Thermal Behavior in Dimple Square Duct with Inclined Perforated Baffles

Abeer Hashim Falih
Engineering College, University of Al-Mustansiyria /Baghdad
Email:Falihabeer@yahoo.com

Received on:8/3/2016 & Accepted on:22/6/2016

ABSTRACT

This research presents a study of heat transfer enhancement and pressure drop in a dimple square duct fitted with $\propto = 30^{\circ}$,60° and 90° inclined perforated baffles (baffles open area ratio of 26.17 %). The baffle to duct height ratio (w/a) of 0.3 and the baffle pitch to duct height ratio, PR=1, 2 are introduced in the present work. The tested duct has a constant wall heat flux condition. The experiments are carried out by varying airflow rate in terms of Reynolds number ranging from 1147.8 to 15304. The experimental data of heat transfer and pressure drop of the duct fitted with the inclined perforated baffles are compared with those of the dimple smooth duct under similar condition. The inclined baffles with PR=1 gives higher heat transfer rate than the one with PR=2 and the smooth duct respectively ,and the highest heat transfer and pressure drop is found by using baffle with 30° .

Keywords: Square duct , Nusselt number ,Friction factor, Inclined baffles**.**

INTRODUCTION

The use of baffle is one mode of the passive heat transfer enhancement techniques used in many engineering applications, including, internal cooling of turbine blades, heat exchangers ,cooling of electronics, and solar collectors. Baffles improve the heat transfer by providing an additional heat transfer surface area and promote turbulence. The presence of baffles causes the flow to separate , reattach and create reverse flow. Some parameters that cause the heat transfer and pressure drop increase such as , pitch ratio, blockage ratio, orientation , geometry of baffle and the Reynolds number.

Various researches about heat transfer enhancement with baffles have been studied . Dutta and Hossain [1], investigated experimentally of the local heat transfer characteristics and the frictional head loss in a rectangular channel with two inclined solid and perforated baffles of the same overall size on. The upstream baffle is attached to the top heated surface, while the orientation, position, and the shape of the other baffle is varied in order to attain the optimum configuration for augment heat transfer. Results showed that the local Nusselt number distribution is dependent on the orientation, position, and geometry of the second baffle plate.

Monsak Pimsarn, et al. [2] investigated the thermal performance of rectangular channel with Z-shaped ribs. These ribs were set on the rectangular duct at 30°,45°,60° of flat rib was set at 90° relative to air flow directions. These ribs were fitted in Z-shape (Z-rib) aligned in series on whole surface of upper plate. The constant heat flux was provided to top surface only .The comparison of the result of Z-ribs with 30°,45°,60° and flat rib with same rib height ,pitch ratio and smooth channel is done. The thermal enhancement factor of all Z-ribs are higher than flat rib. The 45º Z rib provide highest increase in heat transfer rate and best thermal performance.

Promvonge et al. [3] numerically studied of three dimensional laminar periodic channel flows over a 45° inclined baffle fitted only on the bottom surface of the square-channel .The results show that the enhancement of heat transfer is about 2–3 times higher than that for using the 90°

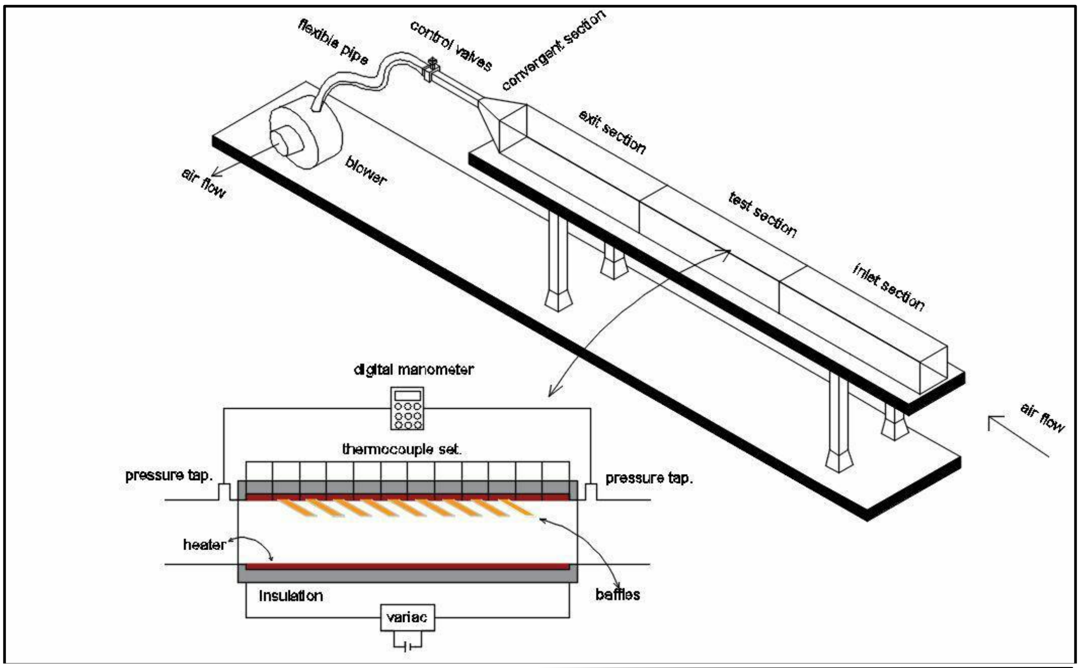
baffle while the friction loss is some 10–25% lower. Sara et al.[4] found that the open area ratio was increased by increasing the diameter of the pores is more effective than the increase in number of pores.

In this paper experimental analysis of heat transfer enhancement is presented with different inclined perforation baffles(baffles open area ratio of 26.17 %) with PR=1 and 2 to achieve optimum inclination condition in dimple square duct.

Experimental Work

A schematic of the experimental set up is shown in Fig. (1). It consists of a test duct with entrance and exit sections, a blower, control valve to regulate the mass flow rate and various devices for measurement of temperature and fluid head. The entrance length is 60 cm (10 d_h) [5] long entry region was provided so that flow would be fully developed as it enters the test section.

The test section is include the dimple duct has a (6*6 cm) square cross-sectional unit and a length of 60 cm (Lt).The duct has been manufacturing from copper with 0.5 mm thickness and made in the local workshop from dimpled wall. Figs.(2&3) show a schematic diagram of geometry of the duct wall and the tested duct respectively.



Figure(1) : Schematic diagram of apparatus.

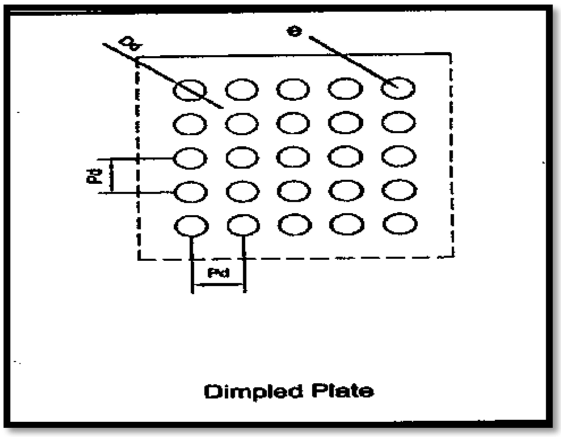
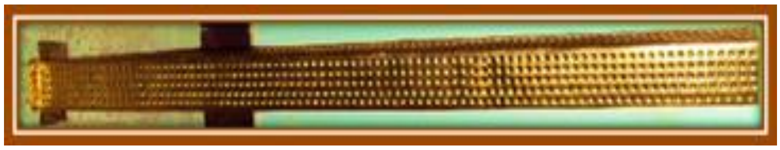


Figure (2) : A schematic diagram of geometry of the duct wall.



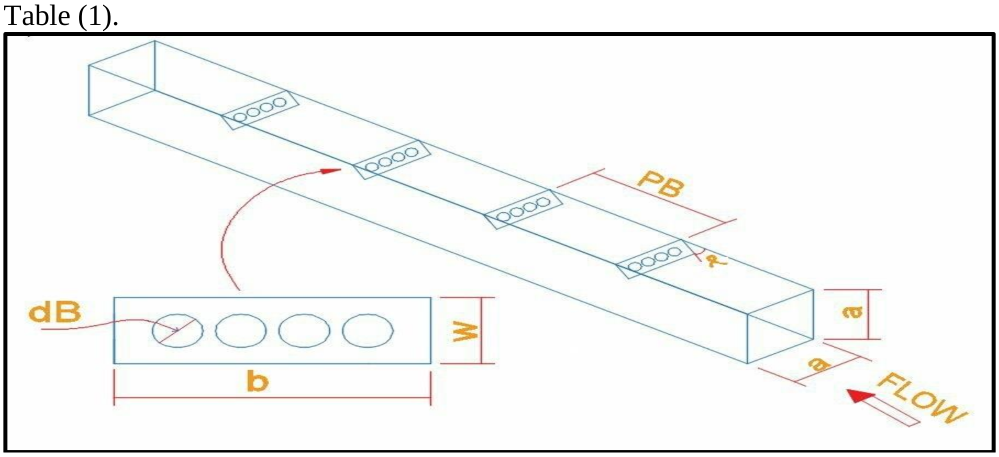
Figure(3): The tested duct .

The baffles insert are set on the top wall of the dimple square duct at 30º,60 ºand 90 º relative to air flow directions as shown in Fig.(4).

The perforated baffles pitch ratio (PR) is defined as the ratio of the distance between two adjacent baffles to the duct height (P_B/a) while the open area ratio β is the ratio of the total pores area to the frontal area of baffle is calculated from [6]:

$$\beta = n\,[(\pi/4)\, d_B^2 / b\,w] \qquad \ldots\ldots\ldots(1)$$

where n is the number of pores in a baffle and d_B is the pore diameter.
The plate-type heater is used for heating all walls of the test section in order to maintain a uniform surface heat flux. All details of the dimple square duct with baffles are demonstrated in Table (1).



Figure(4) : A schematic diagram of square duct with inclined perforated baffles.

The surface temperatures (Ts) on the principal upper, lower and side walls are measured by 12 thermocouples type K located along the test section as shown in Fig.(5). To measure the inlet

and outlet bulk temperatures by type K thermocouples are positioned upstream and downstream of the test duct. Two static pressure taps are fitted at the top of the duct wall to measure axial pressure drops across the test section, which used to evaluate the friction factor. The pressure drop is measured by using digital manometer . The velocity of air is measured by digital vane-type anemometer . The Reynolds numbers for the air flowing through the test section are controlled in the range of 1147.8 to 15304 for laminar and turbulent flow region.
The steady state is reached when the temperature of inner duct surface and temperatures of air changed about is 0.2 ^oC through 15 minutes. Then, the data of the duct surface, inlet and exit air temperatures, and the manometers readings are recorded.

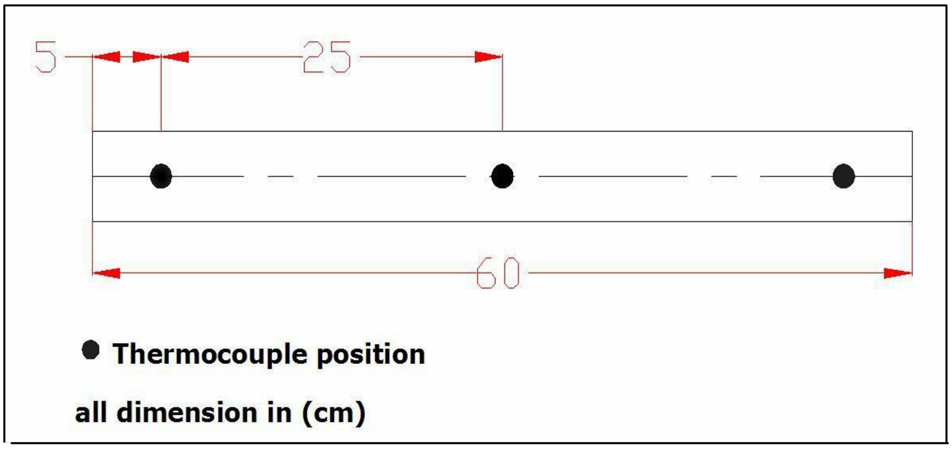


Figure (5): Positions of the thermocouples on the test section.

Data Reduction
The heat transfer coefficient between the air and the heated duct is calculated from [7] :
$h = Q/ \{A_s (T_{sm} – T_{fm})\}$ …………(2)

where the heat transfer rate ,Q, to the air is calculated from [7],
$Q = \dot{m}\, cp\, (T_o – T_i)$ …………..(3)

Where $A_s = (4\ a.\ L_t)$ is the surface area of heat transfer , in this study the baffles surface area is neglected. In all calculations, a mean duct surface temperature T_{sm} is calculated from :
$T_{sm}=\sum T_{sm} / 12$
T_{fm} is the bulk mean air temperature $(T_i + T_o)/ 2$
The heat transfer coefficient used to calculate the Nusselt number is [7] ,
$Nu= h\ d_h/k$ …….(4)
where
d_h : is the hydraulic diameter of square duct
$= 4A_c / p = 4a^2/4a = a$ ……(5)

And the Reynolds number was calculated from [7],
$Re = \rho u d_h/\mu$ ………(6)

The friction factor is calculated over the test section length as [7]:
$f = 2\Delta p d_h/\rho\ L_t\ u^2$ ……… (7)
where
Δp is the pressure drop over the test section

And u is the average velocity.
The thermal enhancement factor is defined as the ratio of the heat transfer coefficient of the enhanced surface to that of a smooth surface at the same pumping power and given by [8]:

$$TEF = (Nu/Nu_s)/(f/f_s)^{1/3} \qquad \text{……… (8)}$$

Table (1): Baffles and test duct details.

Dimpled square duct	
Pitch length of dimpled, Pd	11 mm
Diameter of dimpled, Dd	5.4 mm
Depth of dimpled, e	2.3 mm
Duct thickness	0.5mm
Hydraulic diameter of ducts, dh	60 mm
Test section length, Lt	600 mm
Side length of square duct, a	60 mm
Baffles	
Baffles thickness, δ	0.5 mm
Height, w	20 mm
Pitch, p_B	60mm, 120 mm
Pitch Ratio,PR	1, 2
Baffle thickness to height ratio ,δ/w	0.025
Baffle to duct height ratio ,w/a	0.333
Diameter of the pore	10 mm
open area ratio β	26.17 %
Number of baffles	4 , 8

Validation of Experimental Data

At first the Nusselt number and friction factor data for square duct are collected and then compared with those calculated from Dittus– Boelter equation for the Nusselt number and Blasius equation for the friction factor.
The Nusselt number for flat smooth duct given by the Dittus–Boelter equation is [7]:

$$Nu_s = 0.023Re^{0.8}\, Pr^{0.4} \qquad \text{………... (9)}$$

The friction factor for flat smooth duct given by Blasius equation is [7]:

$$f_s = 0.316\, Re^{-0.25} \qquad \text{………… (10)}$$

Figs. (6) and (7) show the comparison of the experimental data of the Nusselt number and friction factor for flat smooth duct with those from equations as a function of the Reynolds number, respectively .The figures show that the present work agrees with the available correlations.

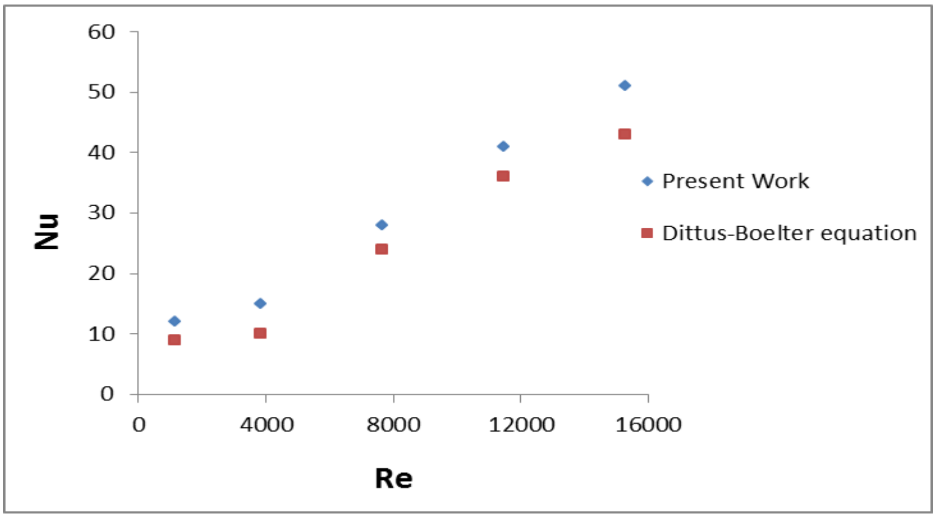


Figure (6):Nusselt number as function of Reynolds number for square duct.

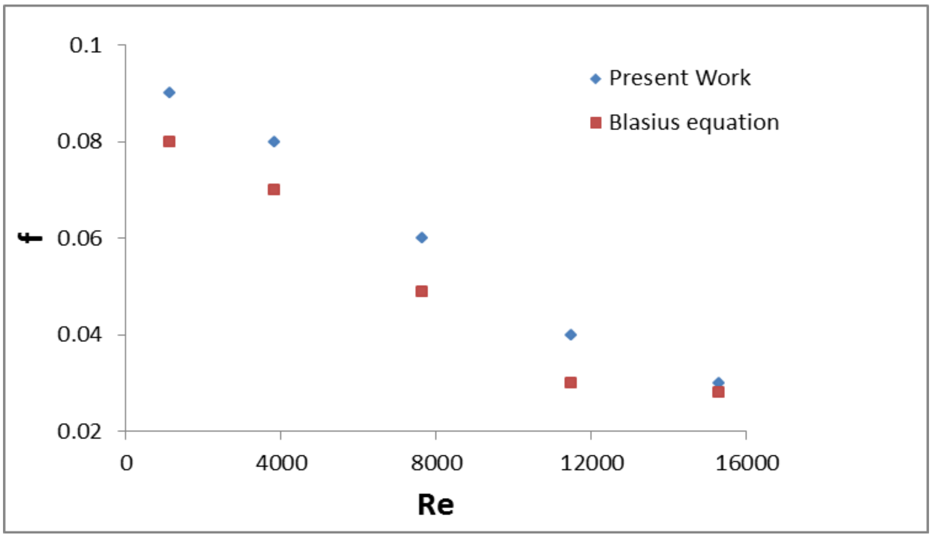


Figure (7): Friction factor as function of Reynolds number for square duct.

RESULTS

The effect of perforated baffle inclination on heat transfer ,friction factor and thermal performance factor, are demonstrated in Figs. 8 –12.

The Nusselt numbers for all cases are presented under laminar and turbulent flow conditions are shown in Fig.(8) . It's clear from this figure, the dimple square duct fitted with baffles yield considerable heat transfer rate with a similar trend in comparison with the smooth duct and the Nusselt number increases with the rise of Reynolds number. This is because the dimple duct roughness combined with perforated baffles interrupt the development of the boundary layer of the fluid flow and increase the turbulence degree of flow.

Fig. (9) shows the variation of Nusselt number ratio with Reynolds number. The Nusselt number ratio, Nu/Nu_s is defined as a ratio of augmented Nusselt number in each case study to Nusselt number of dimple smooth duct . In the figure, the Nusselt number ratio tends to decrease slightly with Reynolds number rise from 1147.8-15304 for all of studied cases. The maximum Nusselt number ratio values are found to be about 3.3 and 2.7 time over the dimple smooth duct for using the 30° inclined perforated baffles with PR = 1and 2 respectively. The 60° inclined perforated baffles give the maximum Nusselt number ratio are found to be about

3.07 and 2 time at PR = 1and 2 respectively and for 90° inclined perforated baffles the maximum Nusselt number ratio is about 2.5 and 1.7 time over the dimple smooth duct.

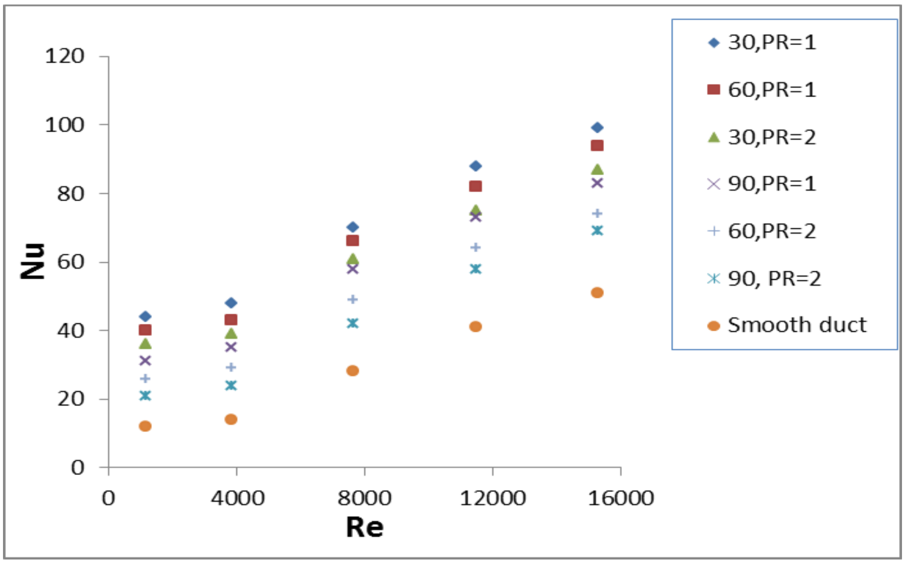


Figure (8) :Variation of Nusselt number with Reynolds number.

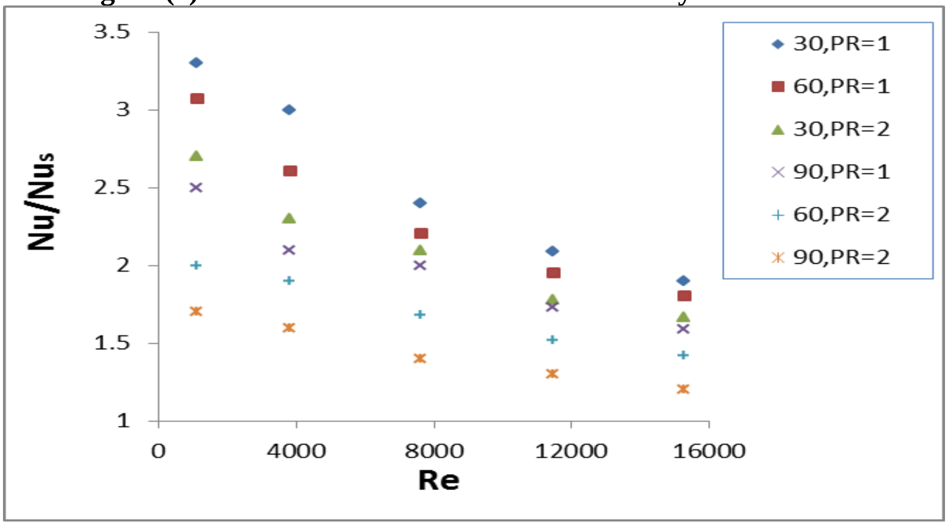
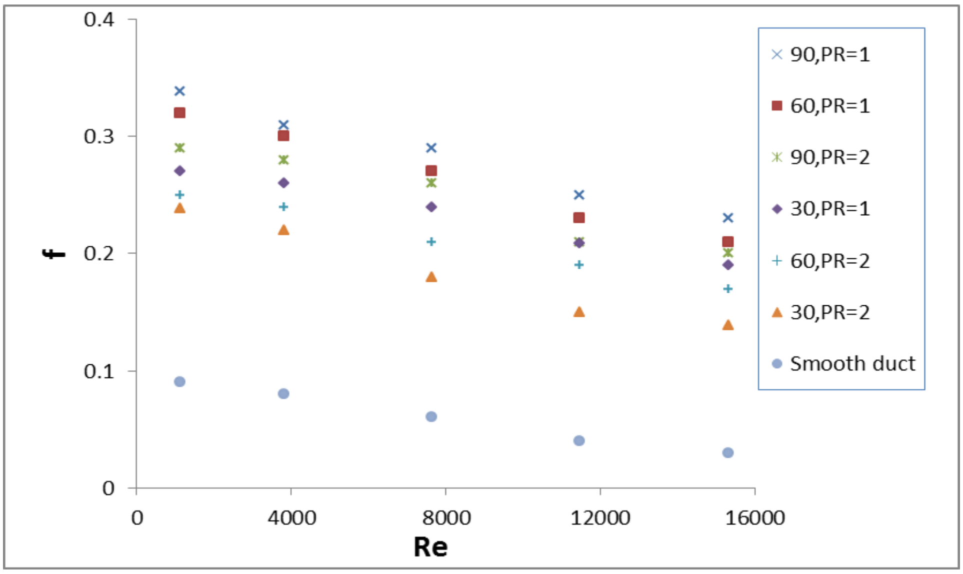


Figure (9): Variation of Nusselt number ratio, Nu/Nu_s with Reynolds number.

Fig. (10) shows the effect of using the perforated baffles turbulators on the pressure drop across the tested duct as shown in terms of friction factor. In the figure, it is apparent that the use of perforated baffles turbulators leads to a substantial increase in friction factor over the dimple smooth duct.

Fig. (11) show the variation of friction factor ratio with Reynolds number. from this figure it is noted that the friction factor ratio is increased with the rise of Reynolds number. The maximum friction factor ratio is found to be about 6.33and 4.6 time over the dimple smooth duct for using the 30° inclined perforated baffles with PR = 1and 2 respectively. The 60° inclined perforated baffles have given the maximum friction factor ratio values are found to be about 7 and 5.5 time at PR = 1and 2 respectively and for 90° inclined perforated baffles the maximum friction factor ratio values are found to be about 7.6 and 6.66 time over the dimple smooth duct.



Figure(10) :Variation of Friction factor with Reynolds number.

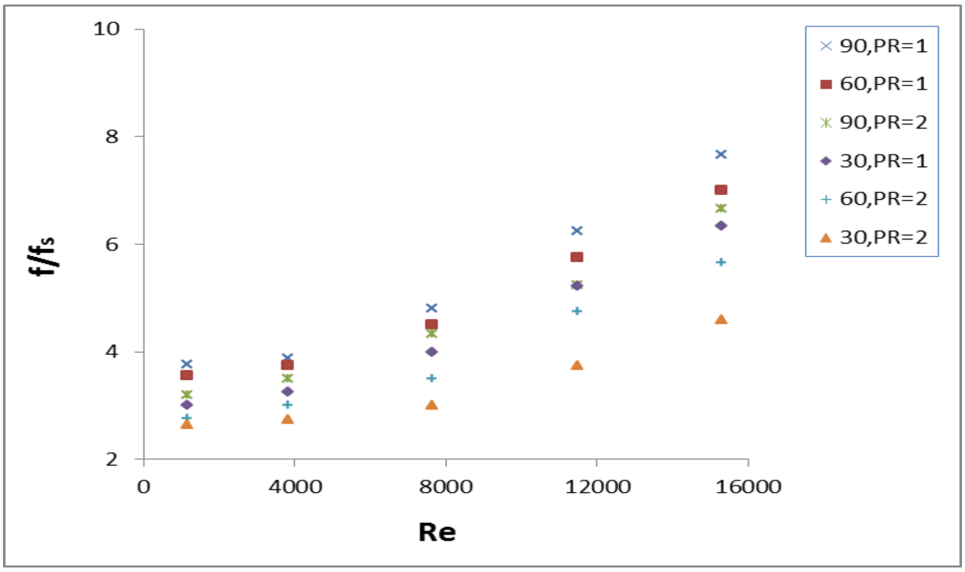
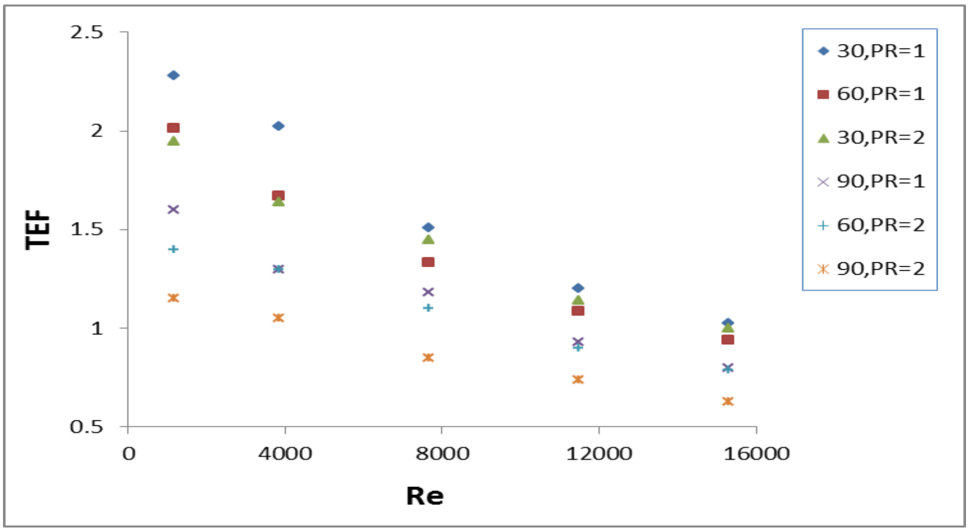


Figure (11): Variation of Friction factor ratio, f/f_s with Reynolds number.

The thermal enhancement factor for the duct with various perforated baffles fitted is compared at the same pumping power in Fig. (12). Apparently, the performance factor tends to decrease with the increasing Reynolds number . The maximum thermal enhancement factor is found about of 2.28 when using the 30° inclined perforated baffles with PR = 1.



Figure(12): Variation of thermal enhancement factor with Reynolds number.

CONCLUSIONS

In this work, the enhancement of heat transfer, friction factor and thermal enhancement factor have been investigated in a dimple square duct for the laminar and turbulent regime, Reynolds number ranging from 1147.8-15304. The experimental results are compared between the duct mounted with inclined perforated baffles and the dimple smooth duct. The duct fitted baffle gives higher heat transfer rate and friction factor than the smooth duct. The inclined perforated baffles with PR=1 gives higher heat transfer rate than one with PR= 2 and the smooth duct respectively, and the highest heat transfer and pressure drop is found by using baffle with 30° .

REFERENCES

[1] Dutta, P., and Hossain, A. , "Internal cooling augmentation in rectangular channel using two inclined baffles", International Journal of Heat and Fluid Flow, 26(2), pp. 223-232, 2005.

[2] Monsak Pimsarn, et al, “Augmented Heat Transfer in Rectangular Duct with Angled Z-Shaped Ribs”, PEA-AIT International Conference on Energy and Sustainable development: Issues and Strategies (ESD 2010) The Empress Hotel, Chiang Mai, Thailand. 2-4 June 2010.

[3] Promvonge P., Sripattanapipat S., Tamna S., Kwankaomeng S., Thianpong C., “ Numerical investigation of laminar heat transfer in a square channel with 45° inclined baffles”, Int. Commun. Heat Mass Transfer, Vol.37. pp. 170–177, 2010.

[4] O.N. Sara,T.Pekdemir,S.Yapici,M. Yilmaz, “Heat transfer enhancement in a channel flow with perforated rectangular blocks”, Int. J.of Heat and Fluid Flow,Vol.22,PP.509-518, 2001.

[5] Adrian Bejan and Allan D. Kraus , Handbook of “ Heat Transfer”,Canada, 2003.

[6] Rajendra Karwa, B.K. Maheshwari, Nitin Karwa, “ Experimental study of heat transfer enhancement in an asymmetrically heated rectangular duct with perforated baffles”, International Communications in Heat and Mass Transfer ,Vol.32.,pp. 275, 2005.

[7] Incropera, F., and Dewitt, P. D., "Introduction to Heat Transfer", John Wiley & Sons Inc., Fifth edition ,2006.

[8] Sombat Tamna, Warakom Nerdnoi, Chinaruk Thianpong and Pongjet Promvonge “Numerical Heat Transfer Study in a Square Channel with Zigzag-Angled Baffles”, The Second TSME International Conference on Mechanical Engineering,19-21 October, 2011, Krabi.

[9] Ahn S.W., “The effects of roughness types on friction factors and heat transfer in roughened rectangular duct”, Int. Commun. Heat and Mass Transfer, Vol.28 (7), pp. 933–942, 2001.

[10] Jenn-Jiang Hwang, Tong-Miin Liou, “ Heat transfer in a rectangular channel with perforated turbulence promoters using holographic interferometry measurement”, International Journal of Heat and Mass Transfer ,Vol.38 (17), pp.3197–3207,1995.

[11] Murata A., Mochizuki S., "Comparison between laminar and turbulent heat transfer in a stationary square duct with transverse or angled rib turbulators", Int. Commun. Heat and Mass Transfer ,Vol.44 ,pp. 1127–1141,2001.
[12] Rajendra Karwa, " Experimental studies of augmented heat transfer and friction in asymmetrically heated rectangular ducts with ribs on the heated wall in transverse, inclined, v-continuous and v-discrete pattern",International Communications in Heat and Mass Transfer ,Vol.30 (2) pp.241–250, 2003.
[13] S.S. Mousavi, K. Hooman, " Heat and fluid flow in entrance region of a channel with staggered baffles",Energy Convers. Manage.,Vol. 47,pp. 2011-2019 ,2006.
[14] B.K. Maheshwari, "Experimental study of flow structure in enhanced rectangular channels with transverse rib-roughness on one of the broad walls", Ph.D. Thesis, Jai Narain Vyas University, Jodhpur, India, 2007.
[15] R.P. Saini _, Jitendra Verma, "Heat transfer and friction factor correlations for a duct having dimple-shape artificial roughness for solar air heaters" ,Energy,Vol. 33, pp.1277– 1287, 2008.
[16] Promvonge P., Thianpong C., "Thermal performance assessment of turbulent channel flow over different shape ribs", Int. Commun. Heat Mass Transfer, Vol.35. pp. 1327–1334, 2008.

List of Symbols

A_s :Surface area of the duct, (m^2)
A_c :Cross sectional area of the duct, (m^2)
Cp :Specific heat , (J/kg. K)
d_h :Hydraulic diameter of ducts,(m)
h :Convection heat transfer coefficient, (W/m^2. K)
K :Thermal conductivity, (W/m. K)
Lt :Test section length , (m)
$\dot{m}$:Mass flow rate , (kg/s)
Nu :Nusselt number for duct = $h.\ d_h / K_a$
P :Wetted perimeter, (m)
Q :Rate of heat transfer,(Watt)
Re :Reynolds number duct = $u.\ d_h / \nu_a$
T :Temperature (^oC)
TEF :Thermal enhancement factor
u :Average flow velocity, (m/s)

Subscripts

a : air
m : mean
i : inlet
o : outlet
s : smooth

Greek Letters

μ : Dynamic viscosity, (Pa. s).
ν : Kinematic viscosity (m^2/s).
ρ : Density, (kg/m^3).