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Research Article

Numerical Heat Transfer Study of Turbulent Square Duct Flow through W-Type Turbulators

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Abstract

Several cooling technique have been developed to enhance heat transfer in square duct. Different rib arrays inside square channel are widely used to enhance heat transfer rate. The reason that ribs increase the fluid flow turbulence near the wall, disrupt the boundary layer and also increase the heat transfer area. In this paper, numerical analysis is carried out three different angles of turbulators were placed in square duct. All turbulators located on bottom side wall of duct. The numerical simulation are carried on square duct having hydraulic diameter (Dh) of 0.05m. Air is working fluid with the flow rate in terms of Reynolds number ranging from 15,000 to 20,000. Details for rib height (e), pitch distance between turbulators (P) and turbulators angle are similar to experimental reference. The model is creating using Ansys ICEM software. Numerical simulations were performed using the CFD software package ANSYS 14.5 FLUENT. Turbulence closure was achieved using k-ε turbulence model, with enhance wall treatment for the simulation were used. In this, the heat transfer characteristics of square duct with internal w-shaped ribs with different angles and pitch ratio 0.3 were plotted.

Keywords: *Turbulators, FLUENT, Rib, Heat transfer, square duct*

1. Introduction

The need of high performance thermal system in many engineering application. Need to find various techniques to improve heat transfer in system. In convention area heat transfer improved by means of various augmentation techniques. Means increase the heat transfer area by ribs, protrusion and roughness. However, the thermal/hydraulic performance of the ribs is affected by many factors including the holes in a rib, size and spacing. Since 1980's many experimental and numerical work have been carried out on heat transfer in cooling passage tube and duct.

Han *et al*.carried out combined effects of the rib angle-of-attack and the channel aspect ratio on the local heat transfer distributions in square and rectangular channels with two opposite rib-roughened walls for Reynolds numbers from 10,000 to 60,000. The rib angle-of-attack was varied from 90^0 to 60^0, to 45^0, and to 30^0, whereas the corresponding channel width-to-height ratio was varied from 1 to 2 and to 4, respectively. It was concluded that the highest heat transfer and the accompanying highest pressure drop can be obtained at $\alpha = 60^0$ in the square channel, the highest heat transfer and the highest pressure drop occur at $\alpha = 90^0$ in the rectangular channel with a channel aspect ratio of 4.

Dhanasekaran *et al*.carried out Computational analysis of mist/air cooling in a two-pass rectangular rotating channel with 45-deg angled rib turbulators. The results show that the mist cooling enhancement is about 30% at the trailing surface and about 20% at the leading surface of the first passage with 2% mist injection. In the second passage, 20% enhancement is predicted for both the surfaces.

Tatsumi *et al*.carried out detailed Numerical simulation for heat and fluid characteristics of square duct with discrete rib turbulators. It was observed that predicted stream wise distribution of Nusselt number is in good agreement with the experimental data for both arrays. Murata *et al*. carried out Effect of cross-sectional aspect ratio on turbulent heat transfer in an orthogonally rotating rectangular duct with angled rib turbulators. Heat transfer in a rotating rib-roughened rectangular duct was numerically simulated by using the large eddy simulation with a Lagrangian dynamic sub grid-scale model. It was observed that the effect of the rotation in the 60^0 rib-roughened duct induced the larger spatial variation in the local heat transfer, and the heat transfer was increased and decreased on the pressure and suction sides.

Liu *et al* present an experimental study on Heat transfer characteristics in steam-cooled rectangular channels with two opposite rib-roughened walls. Most of investigation, have focused on heat transfer characteristics for rib height and spacing ratio for transverse, angled, continuous or broken, square rib. In this present work, the numerical computation for 3-D turbulent flow over the W-discrete thin ribs mounted repeatedly on one side of heated wall of square duct. The main aim being to examine the change in heat transfer characteristics.

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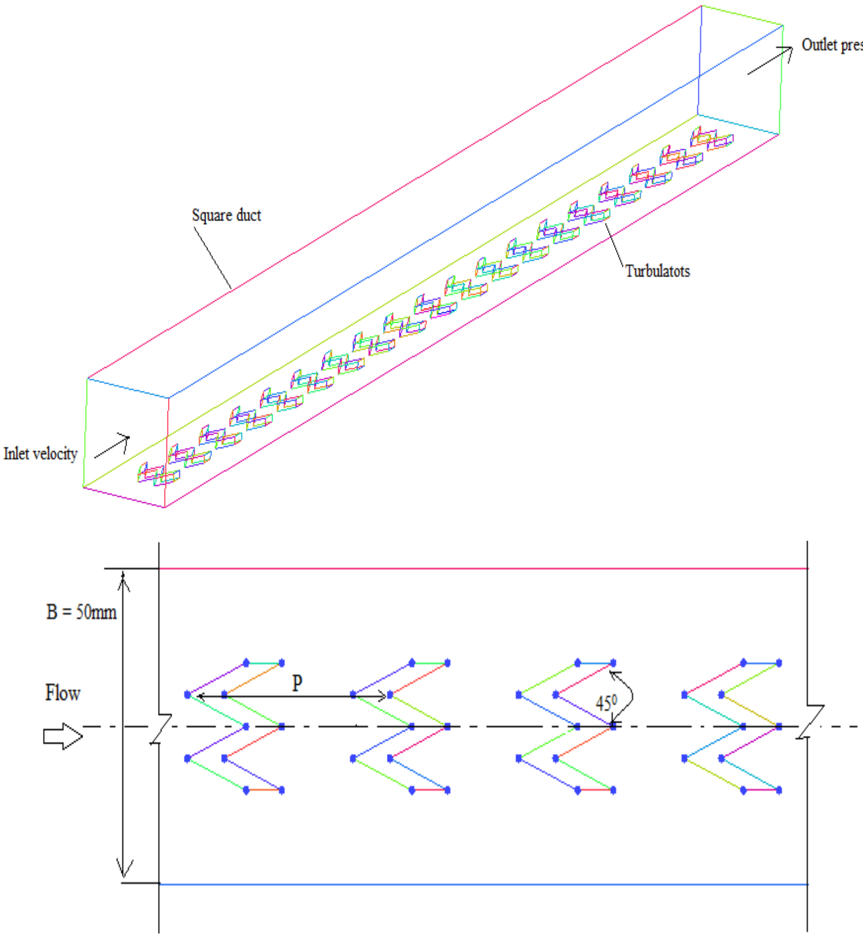


Fig.1 Duct Geometry (a) Duct with Turbulators (b) Turbulators domain

Nomenclature

A convection heat transfer area, m2
Dh hydraulic diameter of square duct
e rib height, m
f friction factor
H duct height, m
h convective heat transfer coefficient, Wm−2 K−1
k thermal conductivity of air, Wm−1 K−1
L duct length, m
Nu Nusselt number
p rib pitch spacing, m
P static pressure, Pa
Pr Prandtl number
PR pitch spacing ratio, p/D
Re Reynolds number, (ρu0D/μ)
T temperature, K

Greek letter

μ dynamic viscosity, kg s−1 m−1
ρ density, kg m−3

Subscript
In inlet
w wall

2. Geometrical model

The square cross section duct dimension are as follows duct hydraulic diameter (D*h*) = 0.05m. The roughness height of rib (e) = 0.0035m. Length of test section duct (L) = 0.7m. The geometrical modeling and duct has been generated in Ansys ICEM modeling tool. Fig. 1 gives the details of for duct geometry with turbulators inserts.

3. Flow configuration

The flow system interest is a horizontal square duct with W-broken ribs repeatedly placed on lower duct wall as depicted Fig. the duct divided into 3 section: entry, test section and exit. The details of test section (0.07m) full length ribbed duct shown in Fig. 1(a). The flow is fully developed. The air enters the duct at an inlet temperature T_{in}, and flow over a 45^0, 50^0 and 55^0 angled rib where e and s rib height and thickness, respectively.

4. Mathematical modeling

The numerical model for fluid flow and heat transfer in square duct is developed under following assumption:

1. Steady state three dimension flow and heat transfer
2. The flow is fully developed, turbulent and incompressible.
3. Constant fluid properties.
4. Body forces, viscous dissipation and radiation heat transfer are ignored.

All the simulation solved using a finite approach. For closure of equation, the k-ε was used in present study. The solution were converged when the normalized residual values were less than 10^{-4} for all variable but less than 10^{-6} only for the energy equation.

There are three parameter of interest in present study, Reynolds number, friction factor, and Nusselt number. The Reynolds number is defined as,

$$\mathrm{Re} = \frac{\rho V D h}{\mu} \tag{1}$$

The friction factor, *f* is computed by pressure drop, ΔP across the length of duct

$$f = \frac{2\Delta P L}{L V^2 D h} \tag{2}$$

The heat transfer is measure by the Nusselt number which can be written as

$$Nu = \frac{hL}{k} \tag{3}$$

5. Grid independence

The computation domain is resolved by regular Cartesian elements. A grid independence procedure was implemented by using Richardson extrapolation technique over grid with different number of cell 450,000 and 761,062. It is found that variation in hear transfer characteristics Nu and *f* values for inline 45^0 w-shaped ribs at PR=0.3 and Re=20000 is marginal when increasing the number of cell from 450,000 to 761,062. Hence, there is no such advantages increasing no of cell beyond this value. Considering both convergent time and solution precision the grid system of 761,062 cell was adopted for current simulation. For other ribbed duct, similar grid density was applied.

6. Computational mesh

For all geometries unstructured mesh was generated using Ansys ICEM. The fluid domain is descitized with unstructured hexahedral elements in order to more precise result. Fig. (1) Shows geometrical model of duct with insert ribs.

7. Boundary condition

For full length of ribbed duct, a uniform velocity introduced at the inlet while pressure outlet condition at the exit. The physical properties of air have been assumed to remain constant at initial temperature. Impermeable boundary and no-slip wall condition have been implemented over the duct wall as well as the rib surface apart from the enhanced wall treatment. The constant heat flux of one wall is maintained at 3160 w/m^2 while rib assumed at adiabatic wall condition.

8. Result and discussion

8.1 Validation

Verification of heat transfer and friction factor of smooth square duct. With no w-shaped rib insert is perforated by comparing value from previous correlation under similar condition as shown in Fig. 2. The present numerical smooth duct result is found to be in excellent agreement with Dittus-boelter and Blassius correlation with both Nusselt number and friction factor, within ±6.1 and ±10% maximum deviation respectively.

The correlation of the Nusselt number and friction factor for turbulent flow with constant heat flux conditions are as follows:

Dittus-boelter correlation,

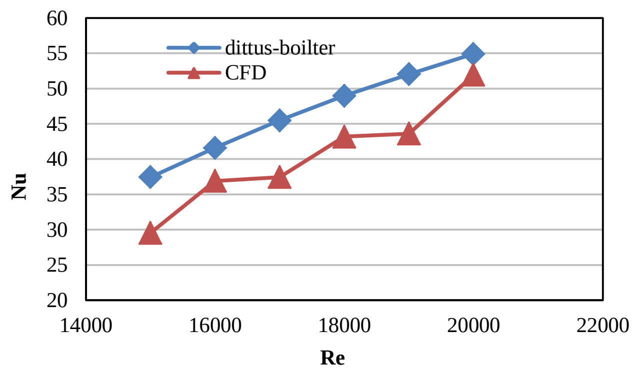
$$Nu = 0.023 Re^{0.8} Pr^{0.4} \tag{4}$$

Blassius correlation,

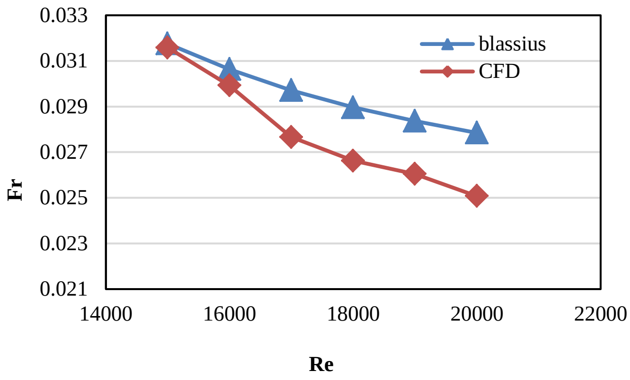
$$f = 0.423\ Re^{-0.275} \tag{5}$$

4.2 Heat transfer

Fig. 3. Display the display the variation of Nu with Re for rib angle (45^0, 50^0 and 55^0) with pitch ratio 0.4. It is visible that the Nu value tends to increase with rise of re for turbulators angle value. Fig. shows that the minor Nusselt number difference between turbulators angle of 50^0 and 55^0.



(a)



(b)

Fig.2 Validation of (a) Nusselt number and (b) friction factor for smooth square duct

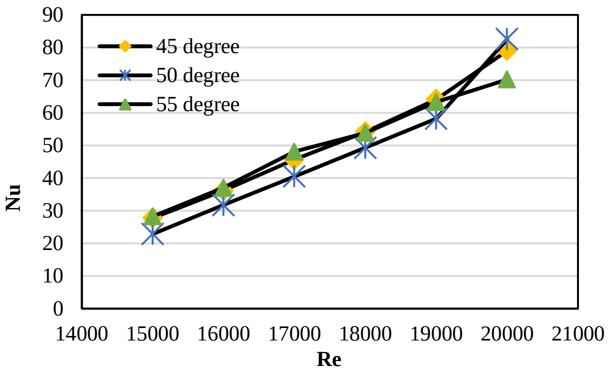


Fig.3 Effect of turbulators angles at PR = 0.4 on Nusselt number vs. Reynolds number

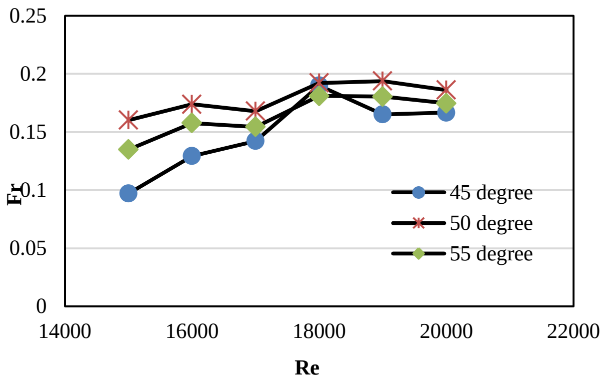


Fig.4 Effect of turbulators angles at PR = 0.4 on Friction factor vs. Reynolds number

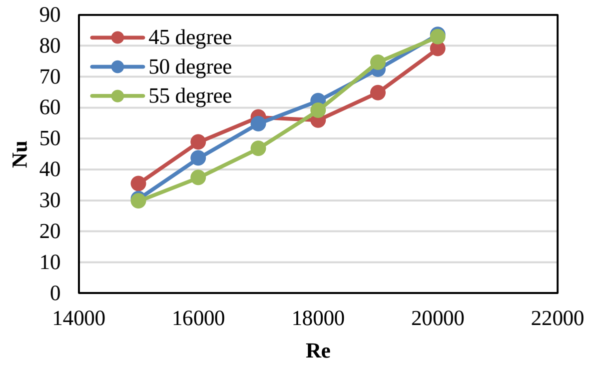


Fig.5 Effect of turbulators angles at PR = 0.3 on Nusselt number vs. Reynolds number

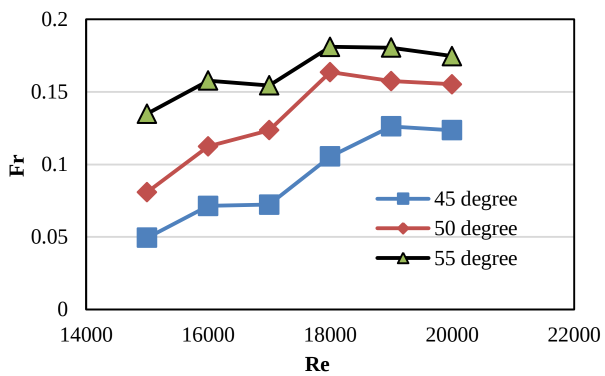


Fig.6 Effect of turbulators angles at PR = 0.3 on Friction factor vs. Reynolds number

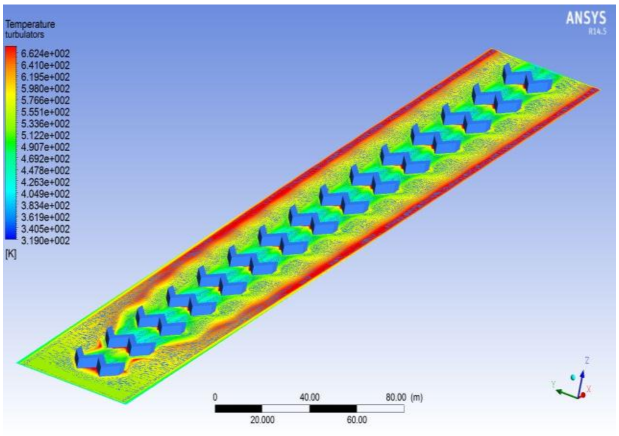


Fig. 7 shows the counter temperature plane for turbulators angle at 45^0 at PR= 0.3

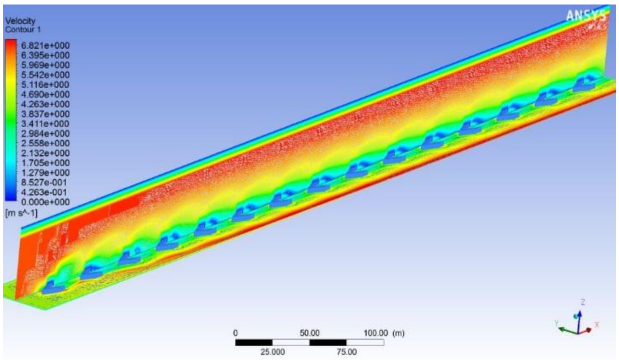


Fig. 8 shows the counter velocity plane for turbulators angle at 45^0 at PR= 0.3

Fig. 4. Display the display the variation of *f* with Re for rib angle (45^0, 50^0 and 55^0) with pitch ratio 0.4. It is noted that *f* tends to increase with rise of Re. The turbulators with PR = 0.4 gives highest friction factor value. Similarly Fig. 5 and Fig. 6 give the variation Nu and *f* with Re with pitch ratio 0.3. The temperature distribution of W-discrete ribs as shown in Fig. 7. Also the velocity distribution of W-discreet rib with vertical plane as shown in Fig. 8.

Conclusions

Fully developed turbulent flow thermal characteristics in square duct mounted repeatedly with three different angle W-discrete ribs are investigated numerically. From the numerical analysis of it is found that Nusselt number and friction factor in duct with W-rib insert increases. The Nusselt number 2.0 times above at higher Reynolds number and friction factor 15 to 20 times above than those in the smooth duct without insert at pitch ratio 0.4. Both pitch ratio of turbulator provide nearly similar heat transfer while 55^0 angle turbulator yields higher Nu and *f* value.

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