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NUMERICAL HEAT TRANSFER ENHANCEMENT IN SQUARE DUCT WITH INTERNAL RIB

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ABSTRACT - *This paper presents numerical investigation of heat transfer characteristics of horizontal square duct 500mm long using internal square-circle ribs of 7.5x7.5mm cross section ribs spacing =75mm, with air as the working fluid. Reynolds number= 34.267x10^{3} was taken. The steel duct (ASM4120) was subjected to different constant surrounding hot air temperatures (673, 773 and 873K^{o}). Heat transfer was enhanced by (ΔT=9, 13 and 15K^{o}) for using ribs at constant surrounding air temperature of (673, 773 and 873K^{o}) respectively. Increases surrounding hot air temperature increases coolant air temperature at the duct centerline. Increases surrounding hot air temperature having no effect on coolant air velocity. Using ribs enhances fluctuation in coolant air velocity and thus enhances heat transfer rate. All studies where carried out using workbench programFlUENT14.5 by using K-ε model.*

Keywords: cooling enhancement, heat transfer and turbulent flow, internal cooling.

1. INTRODUCTION

Enhancement of heat transfer is an active and important field of engineering research since it increases the effectiveness of heat exchangers. Suitable heat transfer techniques may achieve considerable technical advantages and savings of costs. There are various kinds of available techniques, adopted in many applications as heat exchangers for, automotive, process industry, solar heater etc. [Bergles et al, 1985, Bergles, 1998]. There are many enhancement techniques available. They identified thirteen different techniques, which can be segregated into two groupings: ’passive’ and ‘active’ techniques. Passive techniques employ special surface geometries, like coated surfaces, rough surfaces, extended surfaces or swirl flow devices, fluid additives for enhancement. The active techniques require external power sources, such as mechanical aids, electric or acoustic fields and surface vibration[Webb et al., 2005].

A number of investigations have been conducted using various rectangular channels with 180° turns. Metzger and Sham [Metzger et al., 1986] studied heat transfer effects around smooth rectangular channels with sharp, 180° turns. Carlomagno reported heat transfer measurements performed by means of infrared thermography in an internal flow through a 180° turn in a square channel, which is relevant to the internal cooling of gas turbine blades [Carlomagno, 1996].

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Arts et al. investigated the flow and heat transfer in a straight-ribbed cooling channel. They found that the pressure coefficient Cp increases before the ribs and decreases after the ribs[Arts et al., 1997].

Waleed Mohammed Abed and Mohammed Abed Ahmed used corrugated channel in the heat exchanger as these have high heat transfer efficiency and turbulent flow with low velocity. Some of the inventors studied flow and heat transfer of corrugated channel[Waleed et a., 2010].

TANG Xinyi and ZHU Dongsheng studied the turbulent flow and heat transfer enhancement in ducts or channels with rib, groove or rib-groove tabulators[Tang et al., 2012]. Hu and Shen carried out detailed distributions of internal heat transfer coefficients in a convergent cooling passage with a staggered array of 45° discrete ribs, and combination of ribs with grooves. For discrete ribs the area average enhancement factors were found to be 3-4 and that of combined discrete ribs with grooves were 2.5-3.2 [Rajendra et al., 2009].

In this paper, the effect of fitting (square-circle) ribs in square duct at constant surrounding hot air temperature on fluid flow and heat transfer will be investigated.

MATHEMATICAL FORMULATION

This study presents a description of the mathematical basis for a comprehensive general purpose model of fluid flow and heat transfer from the basic principles of conservation of mass, momentum, and energy. This leads to the governing equations of fluid flow and heat transfer used for the analysis of steady state, three-dimensional, turbulent and incompressible flow in addition to thermal performance and cooling effectiveness of pipes with different ribs configuration heat exchanger.

GOVERNING EQUATION

The heat-transfer rate is the amount of heat that transfers per unit time. If a hot bar has a surface temperature of T_o on one side and T_i on the other side, the basic heat-transfer rate due to conduction can be given by:

$$Q = UA \Delta T \qquad (1)$$

If a hot wall at a temperature T_o is exposed to a cool fluid at a temperature T_i on one side, the convective heat-transfer rate can be given by:

$$Q = hA \Delta T \qquad (2)$$

The conventional expression for calculating the heat transfer coefficient in fully developed turbulent flow in smooth pipes is the Dittus Boelter. Because of the many factors that affect the convection heat-transfer coefficient (h), calculation of the coefficient is complex. However, dimensionless numbers are used to calculate (h) for both free convection and forced convection.

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \qquad (3)$$

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D_h is the hydraulic diameter express as:

$$D_h = \frac{4A}{P_w}$$

Prandtl number express as:

$$\Pr = \frac{\mu c_p}{K} \qquad (4)$$

Reynolds number express as:

$$\mathrm{Re} = \frac{\rho u D}{\mu} \qquad (5)$$

Also, the flow Reynolds number is:

$$\mathrm{Re} = \frac{4\dot{m}}{\pi d \mu} \qquad (6)$$

Mass flow is defined in the following equation :

$$m = \rho u A \qquad \mathbf{(7)}$$

The energy which is produced by the heaters is carried away by the air flowing inside the pipe and only a small portion of the heat produced is lost by natural convection from the outer surface of the pipe. The energy balance:

$$Q = mc_p\left(T_o - T_i\right) \qquad (8)$$

where Q is the total heat transferred to air by forced convection and is given by:

$$q = Q / \pi DL \qquad (9)$$

Thus the heat transfer coefficient can be obtained:

$$h = q / (T_{s,x} - T_{b,x}) \qquad (10)$$

The pressure drop can be determined using:

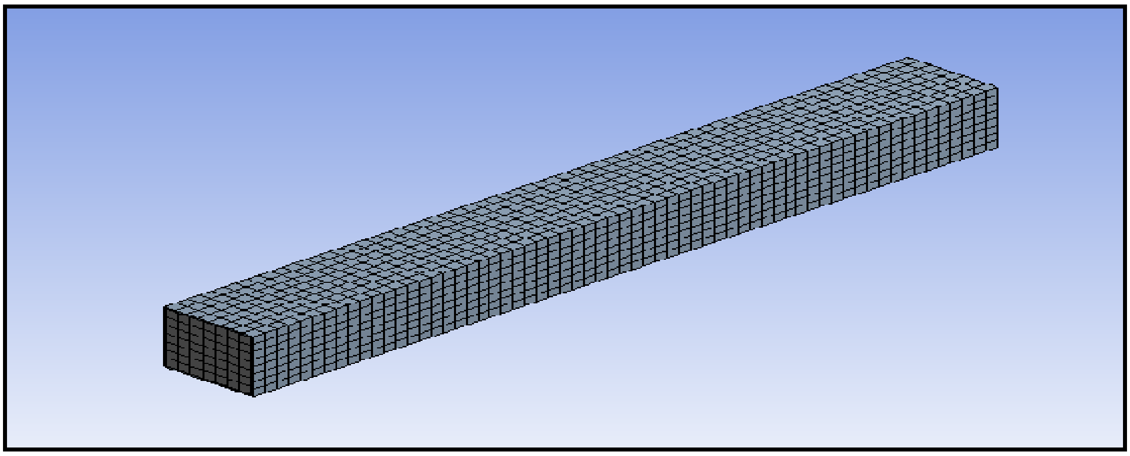
$$\Delta P = f \frac{L\rho u^2}{2D} \quad (11)$$

The friction factor for the smooth pipe can be calculated by:

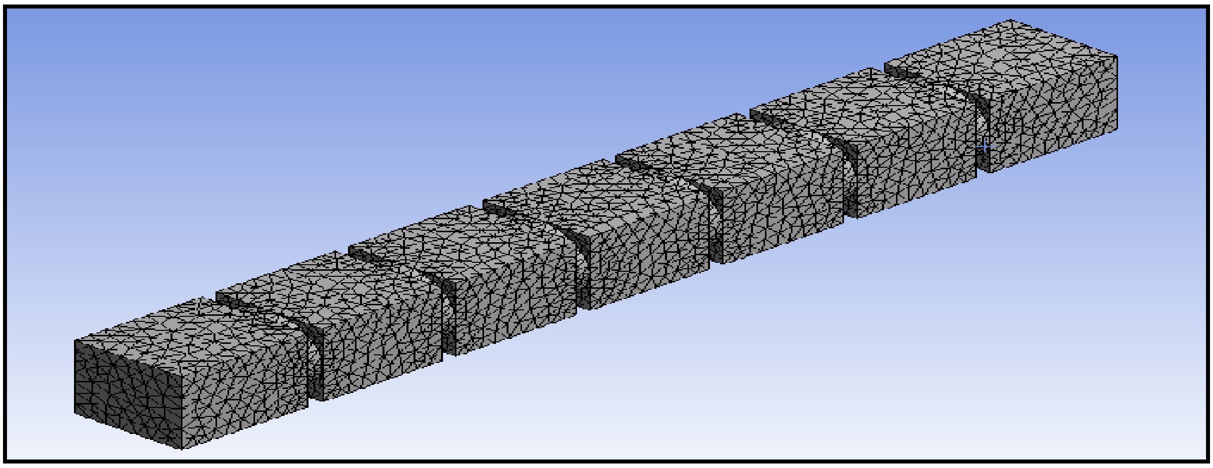
$$f_o = (0.790 \ln R_e - 1.64)^{-2} \quad (12)$$

MESH GENERATION

Volume meshing can be done by two ways of approaches in, structured and unstructured meshing. ANSYS-FLUENT can use grids comprising of tetrahedron or hexahedron cells in three dimensions. The type of mesh selection depends on the application as shown in fig. (1).



a) Smooth duct



b) Ribbed duct

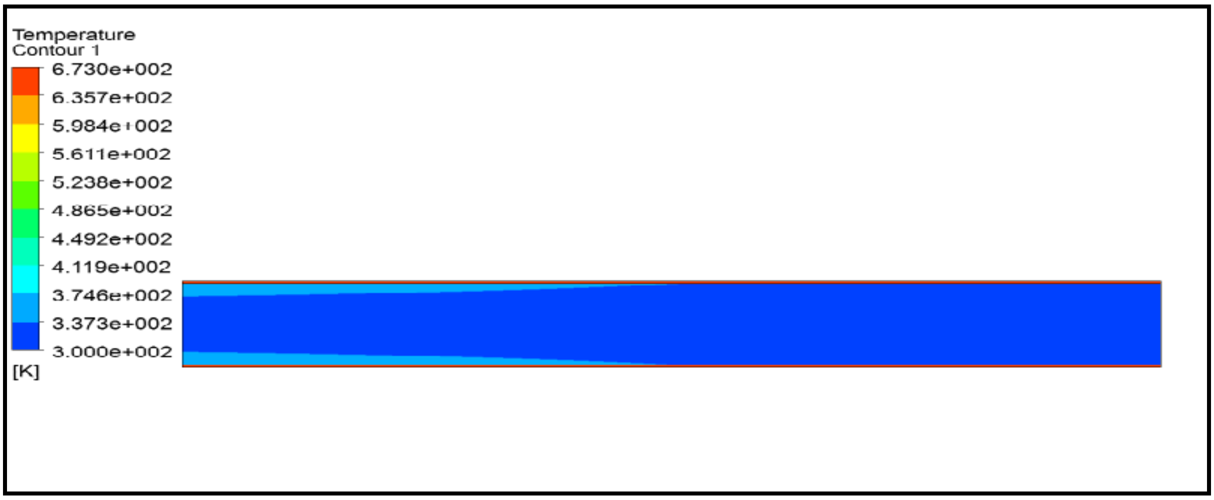
Fig.(1) Mesh generation: a) smooth duct, b) ribbed duct

2. RESULTS AND DISCUSSION

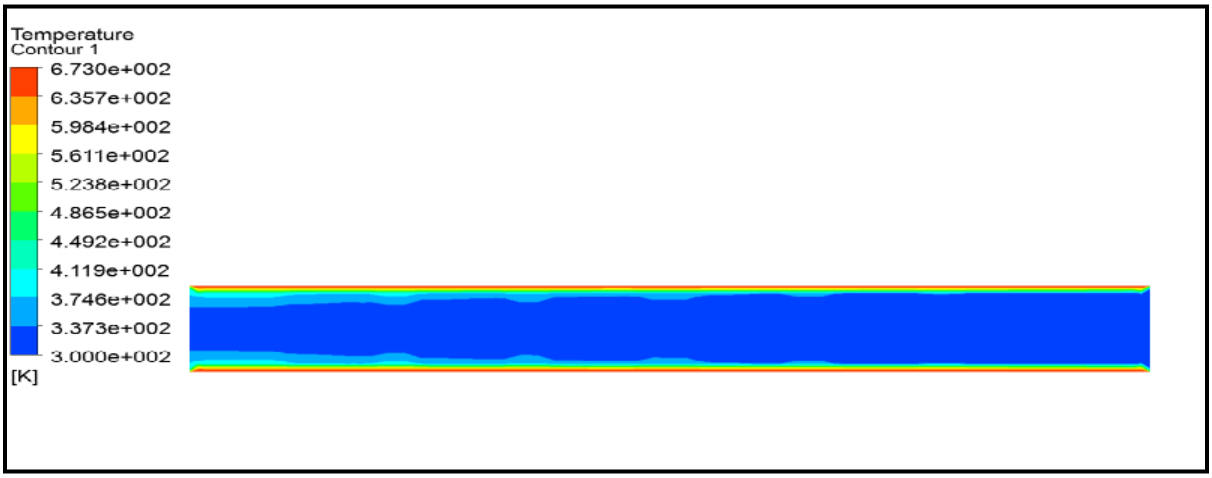
Temperature Distribution

Figures (1, 3 and 5) present contour of coolant air temperature distribution through the smooth duct at surrounding air temperature (673, 773 and 873K^o) respectively, and coolant air flow velocity of (Re=34.267x10^3). Cooling air temperature distribution at tube center line remains constant till the tube exit and increased near the wall.

Figures (2, 4 and 6) present contour of coolant air temperature distribution through ribbed duct at surrounding air temperature (673, 773 and 873K^o) respectively, and coolant air flow velocity of (Re=34.267x10^3). It was shown that at the entrance distance (till to mid) , temperature at the tube center line was remain un-effected and increasing near the wall , while the second half show a clear increase in temperature distribution this because of circulation generation and hence enhances heat transfer.



Figure(1) Contour of temperature distribution for smooth duct at surrounding hot air temperature of (673K^o)



Figure(2) Contour of temperature distribution for ribbed duct at surrounding hot air temperature of (673K^o)

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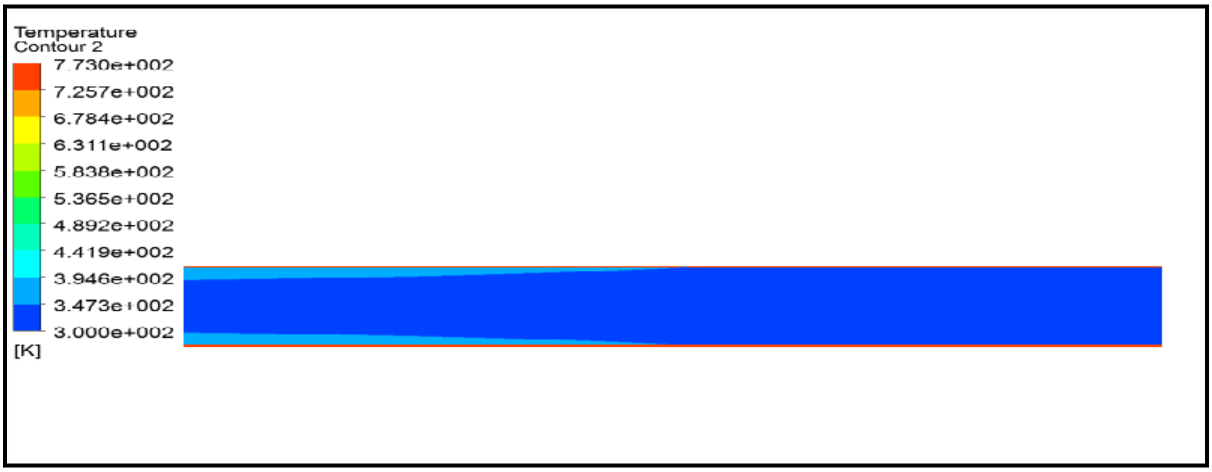
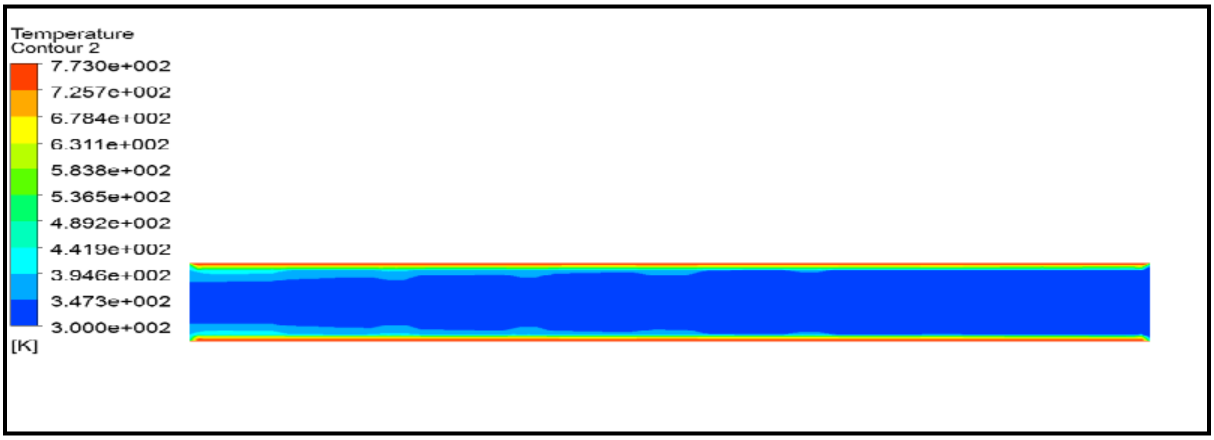
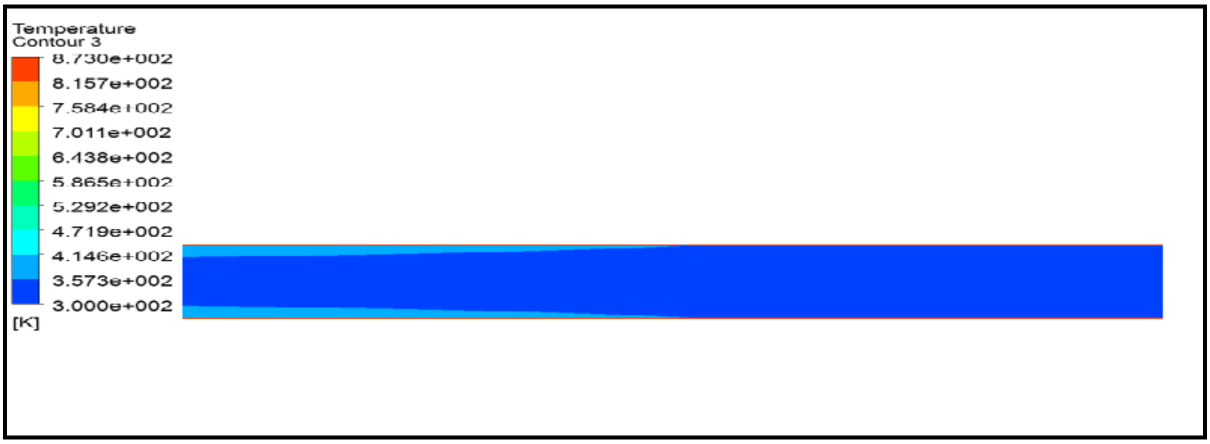


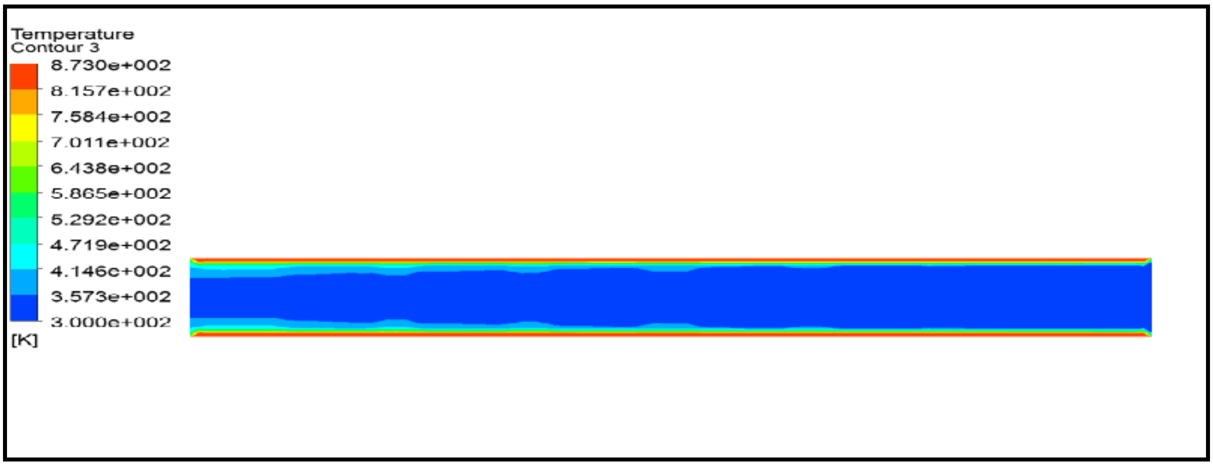
Figure(3) Contour of temperature distribution for smooth duct at surrounding hot air temperature of ($773K^{o}$)



Figure(4) Contour of temperature distribution for ribbed duct at surrounding hot air temperature of ($773K^{o}$)



Figure(5) Contour of temperature distribution for smooth duct at surrounding hot air temperature of ($873K^{o}$)

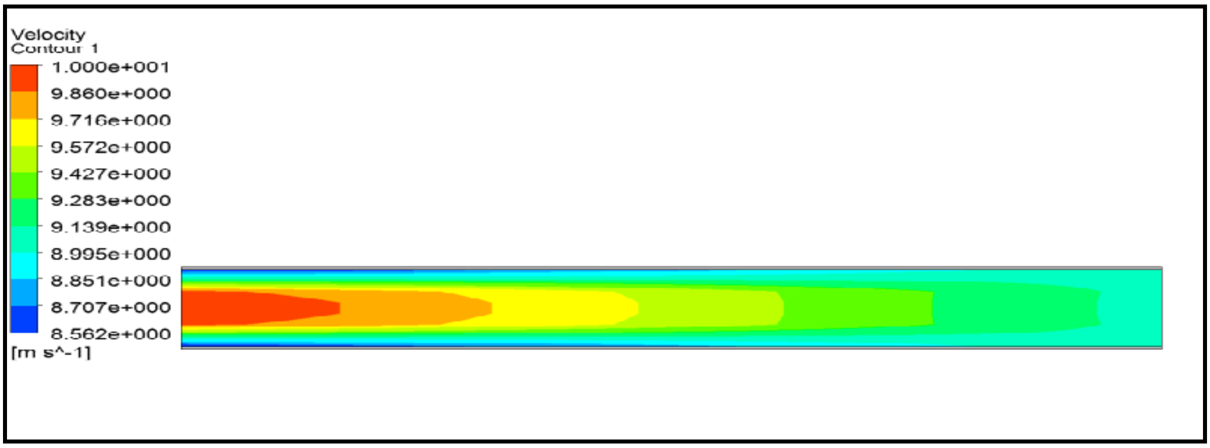


Figure(6) Contour of temperature distribution for ribbed duct at surrounding hot air temperature of (873K^o)

VELOCITY DISTRIBUTION

Figure (7, 9 and 11)) shows contour of velocity distribution through smooth duct at surrounding hot air temperature of (673,773 and 873K^o) respectively and coolant air flow velocity of (Re=34.267x10^3). It was shown that cooling air velocity through the duct remains constant till the tube exit.

Figures (8, 10 and 12) present contour of velocity distribution through ribbed duct at surrounding hot air temperature (673,773 and 873K^o) respectively and coolant air flow velocity of (Re=34.267x10^3), the coolant air flow velocity was accelerated and decelerated through the duct, due to contraction and expansion for using these ribs thus generating disturbance which enhances heat transfer rate.



Figure(7) Contour of velocity distribution for smooth duct at surrounding hot air temperature of (673K^o)

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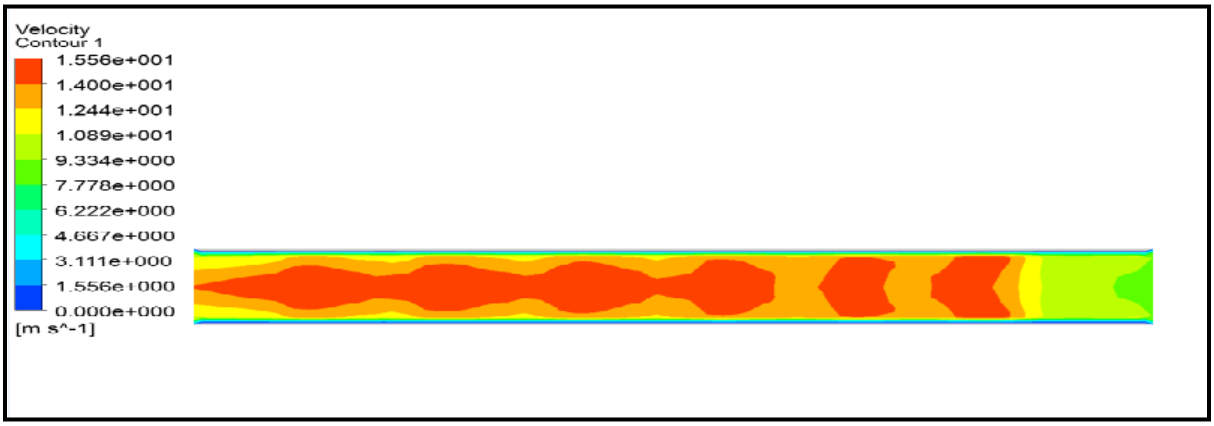
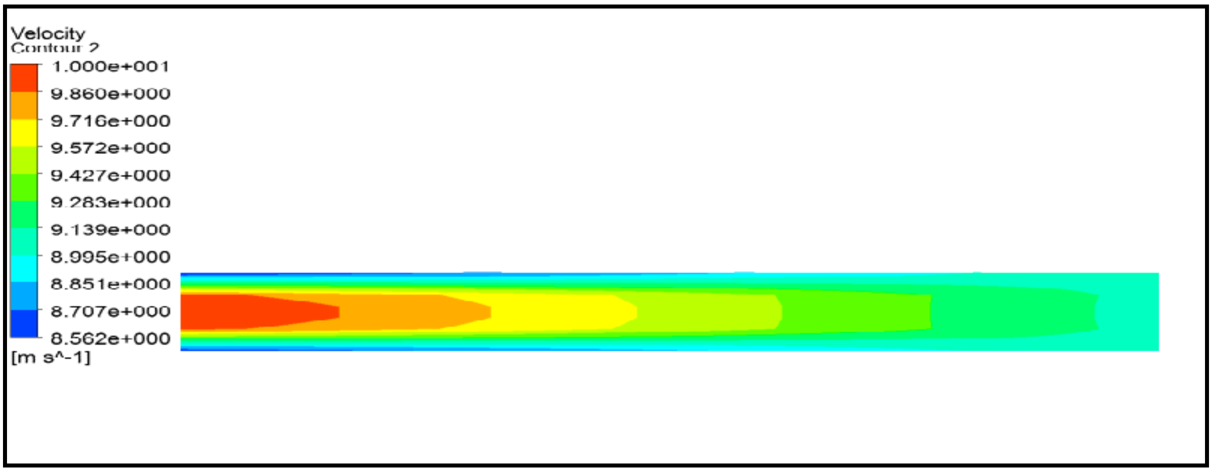
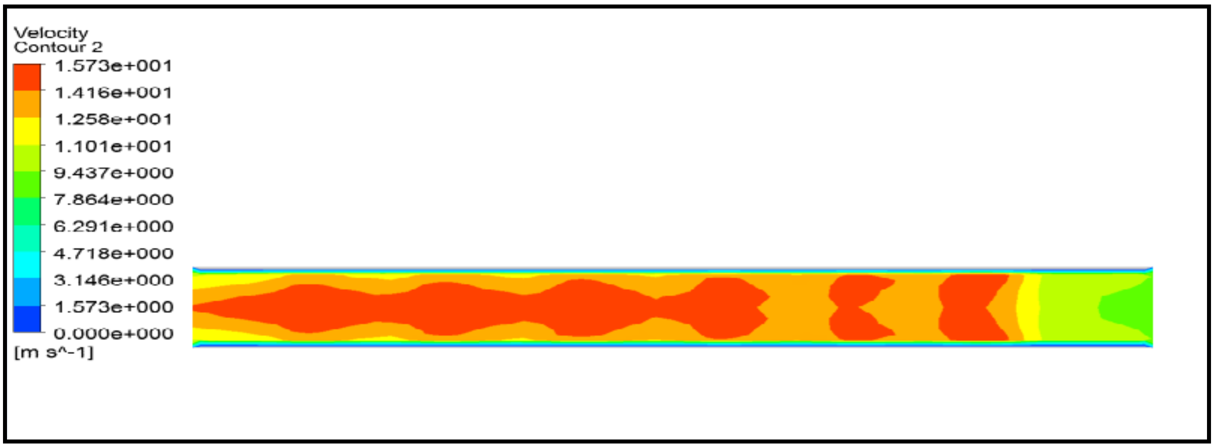


Figure(8) Contour of velocity distribution for ribbed duct at surrounding hot air temperature of (673K^o)



Figure(9) Contour of velocity distribution for smooth duct at surrounding hot air temperature of (773K^o)



Figure(10) Contour of velocity distribution for ribbed duct at surrounding hot air temperature of (773K^o)

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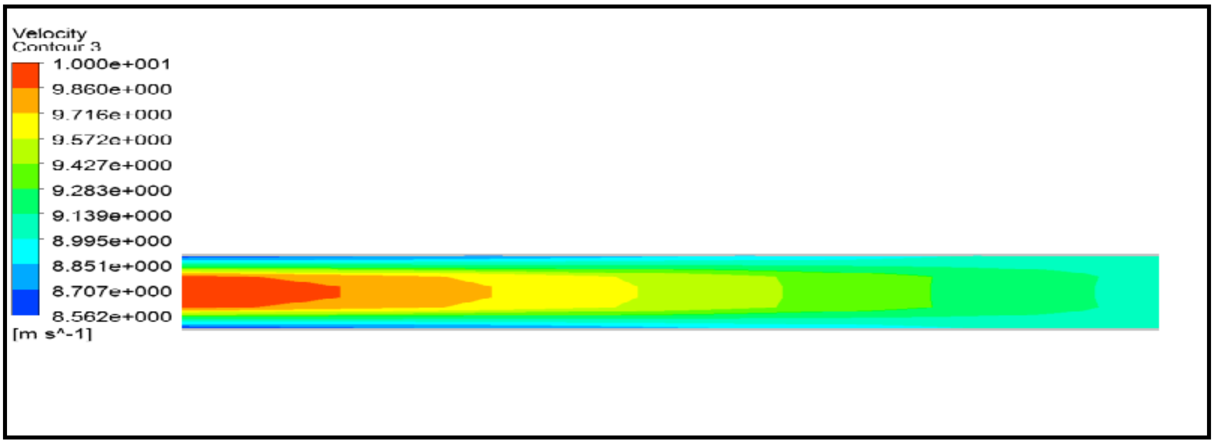
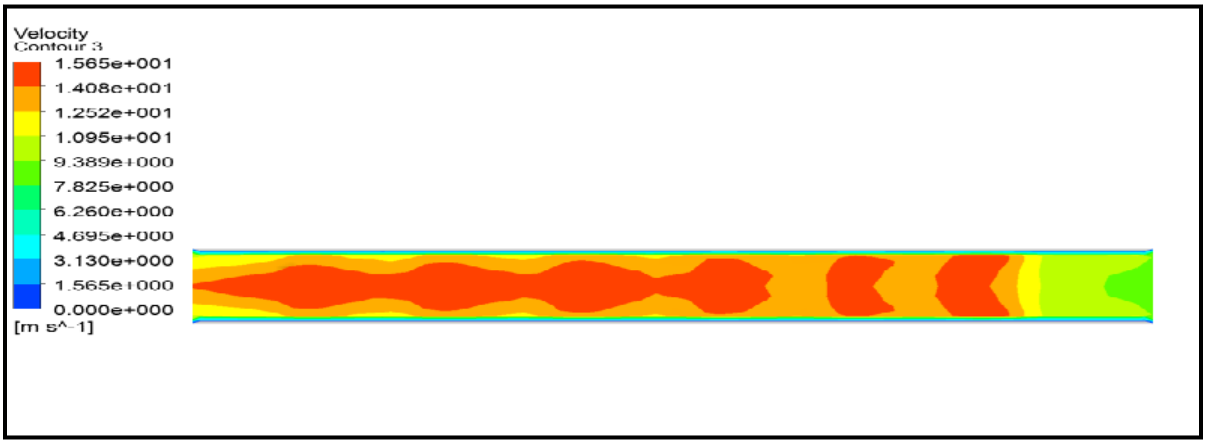


Figure(11) Contour of velocity distribution for smooth duct at surrounding hot air temperature of ($873K^o$)



Figure(12) Contour of velocity distribution for ribbed duct at surrounding hot air temperature of ($873K^o$)

Figures (13, 14 and 15) present temperature distribution at the ribbed duct centerline compared with smooth duct(without ribs) at surrounding hot air temperature of (673, 773 and $873K^o$) respectively and coolant air flow velocity of (Re=$34.267x10^3$). Cooling air temperature for duct with ribs be larger than smooth duct because ribs making wakes which developed to vortices this lead to increase the heat transfer from the tube wall to the coolant air i.e. increasing the coolant air temperature **,** the difference between the inlet and outlet temperature for smooth and ribbed tube is ($0.132K^o$) and ($9.021K^o$) respectively for fig.(13). Increases surrounding hot air temperature increases coolant air temperature at the duct centerline.

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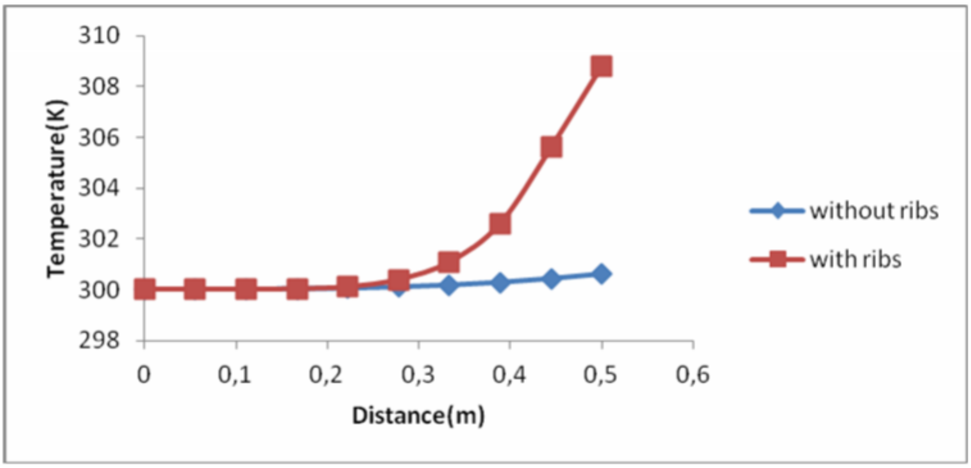
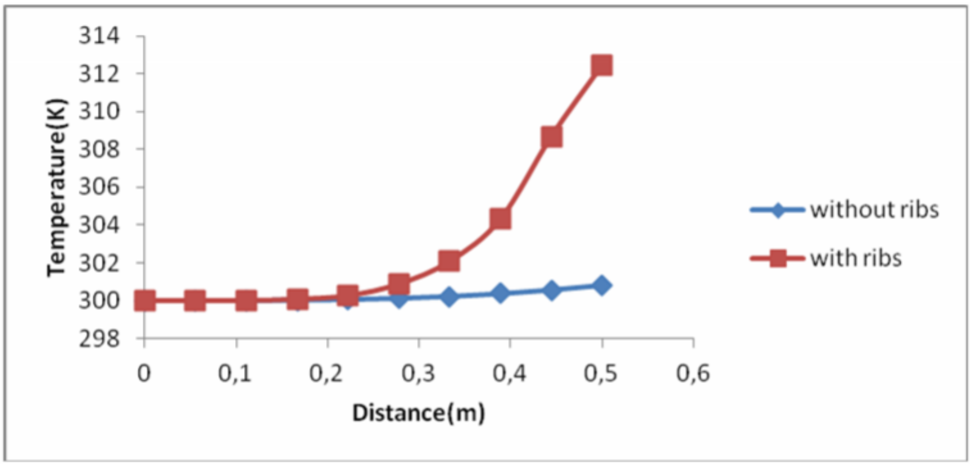
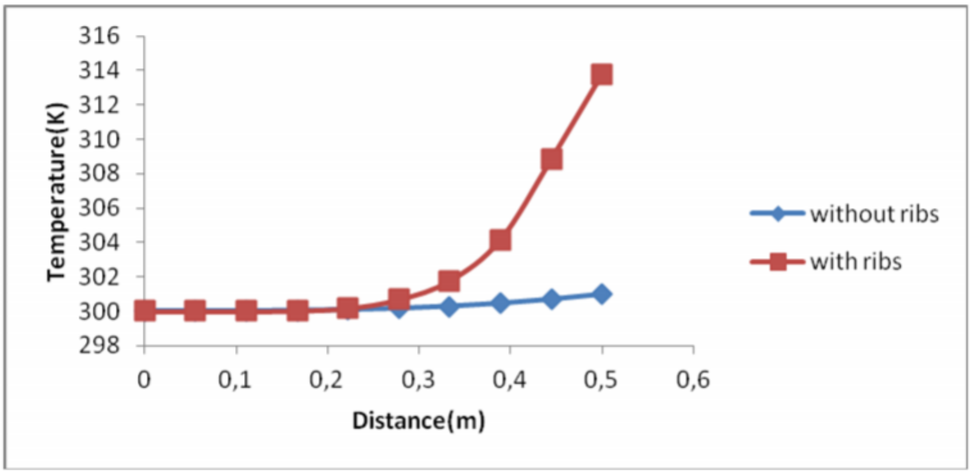


Figure(13) Coolant air temperature along the duct centerline at surrounding hot air temperature of(673K^{o})

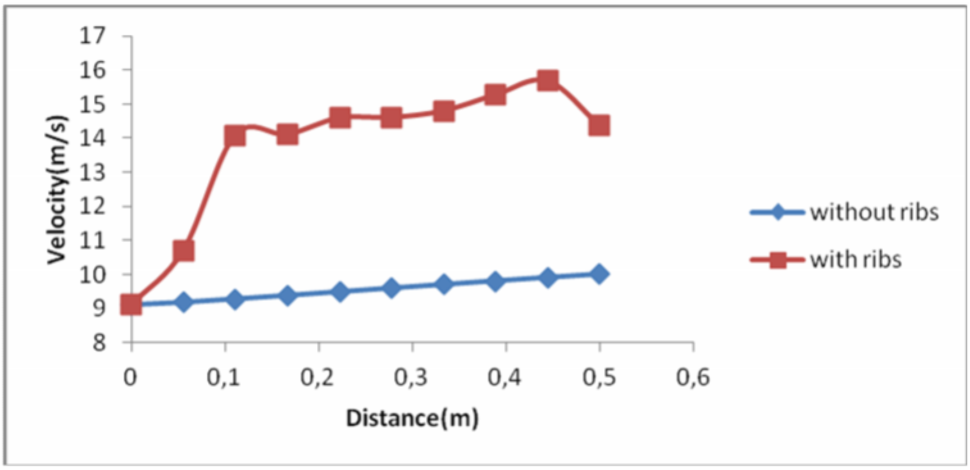


Figure(14) Coolant air temperature along the duct centerline at surrounding hot air temperature of(773K^{o})

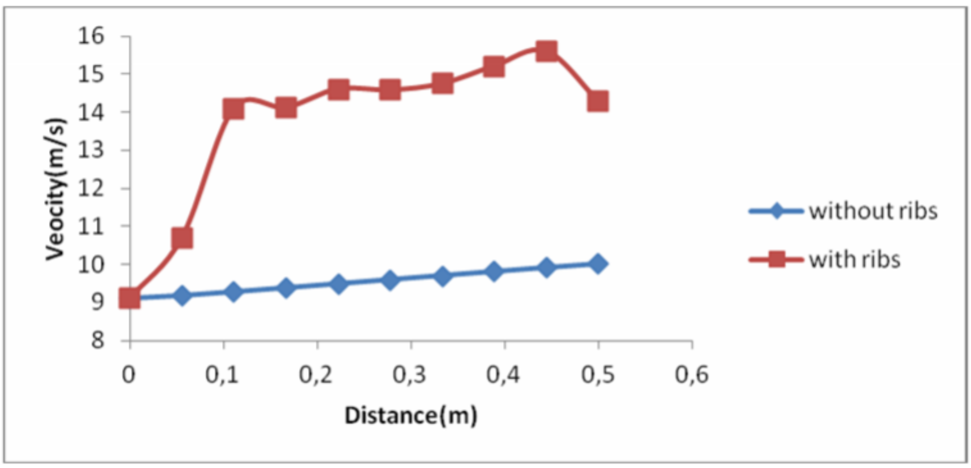


Figure(15) Coolant air temperature along the duct centerline at surrounding hot air temperature of(873K^{o})

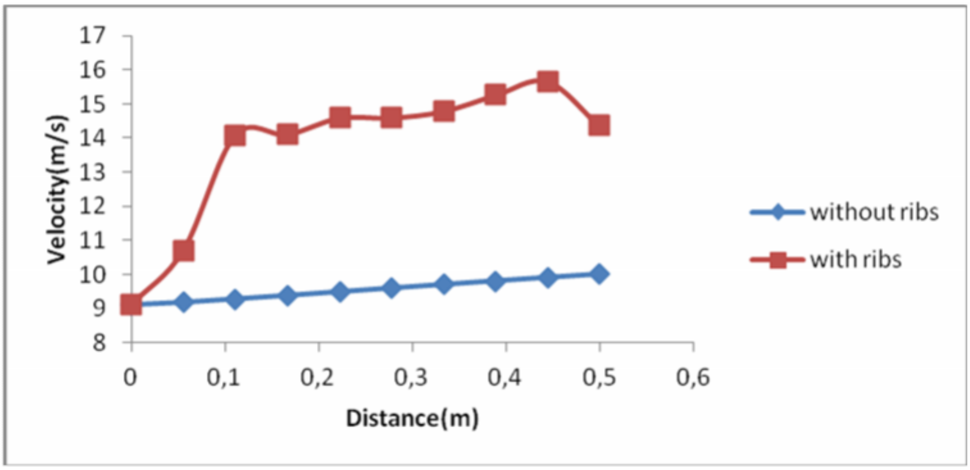
Figures (16, 17 and 18) present coolant air velocity along the ribbed duct centerline compared with smooth duct (without ribs) at coolant air velocity of ($Re=34.267x10^3$) , inlet coolant air temperature (300 K^o) and surrounding hot air temperature of (673, 773 and 873K^o) respectively. It was shown that increases surrounding hot air temperature having no effect on coolant air velocity.



Figure(16) Coolant air velocity along the duct centerline at surrounding hot air temperature of(673K^o)



Figure(17) Coolant air velocity along the duct centerline at surrounding hot air temperature of(773K^o)



Figure(18) Coolant air velocity along the duct centerline at surrounding hot air temperature of(873K^o)

3. CONCLUSIONS

From the present work we can deduce the following conclusions:

1. Heat transfer was enhanced by (ΔT=9, 13 and 15K^o) for using ribs at constant surrounding air temperature of (673, 773 and 873K^o) respectively and coolant air flow of (Re=34.267x10^3).
2. Using ribs enhances heat transfer rate more than the smooth duct.
3. Increases surrounding hot air temperature increases coolant air temperature at the duct centerline.
4. Increases surrounding hot air temperature having no effect on coolant air velocity.
5. Using ribs enhances fluctuation in coolant air velocity and thus enhances heat transfer rate.

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NOMENCLATURE

A Heat transfer area, m^2

Cp Specific heat of air, J/kg.K

D_h Hydraulic diameter, m

f Frictional factor

g Acceleration due to gravity ,m/s^2

h Heat transfer coefficient, W/m^2. k

K Thermal conductivity, w/m.K

m Mass flow rate, kg/s

Nu Nusselt number, dimensionless

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p Pitch of rib, m

Pr Prantdle Number, dimensionless

Q Heat transfer rate, W

R_e Reynold number, dimensionless

T Temperature, K^o

U Overall heat transfer coefficient , $w/m^2.K^o$

u Velocity of flow , m/s

b Bulk

c Center

i Inlet

o Outlet

s Surface

w Wall

μ Viscosity of air , N.s/ m^2

ρ Density of air, kg/m^3