Journal of Engineering and Sustainable Development

www.jeasd.org
Vol. 21, No.03, May 2017
ISSN 2520-0917

EFFECT OF BAFFLES ORIENTATION ON HEAT TRANSFER ENHANCEMENT IN DIMPLED SQUARE DUCT

Abeer Hashim Falih

Assist.lect., Mechanical Engineering Department, Al-Mustansiriyah University, Baghdad, Iraq.

Abstract: The article presents an experimentally investigation on turbulent flow and heat transfer behaviors in a dimple square duct fitted with inline angled baffles having a constant length for two pitch ratio , (PR= P_B/a=0.83and 1.66), and three attack angle, α=45°,60° and 90°. The inline angled baffles are mounted on the lower channel walls to create longitudinal vortex flows throughout the test duct . 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 3900 to 18750. The experimental results are compared between the duct with inclined baffles and the dimple smooth duct. The duct fitted baffle gives higher heat transfer rate and friction factor than the smooth duct. The experimental results reveal that the maximum heat transfer is at PR= 0.83. For the given conditions, the maximum thermal performance enhancement factor (TEF) of the inline -angled baffle with 30° is found to be about 2.4 at PR= 0.83 and Re= 3900.

***Keywords**: Square duct , Heat Transfer Enhancement ,Pressure drop, Inclined baffles*.

تأثير اتجاة العوائق على تحسين انتقال الحرارة داخل مجرى مربع ذي نتوءات

الخلاصة: : تقدم هذه المقالة دراسة عملية لتدفق وانتقال الحرارة لجريان اضطرابي داخل مجرى مربع ذي نتوءات مزود بعوائق مائلة ثابتة الطول وبنسبتين لل (PR =0.83and 1.66) ومائلة بثلاثة زوايا(α =45°,60° and 90°) . هذه العوائق مثبتة على السطح السفلي للمجرى لخلق دوامة طولية على طول مجرى الاختبار.مع تعرض مجرى الاختبار الى فيض حراري ثابت .اجريت التجارب من خلال تغيير تدفق الهواء الذي يتمثل بعدد رينولدز الذي يتراوح من 3900 الى 18750. تم مقارنة النتائج العملية بين المجرى المزود بالعوائق المائلة والمجرى الخالي من هذه العوائق حيث وجد ان المجرى المزود بالعوائق يمتلك اعلى معدل لانتقال حرارة وخسائر احتكاك مقارنة بالمجرى الخالي من تلك العوائق .كم بينت النتائج العملية ان اعلى معدل لانتقال الحرارة وجد عند PR=0.83وان اعلى اداء حراري وجد عند استخدام العائق المائل بزاوية 30 وكانت قيمته تتراوح حوالي 4. 2 عند PR=0.83 و . Re=3900

1. Introduction

The reduction of overall heat exchanger dimensions and increasing efficiency are common development techniques for heat transfer system enhancement. For such a long time, high thermal loads and decreased dimensions of baffles or ribs lead to using them in many thermal systems. The degree of cooling or heating levels used in the design of heat exchangers can be controlled via baffles usage; therefore the flow field and the distribution of the local heat transfer coefficient are completely

*Corresponding Author falihabeer@yahoo.com

changed. A recirculation zone, and the second recirculation zone behind the baffle and a reattachment at the channel wall, which are ahead of a single transverse baffle can be created the mainstream separation. Furthermore, the rise in the heat transfer rate towards the upstream baffle region with respect to the downstream one is caused by the inclination angle of the baffle with respect to the axial direction. Even though the increasing of heat transfer rate through the baffles arrangement, and the increasing of pressure drop across the channel are affected by decreasing of flow area. Thus, the baffle spacing, angle of attack and height are among the most important parameters used in the design of channel heat exchangers.

Inclined solid baffles were investigated by Dutta and Hossain [1], experimentally studied of local heat transfer characteristics and the frictional head loss in a rectangular channel fitted with two inclined solid and perforated baffles of the same overall size. The upstream baffle was attached to the top heated surface, while the orientation, position, and the shape of the other baffle was varied to identify the optimum configuration for enhanced heat transfer. The results showed that the local Nusselt number distribution was strongly dependent on the orientation, position, and geometry of the second baffle plate.

Supattarachai, et al.[2] carried out experimentally investigat of heat transfer enhancement and pressure drop in a square channel heat exchanger fitted with 45° and 90° inclined ribs. Air is the working fluid and Reynolds number ranging from 4000 to 26,000 .The rib to channel height ratio (*e/H*) of 0.1 and the rib pitch to channel height ratio, *PR*=1, 2 and 3 are introduced in this work. The result shows that the inclined rib with PR=1 gives higher heat transfer rate and friction factor than the one with PR=2, 3 and the smooth channel respectively and the rib with 45° provides the higher value of heat transfer and pressure drop than 90° for all rib pitch ratio. Lee et al. [3] studied experimentally the heat/mass transfer in rectangular channels with two different V-shaped ribs: continuous 60° V-shaped and multiple (staggered) 45° V-shaped ribs, and found that two pairs of counter-rotating vortices are generated in the channel and the effect aspect ratio of channel was more significant for the 60° V-shaped rib than for the multiple 45° V-shaped rib. From the above-mentioned analysis of the available literature, it is found that there have not been any investigations on enhancing the heat transfer in dimple square duct with baffles. Therefore, the main objective of this work is to study the heat transfer, pressure drop and thermal enhancement factor of dimple square duct with inclined baffles aligned on the lower surface of the test section using air as a working fluid.

2.2. Experimental Set up

A schematic diagram of experimental setup is shown in Fig. (1). It consists of an entrance section, test section, exit section, and a centrifugal blower with a control valve. The square duct has an internal size of 60*60 mm, which consists of an entrance section, a test section and an exit section of length 600, 600 and 600 mm, respectively. In the entrance section, three baffles with equally spaced are provided for mixing the delivered air, However ,the entrance length is (10 d_h) [4] long entry region was provided so that flow would be fully developed as it enters the baffles section.

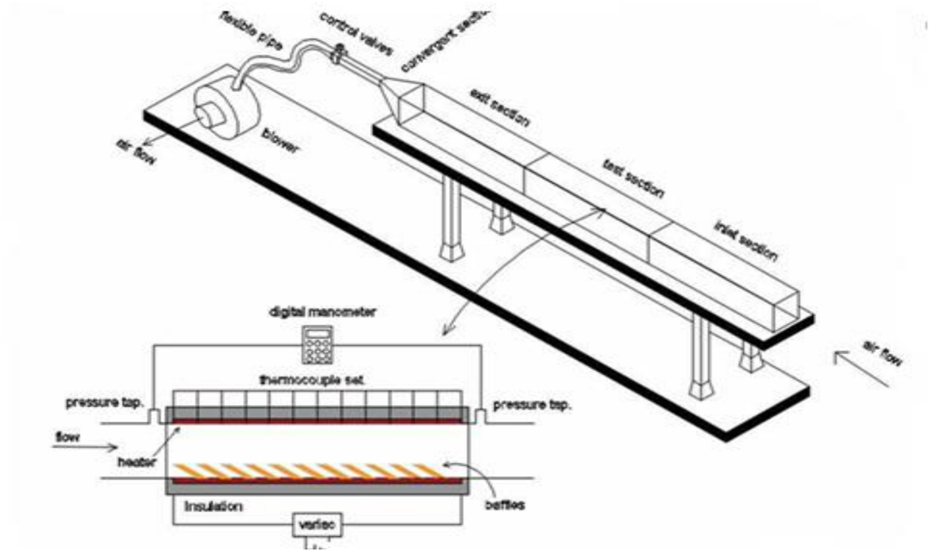
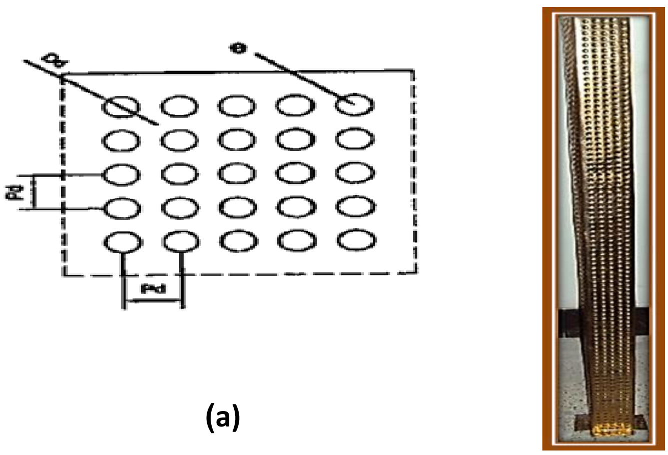


Fig.(1) : Schematic diagram of apparatus.

The test section is included from the dimple square duct and baffles inserts into the duct. The square duct made from copper (CuZn37), 0.5 mm in thickness. A schematic diagram of geometry of the duct wall is given in Fig. (2, a) and the duct sample will be tested as shown in Fig.(2,b).

Baffles insert are manufactured from copper with dimension ; height (w) is 20 mm, width (b) of the baffles is about 2 mm less than the width (a) of the duct .The distance between two successive baffles (pitch, p_B) is (5,10 cm). The baffles insert are set on the lower duct wall at 30º,60 ºand 90 º relative to air flow directions as shown in Fig.(3).



(a)

(b)

Fig. (2): (a) A schematic diagram of geometry of the duct wall. (b) The tested duct .

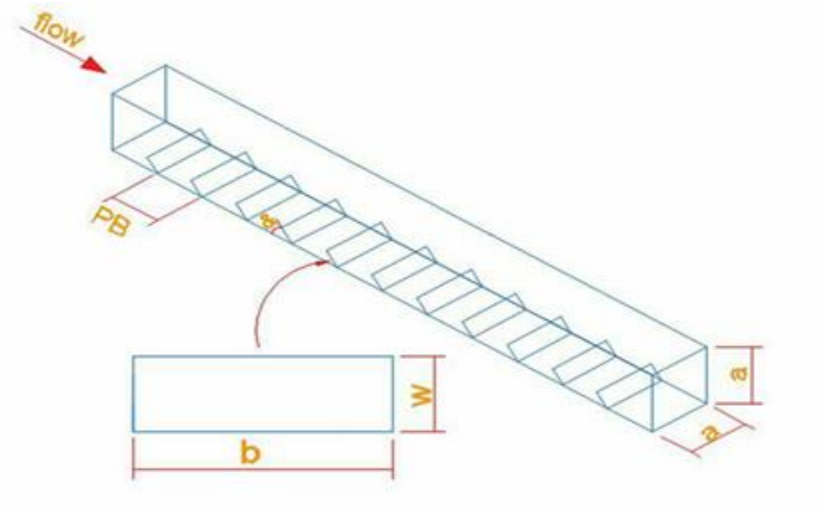


Fig.(3) : A schematic diagram of square duct with inclined baffles.

The baffles pitch ratio (PR) is defined as the ratio of the distance between two adjacent baffles to the duct height (P_B/a) . All details of the dimple square duct with baffles are demonstrated in Table (1).

Table 1: Baffles and test duct details.

Dimpled duct [5]	
Dimpled pitch length, Pd	11 mm
Dimpled diameter, Dd	5.5 mm
Dimpled depth, 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	50 mm, 100 mm
Pitch Ratio,PR	0.83, 1.66
Baffle thickness to height ratio ,δ/w	0.025
Baffle to duct height ratio ,w/a	0.333
Number of baffles	5 , 11

All sides of the test section is heated by supplying uniform heat flux (1000W/m^2) by means of an electrical heater and is insulated with 50 mm-thick glass wool to minimize the heat losses.

To measure the temperature of the duct surface twelve thermocouples are fixed on the sides of the test section surface and another two thermocouples are used to measure the inlet and out let air temperature. The pressure drop is measured with the help of digital manometer under isothermal condition of flow without switching on the heater. Digital vane-type anemometer is used to measure the air velocities.

In the experiments, the cold air with ambient condition was passed through the test section by means of a 0.8 kW high speed blower, and its inlet flow rate was measured and controlled by using vane-type anemometers and a control valve, respectively. After switching on the electric heater the sufficient time is given to reach to the steady state condition. Reynolds numbers for the air flowing through the test section are controlled in the range of 3900 to 18750 for turbulent flow region. Experiments are conducted for smooth duct, and subsequently by inserting the baffles at different flow rates. In each run, data are taken for airflow rate, walls temperatures of the test section and inlet and outlet air temperature**.**

3. Data Reduction Equation

The heat transfer coefficient between the air and the test section is calculated from [6]:

$$h = Q/\ [A_s\,(T_s - T_f)] \tag{1}$$

Where the heat transfer rate ,Q, to the air is given by [6]

$$Q = \dot{m}\ cp\ (T_o - T_i) \tag{2}$$

In equ. (1) , $A_s = (4\ a.\ L_t)$ is the heat transfer surface area ,neglecting the surface area of the baffles.

$$T_s = \sum T_s / 12$$

T_f is the bulk mean air temperature $(T_i + T_o)/\,2$

The heat transfer coefficient has been used to calculate the Nusselt number [6] :

$$Nu = h\ d_h/k \tag{3}$$

Where
d_h : is the hydraulic diameter of square duct

$$d_h \quad = 4A_c\,/\,p = 4a^2/4a = a \tag{4}$$

And the Reynolds number is calculated from [6],

$$Re = \rho u d_h/\mu \tag{5}$$

The friction factor is calculated over the test section length as:

$$f = 2\Delta p d_h/\rho\ L_t\, u^2 \tag{6}$$

Where
Δp is the pressure drop over the test section, while u is the average inlet velocity.

The thermal enhancement factor is the ratio of the heat transfer coefficient of a roughed surface to that of a smooth surface at equal pumping power and can be expressed as [2]:

$$TEF = (Nu/Nu_s)/(f/f_s)^{1/3} \tag{7}$$

Where Nu_s and f_s are the Nusselt number and friction factor of a smooth duct.

4. Results

In order to prove the reliability of the experimental data, the present smooth duct results are compared with the results obtained 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 [6]:

$$Nu_s=0.023Re^{0.8}Pr^{0.4} \tag{8}$$

The friction factor for flat smooth duct given by Blasius equation is [6]:

$$f_s=0.316Re^{-0.25} \tag{9}$$

For the Reynolds number range of this work, the experimental data of the present smooth duct are found in good agreement with the results from the previous correlations as shown in Figs. 4 and 5, respectively.

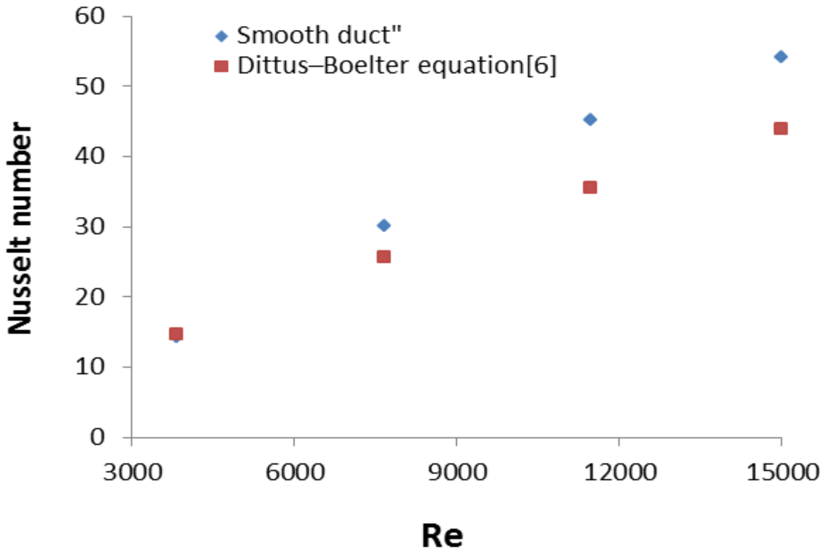


Fig. (4):Nusselt number as function of Reynolds number for square duct.

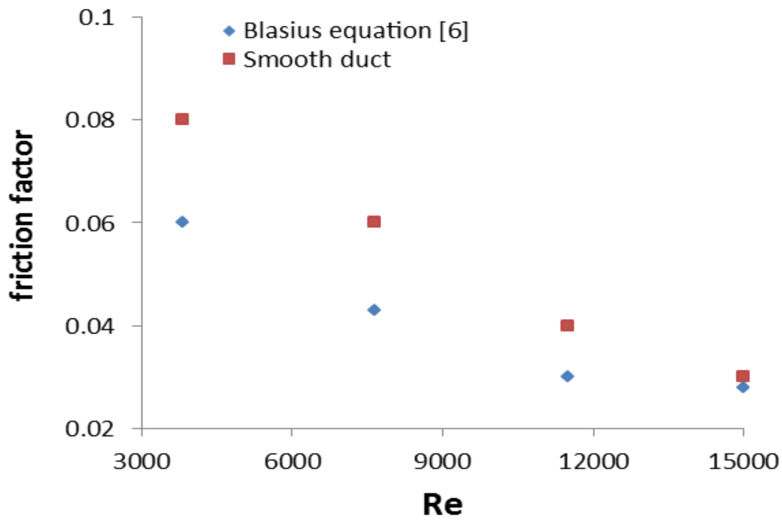


Fig. (5): Friction factor as function of Reynolds number for square duct.

The Nusselt numbers for all cases are presented under turbulent flow conditions are shown in Fig.(6) . It's clear from this figure, the Nusselt number increases with the rise of Reynolds number and the heat transfer rate of dimple square duct fitted with baffles yield considerable with a similar trend in comparison with the smooth duct . This is because the dimple duct roughness combined with baffles interrupt the development of the boundary layer of the fluid flow and increase the turbulence degree of flow. Fig. (7) show the variation of Nusselt number ratio with Reynolds number. It's clear from this figure, the Nusselt number ratio tends to decrease slightly with Reynolds number rise from 3900-18750 for all of studied cases. The maximum Nusselt number ratio values are found to be about 3.7 and 2.9 time over the dimple smooth duct for using the 30° inclined baffles with PR = 0.83 and 1.66 respectively. The 60° inclined baffles give the maximum Nusselt number ratio are found to be about 3.4 and 1.99 time at PR = 0.83and 1.66 respectively and for 90° inclined baffles the maximum Nusselt number ratio is about 2.45 and 1.6 time over the dimple smooth duct.

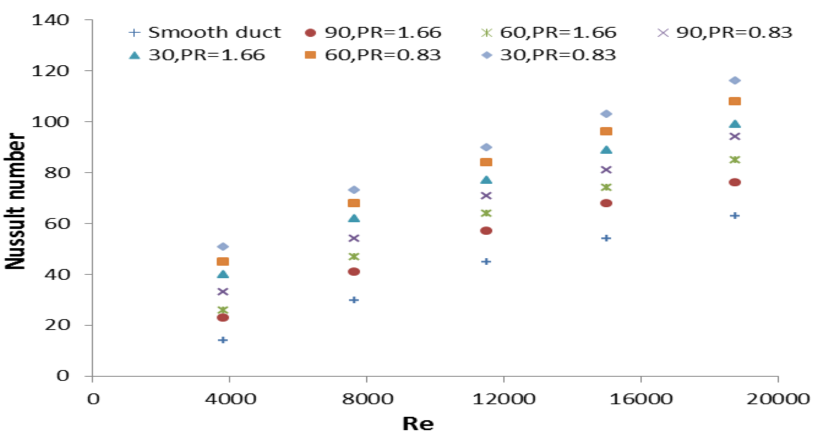


Fig.(6) :Variation of Nusselt number with Reynolds number.

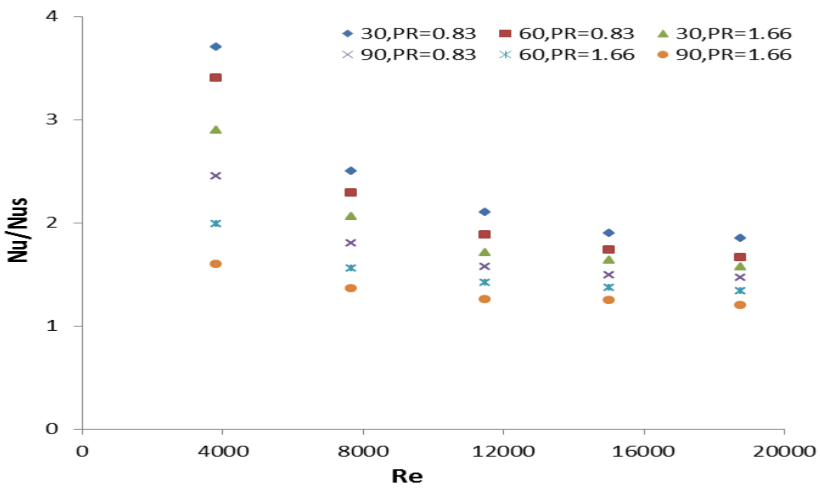


Fig. (7): Variation of Nusselt number ratio, Nu/Nu_s with Reynolds number.

Fig. (8) shows the effect of using the inclined baffles turbulators on the pressure drop across the tested duct as shown in terms of friction factor. It's clear from this figure the use of baffles turbulators leads to a substantial increase in friction factor over the dimple smooth duct. Fig. (9) show the variation of friction factor ratio with Reynolds number. It's clear from this figure, The maximum friction factor ratio is found to be about 7.5and 5.5 time over the dimple smooth duct for using the 30° inclined baffles with PR = 0.83and 1.66 respectively. The 60° inclined baffles have given the maximum friction factor ratio values are found to be about 9.5 and 6.5 time at PR = 0.83and 1.66 respectively and for 90° inclined baffles the maximum friction factor ratio values are found to be about 11 and 8.5 time over the dimple smooth duct.

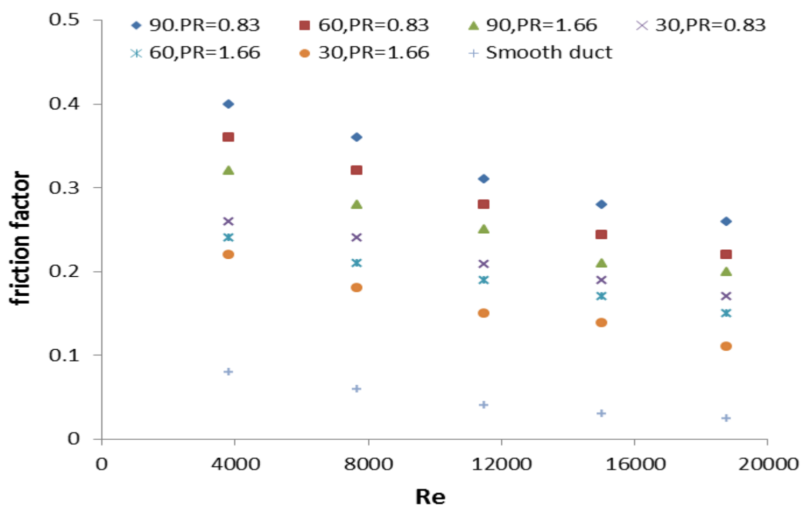


Fig.(8) :Variation of Friction factor with Reynolds number.

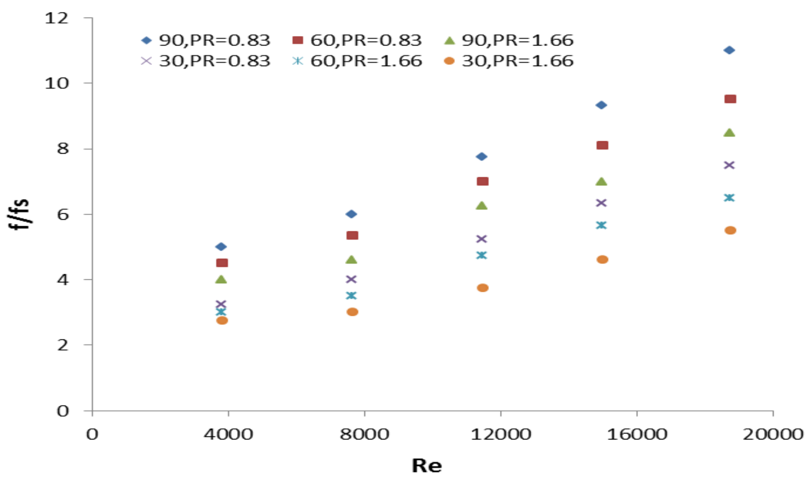


Fig. (9): Variation of Friction factor ratio, f/f_s with Reynolds number.

The thermal enhancement factor for the duct with various inclined baffles fitted is compared at the same pumping power in Fig. (10).

Its be noted from this figure, the performance factor tends to decrease with the increasing Reynolds number and the maximum thermal enhancement factor is found about of 2.4 when using the 30° inclined baffles with PR = 0.83.

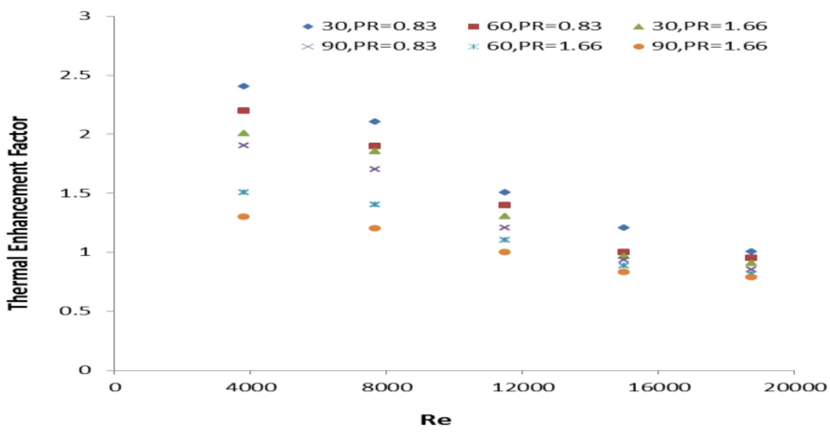


Fig.(10): Variation of thermal enhancement factor with Reynolds number.

5. Conclusions

The following conclusions are drawn from the results of this investigation.

1. The Nusselt number increases with the rise of Reynolds number and the heat transfer rate of dimple square duct fitted with baffles yield considerable with a similar trend in comparison with the smooth duct.
2. The Nusselt number ratio tends to decrease slightly with Reynolds number rise from 3900-18750 for all of studied cases.
3. The maximum Nusselt number ratio values are found to be about 3.7 and 2.9 for using the 30° inclined baffles, for 60° inclined baffles is about 3.4 and 1.99 and

for 90° inclined baffles is about 2.45 and 1.6 time over the dimple smooth duct with PR = 0.83 and 1.66 respectively.

4. The use of baffles turbulators leads to a substantial increase in friction factor over the dimple smooth duct.
5. The maximum friction factor ratio is found to be about 7.5and 5.5 for using the 30° inclined baffles, for 60° inclined baffles is about 9.5 and 6.5 and for 90° inclined baffles is about 11 and 8.5 time over the dimple smooth duct with PR = 0.83 and 1.66 respectively.
6. The performance factor tends to decrease with the increasing Reynolds number.
7. The maximum thermal enhancement factor is found about of 2.4 when using the 30° inclined baffles with PR = 0.83.

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 = $\rho u d_h / \mu$
T	Temperature (°C)
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).

6. References

1. Dutta, P., and Hossain, A. (2005)."Internal cooling augmentation in rectangular channel using two inclined baffles".International Journal of Heat and Fluid Flow, 26(2), pp. 223-232.
2. Supattarachai, Suriya, Chinaruk and Pongjet Promvonge, (2011). "Thermal Behavior in a Square Channel with Angled Ribs". The Second TSME International Conference on Mechanical Engineering 19-21 October, Krabi.
3. Lee D.H., Rhee D.H., Kim K.M., Cho H.H., Moon H.K. (2009)."Detailed measurement of heat/mass transfer with continuous and multiple V-shaped ribs in rectangular channel". Energy Vol.34. PP. 1770–1778.
4. Adrian Bejan and Allan D. Kraus,(2003). Handbook of "Heat Transfer",Canada.
5. Mohamed.H, Fouad A. and Abeer H.(2012). "Compound heat transfer enhancement in dimpled and sinusoidal metal solar wall ducts fitted with wired inserts", Journal of engineering,Vol. 18,No. 5,PP. 591-610.
6. Incropera, F., and Dewitt, P. D. (2006)."Introduction to Heat Transfer". Fifth edition, John Wiley & Sons Inc.
7. Sombat Tamna, Warakom Nerdnoi, Chinaruk Thianpong and Pongjet Promvonge .(2011) ."Numerical Heat Transfer Study in a Square Channel with Zigzag-Angled Baffles". The Second TSME International Conference on Mechanical Engineering,19-21 October, Krabi.
8. Ahn S.W. (2001)."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.
9. Murata A., Mochizuki S. (2001). “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.
10. Mousavi S.S., Hooman K.(2006). " Heat and fluid flow in entrance region of a channel with staggered baffles". Energy Convers. Manage.,Vol. 47,PP. 2011-2019.
11. Maheshwari B.K. 2007."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.
12. Saini R.P., Jitendra Verma.(2008)."Heat transfer and friction factor correlations for a duct having dimple-shape artificial roughness for solar air heaters". Energy, Vol. 33, PP.1277– 1287.