUCTION

Performance turbulence models in predicting the flow and temperature fields have become increasingly important for several industrial problems. This also applies to a turbulent flow in a conduit contains obstacles on one of its walls, and this frequently occurs in many industrial applications such as; heat exchangers, electronic systems, solar collectors plane and the gas turbine cooling systems. The improvement of heat transfer by convection in these thermal systems is important because it guarantees a good operation and prevents any malfunction which degrades the performance of the latter.

In this perspective, many studies conducted by researchers and scientists, on the numerical or experimental approaches concern turbulent forced convection heat transfer, such as Tong Miin. Liou and al.[1] have performed an analysis on the heat transfer and fluid flow behavior in a rectangular channel flow with stream-wise periodic ribs mounted on of the bottom wall. The Characteristics of heat transfer and turbulent flow over a repeated-rib-geometry, rough-walled surface with square rib on the top and the bottom surface in a rectangular channel were experimentally investigated by Hiroshi Sato and al. [2]. Lorenz and al. [3] have studied the distributions of the heat transfer coefficient and the pressure drop along the wall inside an asymmetrically ribbed channel measured for thermally developing and turbulent flow at $10^4 < Re < 10^5$. Luo and al. [4] studied numerically two turbulence models, the standard (k-ε) model and the Reynolds stess model (RSM) to predict the characteristics of heat transfer by turbulent forced convection flow fully developed between two horizontal parallel plates. Smith Eiamsa and al. [5] have conducted a numerical investigation of turbulent forced convection in a two-dimensional channel with periodic transverse grooves on the lower channel. Yemenici and Sakin [6] conducted a numerical analysis of the characteristics of heat transfer by forced convection for laminar and turbulent flows over heated ribbed walls and studied the effect of the Reynolds number and rib height.

The objective of this work is to investigate numerically the heat transfer by turbulent forced convection in a horizontal channel with a ribs mounted in bottom wall using CFD fluent code for several turbulence models.

2. GEOMETRY AND MATHEMATICAL MODEL

The problem studied is a horizontal channel with eight periodic ribs placed along the bottom wall (Figure 1). The bottom wall is subject to a uniform heat flux condition while the top wall is adiabatic. The channel height is H, the channel length is 75 H, and each rib has a height (e) and length (w). The ratio between the length and height of the rib is set 2. The hydraulic diameter is set H. The distance before the first rib and the channel outlet is L1 = L2 = 10H. The fluid is Newtonian, incompressible and purely turbulent enters the channel with an ambient temperature (T0 = 300 K) and a uniform velocity U0.

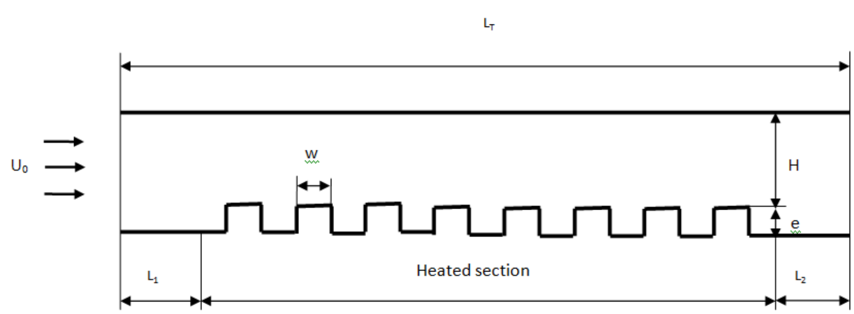


FIGURE 1. Geometry of the present problem

The governing equations are the steady-state continuity, time-averaged momentum, and energy equations for a turbulent flow, ie.,

$$\frac{\partial(\rho U_j)}{\partial x_j}=0 \qquad (1)$$

$$\frac{\partial(\rho U_j U_i)}{\partial x_j}=-\frac{\partial P}{\partial x_i}+\frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial U_i}{\partial x_j}+\frac{\partial U_j}{\partial x_i}\right)\right]+\frac{\partial}{\partial x_j}\left(-\rho\overline{U_i U_j}\right) \qquad (2)$$

$$\frac{\partial}{\partial x_j}(\rho U_j T)=\frac{\partial}{\partial x_j}\left((\Gamma+\Gamma_t)\frac{\partial T}{\partial x_j}\right) \qquad (3)$$

Where Γ and Γt are molecular thermal diffusivity and turbulent thermal diffusivity, respectively and are given by

$\Gamma=\frac{\mu}{\Pr}$ and $\Gamma_t=\frac{\mu_t}{\Pr_t}$

The Reynolds stress term $-\rho\overline{U_i U_j}$ is defined as

$$-\rho\overline{U_i U_j} = \mu_t\left(\frac{\partial U_i}{\partial x_j}+\frac{\partial U_j}{\partial x_i}\right)$$

Where µt is turbulent viscosity

The boundary conditions for the present problem are established as follows,

1- Along the upper surface (0<x<Lt, y=e+H)u=0, v=0 et $\frac{\partial T}{\partial y}=0$

2- Along the bottom surface (y=0),u=0, v=0 , $\frac{\partial T}{\partial y}=0$ et $k_s\frac{\partial T}{\partial n}\Big|_{y=0}=k_f\frac{\partial T}{\partial n}\Big|_{y=0}$

3- At the inlet plane (x=0, 0<y<e+H)

u=U, v=0 et T= T_0

4- At outlet plane (x=Lt, 0<y<e+H)

$\frac{\partial u}{\partial x}=0$, $\frac{\partial v}{\partial x}=0$et $\frac{\partial T}{\partial x}=0$

3. RESULTS AND DISCUSSION

3.1. EFFECT OF LOCAL NUSSELT NUMBER

Figure 2 present the local Nusselt number distribution along the ribs at Re_D = 94000, where the peripheral distance measured starting from the left lower corner of the rib. For both models, a similar trend of Nusselt numbers was observed along the ribs, and the two maximum values are obtained around the rib: one is along the ribs left top corner (point b), and another is immediately in front of the next rib (before point e). We see also a great heat transfer on the ribs left wall (a-b), the transfer begin to decrease on the wall which is parallel to the flow (b-c) and then a low heat transfer is localized to the right of the wall (c-d), this is due to the presence of a vortex between the two ribs.It can be observed that a light enhancement of the heat transfer is obtained by standard k-ε model.

A comparison between the predicted Nusselt number distribution with the experimental data along the rib at Re_D = 94000, as shown in figure 2. Very close predictions of Nusselt number distributions along the rib's left surface (a-b) were obtained with k– ε models. A gap of values of the Nusselt numbers was found in the cross-section area of the rib (b-c). In area (c-d) et (d-e) the results predicted by the k– ε models are close to experimental results, and take almost the same path.

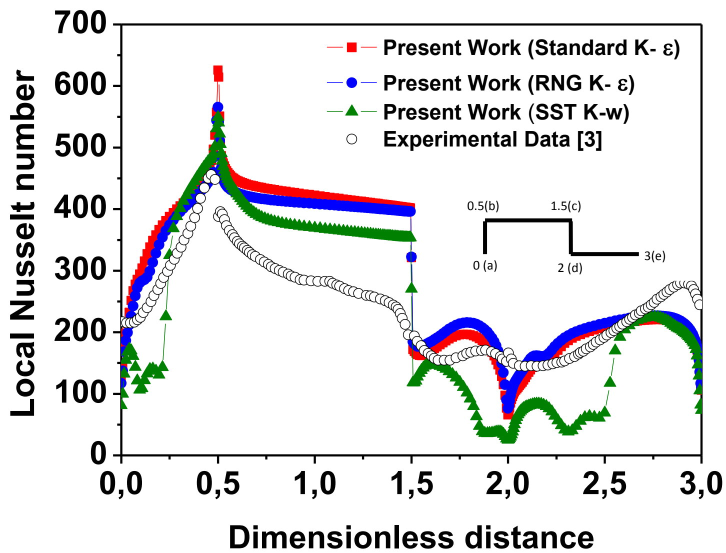
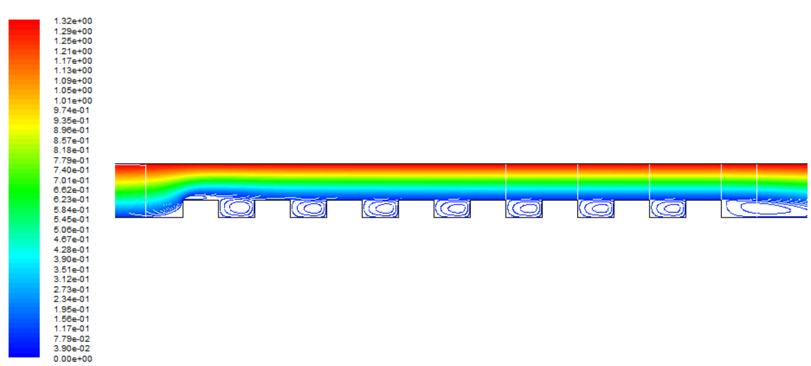


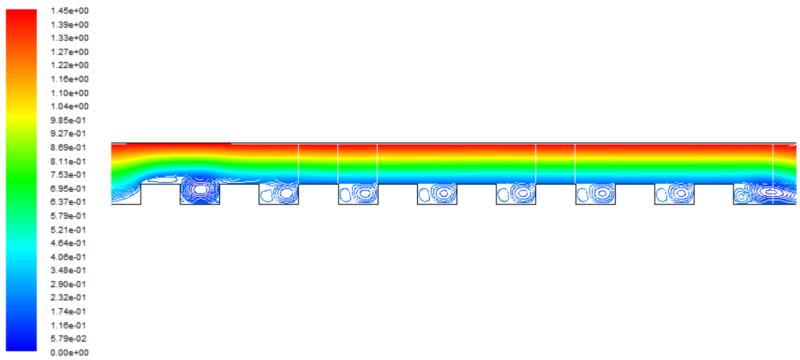
FIGURE 2. Distribution of local Nusselt number for Re_D = 94000 with experimental data.

3.2. Flow Pattern

Figures 3. (a) and (b) showed the streamlines characteristics predicted by the standard k-ε model and the SST k-ω at Reynolds number Re= 9.4×10^4.It can be shown that for both models, an clockwise at the downstream surface of the rib by the two models. However, both models gave almost similar results regarding the flow structure. The SST k-ω model predicts a stronger recirculating zone, comparatively to that predicted by standard k– ε model.



(a)

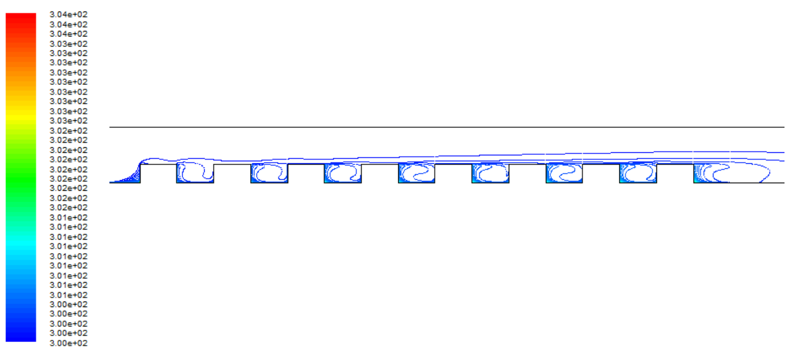


(b)

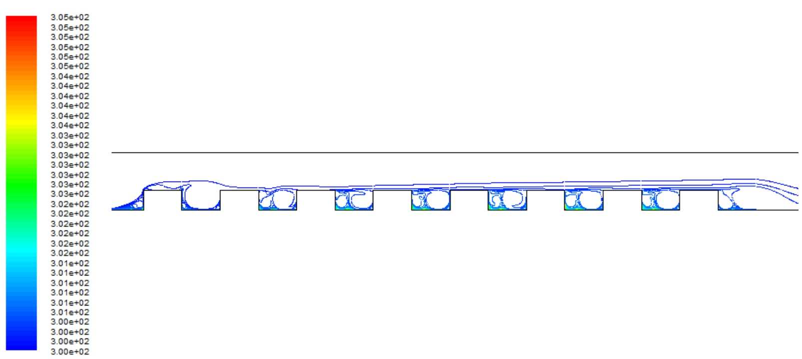
FIGURE 3. Streamlines of the turbulent flow at Re= 9.4×10^4:
(a) Standard k-ε and (b) SST k-ω

3.3. The isotherm lines

The isotherm lines around the ribs at Reynolds number of Re=94000 are presented in Figure 4. It is seen that the secondary flow in the downstream region of the rib has affected the temperature field significantly. It can be concluded that the thermal boundary layer was developed in this section.



(a)



(b)

FIGURE 4. The isotherm lines of the turbulent flow around the ribs at Re= 9.4×10^4:
(a) Standard k-ε and (b) SST k-ω

4. CONCLUSIONS

The forced convection characteristics of the turbulent flow through two-dimensional rectangular channel with a ribbed bottom surface are simulated by using three turbulence models: the standard k-ε, the RNG k-ε and shear stress transport (SST) k-ω. It was concluded that the standard k–ε model gave a better estimation of heat transfer than the others.

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