Experimental Investigation on the Thermal-Hydraulic Performance of Channel with Gradient Metal Foam Baffles

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Metal foam is a novel material recently utilized in baffles as an alternative to solid baffles for reducing flow resistance. However, the metal foam baffles are accompanied by low heat transfer efficiency. To overcome this issue, a new design of copper foam baffles has been suggested in this research, called baffles having a gradient pore density of the copper foam. The pore density either increases or decreases towards the wall. So, the experimental tests were carried out in a square channel and heated uniformly at the bottom wall of the test section. Its walls are mounted copper foam baffles at a fixed porosity of 95%. Baffles were alternately fixed upon the walls' bottom and top in staggered mode. The results were determined for various kinds of copper foam (10 and 20) pores per inch (PPI), and the gradient pore density was either with the order decreasing (DPPI) 10/20 PPI or increasing (IPPI) 20/10 PPI with a window cut ratio of 25% and a constant heat flux of 4.4 kW/m^2. The Reynolds number was changed from 3.8x10^4 to 5.4x10^4. The data for conventional copper solid baffles were used to compare the influence of foam metal type. The obtained results revealed an enhancement in thermohydraulic performance for baffles with a gradient pore density of the order decreasing DPPI (10/20 PPI) higher than all the models of copper foam baffles.

Keywords: *Gradientmetal foam, Heat exchanger, Heat transfer enhancement, Gradientmetal foam baffles*

1. INTRODUCTION

The employments of channels with baffles are one of the most prevalent passive heat transfer augmentation solutions in single-phase internal flows. This passive heat transfer enhancement technique has been used for various industrial applications, such as shell-and-tube heat exchangers, labyrinth seals for turbo-machines, internal cooling systems of gas turbine blades, thermal regenerators, and electronic cooling devices, Hwang [1]. Despite this wide range of applications, the existence of such baffles causes the flow to separate, re-attach, and therefore generate the zones of opposite flow and the high shearing rates that affect the thermal-hydraulic performance. However, some researchers investigated the effect of the appropriate geometric parameters of the baffles, such as baffle height (or window cut ratio), baffle spacing, and the relative arrangement of baffles that gives the best heat transfer performance for a given pumping power or flow rate. For example, Habib et al.[2]studied the effect of a window-cutting ratio of solid baffles on the flow and heat transfer characteristics at turbulent flow.

The experiments were carried out in a rectangular channel with baffles arranged as staggered with heating at a constant wall heat flux along the top and the bottom using air as the working fluid. The tests were performed at a range of Reynolds numbers from 8x10^3 up to 1.6 x10^3 with different values of window cut ratio of 0.3, 0.5, and 0.7. The researchers showed that as the window cut ratio decreased and the Reynolds number increased, the local and average heat transfer and pressure loss increased. However, the increase in the pressure loss was found to be much higher than the increase in the heat transfer coefficient. Li and Kottke[3] investigated the impact of the distance between the baffles in the models of shell-and-tube heat exchangers on the heat transfer and pressure drop experimentally. The experiments were conducted at a range of Reynolds numbers with different values of the distance between the baffles. Results demonstrated that for a fixed value of the Reynolds number, the heat exchange coefficient and the pressure drop increased as the distance between the baffles increased. Analysis of the turbulent flow in a rectangular channel with baffled plates was studied numerically and experimentally by Demartini et al.[4]. It was found that the greatest fluctuations in the pressure and velocity fields occurred near the deflectors. Saim et al. [5] investigated numerically the fluid flow and the heat transfer performances of a rectangular channel equipped with solid plate baffles, which are arranged on the top and bottom channel walls in a periodically staggered method. It was found that the vortex shedding produced by the baffle on the upper wall can additi–

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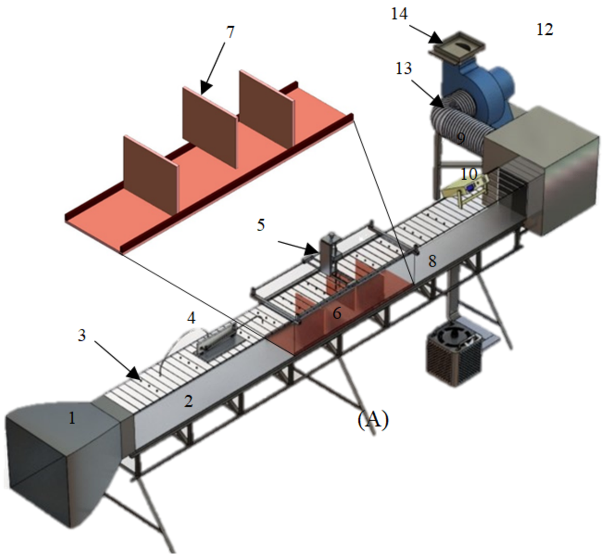
onally improve the heat transfer together with the baffle surfaces. As the spacing between the baffles decreases, the friction coefficient increases.

However, utilizing solid baffles improves the heat transfer, but on the other side, it causes an increased pressure drop and a higher local thermal stress at the baffle's root, leading to worry. Another passive method for enhancing the heat transfer characteristics in thermal systems represented porous media/metal foam, which caused a decrease in the pressure drop, Stanojevic et al. [6] and Soltan-Tehrani et al. [7]. As a result, using po–rous baffles to reduce pressure loss has been studied by many researchers, such as Ko and Anand[8], where an experimental study was achieved using aluminum foam baffles as heat sinks. The results depicted that the staggered porous metal foam blocks with a pore density of 20 PPI can decrease the frictional resistance in the channel via(3) quarters compared with the solid baffles having similar size and structure. Karwa and Maheshwari [9, 10]conducted an experimental study of heat transfer and friction in a duct with fully perforated (open area ratio of 46.8%) or half-perforated baffles (open area ratio of 26%) at a pitch-to-baffle height ratio of (7.2–28.8) for a turbulent flow. This study indicated that the half-perforated baffles have better thermo-hydraulically performance than the fully perforated baffles at the same pitch. Mahadevan et al. [11] studied the heat transfer and the pressure drop in a channel with a low aspect ratio experimentally and placed staggered carbon foams serving as porous baffles. Besides, the channel blockage ratio (0.09 and 0.37) influence on the heat transfer coefficient and pressure drop was included. It was observed that though the heat transfer was high in a channel with solid baffles and in a channel with porous baffles, the porous baffles benefited in terms of pressure drop. In another study conducted by Hamadouche et al.[12], three aluminum foam blocks staggered on the top and bottom walls of a rectangular channel were made. The authors studied the impact of channel blockage ratio (0.6, 0.8, and 1) on the thermo-hydraulic performance under a turbulent flow experimentally. The results evinced that the metallic foam block insertion resulted in better thermal performance with pressure drops less than that created with the utilization of the solid aluminum blocks. From the other side, it was shown that the arrangement with aluminum foam occupying the whole channel empties double as much heat as the arrangement with aluminum fins and much lesser pressure drop. Chen et al. [13] studied the thermal-hydraulic performance of the metal foam baffle heat exchanger systems numerically. More–over, it was compared with heat exchangers with and without traditional baffles. It was found that the metal foam baffles can reduce the pressure drop efficiently. That is to mention, implementing the metal-foam baffle can efficiently encourage the complete performance of heat exchangers.

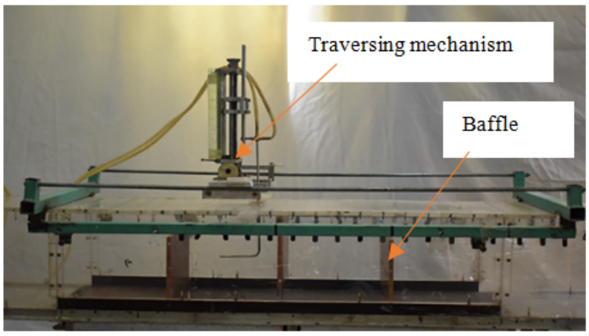
The above analysis shows that the metal foam baf–fles are one of the possible solutions for improving heat transfer and reducing flow resistance. Thus, the inade–quacy of less heat transfer efficacy can be overwhelmed via utilized non-uniform metal foam configurations of baffles(gradient metal foam), wherein the metal form morphological parameters are varied in parallel or normal direction to the flow, which can alter the thermal-hydraulic performance. So, Wang et al. [14] analyzed the heat transfer numerically in pipes either incompletely or filled with a gradient porous material (GPM), which linearly changed in the radial or axial direction. Results showed that in certain circumstances, the gradient porous material could possess better heat transfer and lesser pressure drop with regard to the uniform porous-filled ones. As well as, Xu et al. [15]numerically analyzed the forced convection heat transfer in circular tubes incompletely filled with graded-metal foams; the range of porosity was (0.80-0.98), and the range of PPI was (5-40). Nusselt number and the friction factor are vigorously influenced via the foam features. Furthermore, for a similar arrangement, Xu and Gong [16]elucidated that the gradient of PPI more highly influences the friction factor than via the gradient of porosity, where the Nusselt number reduced with the rising porosity and the gradients of PPI. Kotresha and Gnanasekaran [17] analyzed graded foams with PPI in a range of (20–40) and a porosity of 0.93; a decrease in the pressure drop without notable changes in the thermal performance was obtained. The above lite–rature survey indicates that, recently, numerous inves–tigators have concentrated on the hydro-thermal inf–luences of metal foam baffles that consist of a single-layer porous medium. Also, limited studies have been performed in pipes partially filled or filled with a gradient porous material. It is found from the preceding research that the gradient porous material could possess better heat transfer and lesser pressure drop than the uniform porous-filled ones. Nevertheless, the authors found that the experimental data on the graded pore density of metal foam employed in the baffles at the channel to enhance the heat transfer is unavailable. Therefore, the present work aims to investigate experi–mentally for the two proposed models the optimum arrangement of the two layers of metal foam for the baffles with the same porosity but with different pore densities that gave the highest system hydrothermal performance. One has an order of decreasing pore density toward the wall, and the other has a charge of increasing pore density toward the wall. The experi–mental measurements were performed in a square channel subjected to a constant heat flux for the Reynolds number range (3.8x104-5.4x104) with wall-mounted metal foam baffles having a gradient pore density.

2. EXPERIMENTAL APPARATUS

The experimental set-up, as displayed in Figure (1), was divided into(4) main constituents:(1) measurement system,(2) test section, (3) test specimens, and (4) air supply system. The flow system was suction-operated and horizontally orientated. The air is drawn into the channel's entrance, flows throughout the test section downstream, and reaches the settling chamber linked to a blower by a flexible duct to minimize the vibration. The flow rate was controlled using a gate valve. The Perspex square channel has an internal size of 250mm X 250 mm X 3250 mm, which consists of an entrance section, a test section, and an exit section of lengths 1250, 1000, and 1000 mm, respectively. A 5 mm-thick heated copperplate with relative roughness (ε=7x 10^{-7})was used as abroad bottom wall of the test section.



1	contraction cone	2	Entrance section
3	Tapping static	4	Inclined manometer
5	Traverse mechanism	6	Test section
7	Baffle	8	Variac
9	Exit section	10	Data logger
11	Flexible duct	12	Settling chamber
13	Centrifugal fan	14	Gate valve



(B)

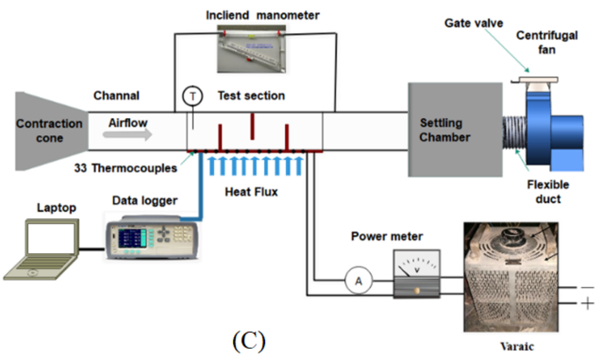


Figure 1 (A) Layout of test rig (B) Photo of the Test section (C) Schematic diagram of the experimental

Moreover, the absorber plate was heated from the bottom by providing a (4.4 kW/m^2) homogenous heat flux via an electrical heater and was insulated with 50 mm-thick glass wool topped with 10 mm-thick plywood. The electrical power given to the heating element was governed via a variac and measured via a power meter Hameg HM8115-2 with an accuracy of (±0.4%) for the voltage and current. Three baffles were fixed alternately at the test section and staggered into the bottom and top walls. Five types of copper baffles were used in this study, as illustrated in Figure (2). The copper baffles structures are as follows; model 1 is solid; models 2 and 3 consisted of uniform copper foam with pore density 10 and 20 PPI, respectively. Whereas models 4 and 5 are non-uniform copper with the gradient pore density, which is either with the order of decreasing or increasing pore density towards the wall, i.e.(10/20 PPI)(DPPI)or (20/10PPI)IPPI, respectively. Table 1 gives the physical properties of copper baffles [18]. These baffles have dimensions such as a depth of (250 mm), baffles spacing of (250 mm), baffle thickness of (10 mm), and window cut ratio of (25%). The bottom of the baffles was glued with the test section**,** as well as the interface between the two layers for the gradient by a thin layer of thermally conductive epoxy glue was produced by Shenzhen Halnziye Electronics Company Ltd. while the other walls of baffles were glued with the side walls and top surface firmly using thermal epoxy. To measure the wall temperature of the copper plate, the test section was provided with (33) thermocouples Type K spread lengthways and crossways of the copper plate, where they were cautiously introduced in drilled holes beneath the copper plate surface.

Table 1. The physical properties of copper baffles

Sample	Pore density (PPI)	K (m2) E-07	Ø (%)	Metal
Model 1	-	-	-	Solid copper
Model 2	10	2.61	95	Uniform copper foam
Model 3	20	1.56	95	Uniform copper foam
Model 4	10/20	2.51	95	Non-uniform copper foam with gradient pore density having order decreasing PPI (DPPI)
Model 5	20/10	1.89	95	Non-uniform copper foam with gradient pore density having order-increasing PPI (IPPI)

In order to ensure the appropriate contact between the copper plate and thermocouples, the thermal compound paste was used. A traverse thermocouple was utilized to measure the air temperature spreading locally at the test with K-type thermocouples. All thermo–couples were calibrated with (± 0.01 K) accuracy. The temperature was recorded by a data logger (Appellant AT4532x). The pressure drop across the test section for the various baffles arrangements was measured with MK 4&5 inclined manometer kind linked to pressure taps mounted at (80mm) upstream and downstream of the test section with (1 Pa) accuracy. A static pitot tube of (1 mm) diameter was utilized to measure the velocity magnitude. The probe has a traversing mechanism and MK 4 and 5 inclined manometer type. The differential pressure and temperatures at steady state were measured for every inlet velocity.

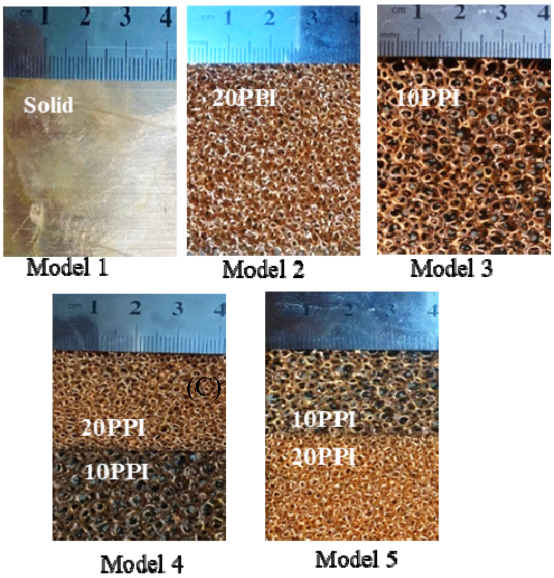


Figure 2: Models of copper baffles

3. DATA REDUCTION

The Reynolds number founded upon hydraulic diameter Dh=4Ac / 2(W+H) was determined as follows Hwang [1]:

$$\mathrm{Re}_{Dh} = \frac{\bar{U}Dh}{v} \tag{1}$$

where $\bar{U}$ is the average velocity of air at the test sec– tion. The friction factor in a baffled channel flow can be obtained by measuring the pressure drop ΔP crossways in the test section and the average air velocity. Then, the friction factor was computed by the equation below Ko and Anand [8]:

$$f = \frac{\left|(\Delta P / l)\, Dh\right|}{\rho \bar{U}^2 / 2} \tag{2}$$

where l is the test section length.

This equation computed the air bulk temperature (Tb) Habib et al. [2]:

$$T_b = \int_0^H |u| T dy \Big/ \int_0^H |u| dy \tag{3}$$

After that, the air bulk temperature at any random point within the test section was computed utilizing the mass flow rate and the net heat flow at the same point. The net heat input to the heated element Q_i of the baffled test section is defined as Habib et al. [2]:

$$\overline{Nu} \tag{4}$$

where, T_{bo} and $T_{bin}$are the air bulk temperature at the outlet and inlet of the heated element at the baffled test section, respectively, and $\dot{m}$ is the rate of the mass flow of air passing throughout the test section, which is defined as:

$$\dot{m} = \rho A c \bar{U} \tag{5}$$

This equation computed the heat flux for element q''_i by Habib et al. [2]:

$$q''_i = \frac{Q_i}{W x_i} \tag{6}$$

where W and x_i denote the width and length of the heated element at the test section's bottom wall, respectively. Then, the local heat transfer coefficient h_{li} through the test section was assessed by this equation Habib et al. [2]:

$$h_{li} = \frac{q''_i}{(T_w - T_b)_i} \tag{7}$$

T_{bi} is the bulk temperature of the air, and T_{wi} is the wall temperature of the heating element. The average convective heat transfer coefficient was calculated as follows by Habib et al. [2]:

$$\bar{h} = \frac{1}{l}\int_0^l h dx \tag{8}$$

Then, the average Nusselt number was calculated from the equation below by Habib et al. [2]:

$$\overline{Nu} = \frac{\bar{h} D_h}{kf} \tag{9}$$

In all calculations, the values of the thermo-physical properties of air were obtained at the bulk mean tempe– rature.

The performance evaluation criteria (η) was used to compare the enhanced forms of an original configu– ration (i.e., a channel with baffles) and the plain configuration (a channel without baffles), which was determined as the ratio of the average Nusselt number to the friction factor raised to the power of 1/3 as a described by the equation below, Ko and Anand [8].

$$\eta = \left(\frac{\left| \overline{Nu} / (f)^{1/3} \right|_B}{\left| \overline{Nu} / (f)^{1/3} \right|_s} \right) \tag{10}$$

The heat losses through the test section to the ambient air were measured with an empty test section. The heat losses were less than 12%. Experimental data were acquired at atmospheric pressure over the follo– wing range of parameters: Reynolds number ($3.8\text{x}10^4$-$5.4\text{x}10^4$) and heat flux q" = 4.4 kW/m2. The steady-state values of the heating wall and the air temperatures in the duct at different positions were utilized to determine the heat transfer coefficient values, the parameter of thermal performance, the friction factor, and the average Nusselt number. The propagated uncertainty analysis was conducted using the technique clarified in Coleman and Steel [19], and the values were 7.5 % and 21.1% for friction factor and Nusselt number, correspondingly

4. RESULTS AND DISCUSSION

4.1 Validation

In order to validate the experimental procedure and start the present study with the baffles, the results of friction factors and average Nusselt numbers for the smooth channel without baffles in the current work for fully developed flow were compared to those provided in the literature. So, the comparison between the calculated friction factor and the Moody correlation [20] for fully developed flow is represented in Figure (3). This comparison showed that the experimental results gave the same trend as the Moody correlation, where the friction factor decreased slightly with increasing Reynolds number. Also, the experimental average Nusselt number results were compared with the results obtained from the correlations of Dittus-Boelter [21], as evinced in Figure (4). It was found that the maximum difference between the average Nusselt number obtained in the present work and the correlation of Dittus-Boelter is about 22%.

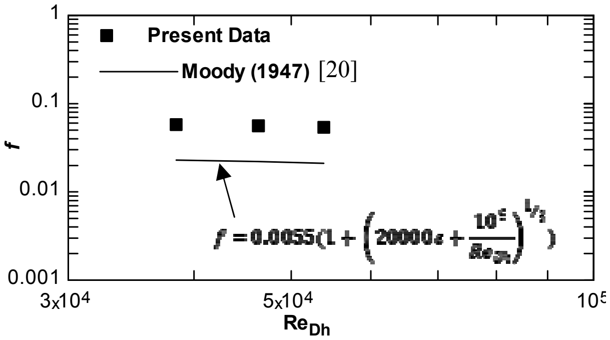


Figure 3. Validation of the friction factor

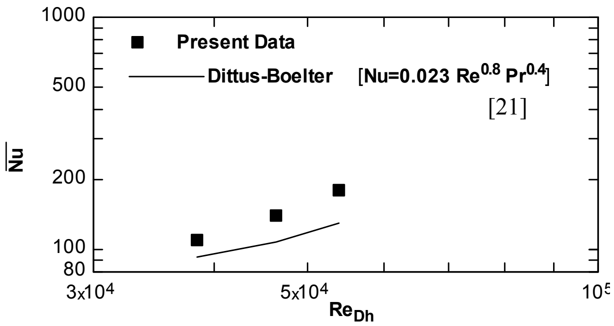


Figure 4. Validation of the average Nusselt number

4.2 Friction factor

The selection requirement of including structural materials for the baffles at the channel is getting low pumping power with high contact area. So, the friction factor of the channel with and without baffles (solid and copper foam) as a function of the Reynolds number can be demonstrated in Figure (5). It is clear from this figure that the friction factor values for all baffles models are higher than that of smooth surfaces, and the friction factor variation is similar behavior, which decreases slightly as the Reynolds number is increased. According to this figure, the friction factor for solid baffles is higher among all specimens; this is because that using the baffles can motivate the occurrence of recirculation that appears backward from the solid baffles, which cannot be observed on the copper foam baffles, as reported byKo and Anand [8]. The friction factor for the solid baffles and copper foam baffles (10, 20, DPPI (10/20), and IPPI (20/10)) PPI is about (460), and (20,38, 29, and 30) times, respectively, above the smooth surface. The copper foam baffles have a lower friction factor than the solid baffles.

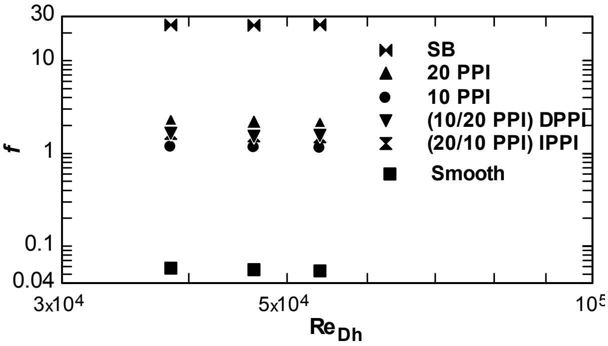


Figure 5. The friction factor variation against the Reynolds number for five models of baffles

However, the friction factor for a uniform copper foam baffles model(20PPI) is higher than those for the models (10, DPPI (10/20), and IPPI (20/10)) PPI. This behavior can be explained as follows: according to Darcy's equation, at a constant rate of flow, when the permeability for metal foam is small, the pressure drop crosswise the baffle is the largest. Therefore, the baffles (metal foam with 20PPI), which have low permeability, as seen in Table (1), caused the highest pressure drop.

4.3 Average Nusselt number

The influence of the Reynolds number on the average Nusselt number for all baffles (solid and copper foam) is elucidated in Figure (6). Also, the average Nusselt no. for the smooth surface was presented in this figure for comparison. In general, it is visible that all specimens have the same trend of average Nusselt number with Reynolds number, which increased with increasing it. As well, the average Nusselt number of copper baffles solid/foam (10, 20, DPPI (10/20), and IPPI (20/10)) PPI is higher than that for the smooth surface with 5/ (1.5,2,1.8, and 1.7) times, respectively. The solid baffles model has a higher heat transfer than other models. This results from the recirculating region of flow, which seems to be behind the baffle, where the heat is removed locally by the shedding of vortices only.

However, it is clear from the figure for baffles mo–dels with uniform copper foam of (10PPI and 20PPI) that the average Nusselt number is affected by the increased pore density from 10 PPI to 20 PPI, where it increases as the pores density is increased. This incre–asing caused by the increased surface area of copper foam for 20 PPI than that of 10 PPI.

The effect of the gradient pore density of baffles models (DPPI and IPPI) on the average Nusselt number is due to its specific surface areas and the permeability of the copper foam models. For the fixed porosity (95 %), models of baffles with higher pore density (20PPI) have higher specific surface areas than models with lower pore density (10PPI), which provide improved heat transfer (positive factor), but at the same time, the permeability is low (negative factor), as portrayed in Table 1. So, the trade-off between the positive and negative factors is required to find the best order gradient of pore density for two layers of copper foam. It is obvious from Figure (6) that the average Nusselt number of the baffles models with a gradient pore density of decreasing order (10/20 PPI)DPPI has **a** higher average Nusselt number than the gradient pore density with increasing order (20/10PPI)IPPI with (19.3%). This can be explained by Ghaneifar et al. [22]in the case of gradient pore density with an order of decreasing (10/20 PPI) DPPI. As air approaches copper foam with high resistance (20PPI), it is more prone to avoid the foam and move away from it. Hence, the air that has passed through the foam will be smaller. Then, when the air moves away from 20PPI to the 10 PPI foam region, it has higher permeability and, consequ–ently, higher velocity. As a result, the fluid has a better chance of finding alternative paths toward the heated wall, which leads to the higher transfer of heat from the heated wall, and, subsequently, the heat transfer proce–dure enhances. Moreover, the level of heat transfer aug–mentation of baffles with gradient pore density having order (10/20 PPI) DPPI is slightly lower than that of the uniform copper foam baffles with pore density (20 PPI) by 2%.

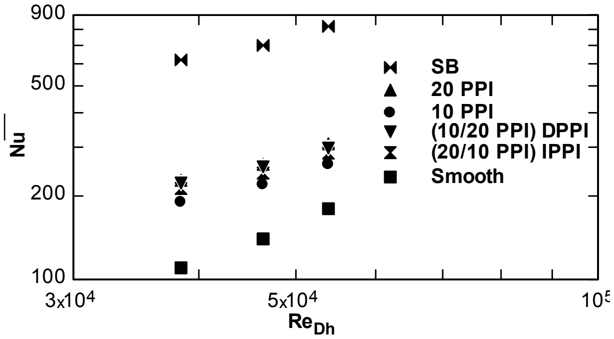


Figure 6. The average Nusselt number variation against the Reynolds number for five models of baffles

4.4 Thermo-hydraulic performance

The improvement of heat transfer in the baffled channel is related to a considerable increment in the friction factor. Therefore, the baffled channel's performance evaluation with regard to a smooth duct was utilized. Thermo-hydraulic performance evaluation criteria (η) were used for trading the advantages of utilizing copper foam baffles in the heat transfer improvement and their side influences in forcing further pumping power to the system as an alternative to solid baffles. So, variation of the (η) number within the Reynolds number (3.8×10^4-5.4×10^4) range can be demonstrated in Figure (7) for all baffles models (solid, uniform copper foam (10 and 20)PPI, and non-uniform copper foam (10/20 PPI)DPPI and (20/10 PPI)IPPI).In general, it is clear that the thermo-hydraulic performance factor (η) was less than one over the range of the parameters studied in the present work, which agrees with Ko and Anand [8]. Also, it decreased with increasing Reynolds number due to the increased friction coefficient that overcomes the improvement of the average Nusselt number. As expected, the solid baffles' performance factor is 79.1%-56.4%, while the baffles with uniform copper foam (10 and 20) PPI and non-uniform copper foam ((10/20 PPI) DPPIand (20/10 PPI) IPPI) give the thermal transfer rate of (66.5%-46.3%, 64.5%-48.1%, 68.9%-50.5%, and 65%-47.9%), respectively that have lower thermal per–formance than solid baffles with 14%,17%,11.4%, and 16%. Asa result, it can be concluded that the new design of foam baffles with DPPI gave the highest performance among all copper foam baffles models despite having a lower heat transfer rate than copper foam baffles of 20 PPI with around 2%. This is because of a moderate heat transfer improvement and a dramatically lower pressure drop followed by copper foam baffles.

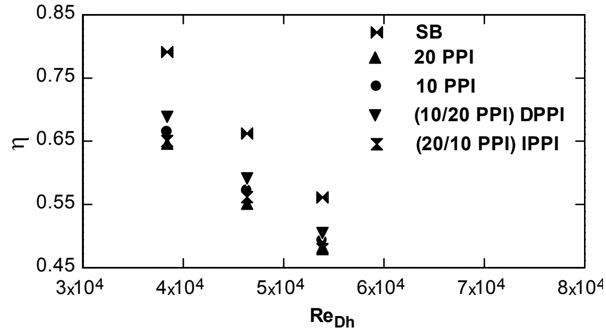


Figure 7.Thermo-hydraulic performance variation against Reynolds number for five models of baffles

4.5 Comparison of present experimental work with previous work

The present experimental work was carried out for three baffles that were mounted in a channel-staggered manner with a window cut ratio of 25%, a pitch ratio equal to one, and a constant heat flux of (4.4) kW/m^2 at the test section' slower wall. The experimental data for ($\overline{Nu_B}$ / $\overline{Nu_s}$) and the Thermo-hydraulic performance of the present work for baffles (solid/ copper foam (10 and 20)PPI) was compared to the correlation presented by Ko and Anand [8], as revealed in Figures (8) and (9), respectively. The comparison for the ($\overline{Nu_B}$ / $\overline{Nu_s}$) and thermo-hydraulic performance depicts that the corre–lation predicted very well for specimens of copper baffles (solid/foam (10 and 20 PPI) with a mean abso–lute error (MAE) value of 10.1%, 8%, and 15.4%, respectively for the ($\overline{Nu_B}$ / $\overline{Nu_s}$) and withaMAE value of 17.1%, 16.5%, and 25.7% for thermo-hydraulic performance.

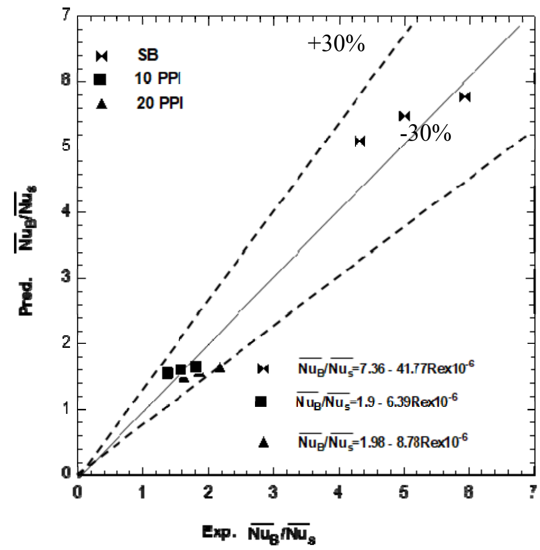


Figure 8. Comparison of present experimental data with a correlation of Ko and Anand [8] **for (** $\overline{Nu_B}$ / $\overline{Nu_s}$ **)**

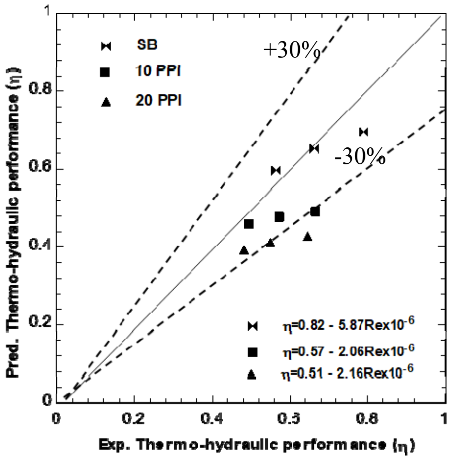


Figure 9. Comparison of present experimental data with a correlation of Ko and Anand [8] **for Thermo-hydraulic performance.**

5. CONCLUSIONS

Experiments were carried out for a turbulent flow in a heated square channel with solid baffles or metal foam baffles at a fixed porosity of 95% copper to study the heat transfer enhancement. The baffles were mounted on the top, and bottom walls in a staggered manner with baffles spacing (250 mm) and a window cut the ratio of 25%. The results are presented for baffles with uniform copper foam (10 and 20 PPI) and non-uniform copper foam (DPPI and IPPI) for a Reynolds number range of (3.8×10^4-5.4×10^4). Also, the results for solid baffles are introduced for comparison. The following concluded remarks can be drawn as follows:

1) Relative to the solid-copper baffles, the copper foam baffles have a highly lower friction factor, where the friction factor for the solid baffles and copper foam baffles (10, 20, 10/20 DPPI, and 20/10 IPPI) is about (460), and (20,38, 29, and 30) times, respectively above the smooth surface.
2) The baffles of copper foam with low permeability (i.e., 20 PPI) provided a higher average Nusselt number with associated higher friction factor than the other models of copper foam baffles (10, (10/20)DPPI and (20/10) IPPI) PPI.
3) As compared among the all copper foam baffles models, the highest thermo-hydraulic performance was provided by the copper foam baffles having gradient pore density with an order of decreasing (10/20 DPPI) with 69.8%-50.5%.In addition, they had the lowest difference value between them and the solid baffles by around 11.4%.

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NOMENCLATURE

Ac	Flow cross-section area, m^2
Cp	Specific heat at constant pressure, J/kg.K
Dh	Hydraulic diameter, m
f	Friction factor
H	Channel height, m
h	Local heat transfer coefficient, W/m^2.k
$\bar{h}$	Average heat transfer coefficient, W/m^2.k
kf	Thermal conductivity of the fluid, W/m.k
K	Permeability, m^2
l	Length of the test section, m
$\dot{m}$	The mass flow rate of air, kg/s
P	Pressure, Pa
Q	Heat convection, W
q''	Heat flux, kW/m^2
T	Temperature, K
u	Local velocity, m/s
$\bar{U}$	Average velocity of air, m/s
W	Channel width, m
x	Length of the heated element, m

Greek symbols

ε	Relative roughness
η	Thermo-hydraulic performance
ν	Kinematic viscosity, m^2/s
ρ	Density, kg/m^3
Ø	Porosity
Δ	Difference

Subscripts

B	Baffles
b	Bulk
i	Element
in	Inlet
l	local
o	Outlet
s	Smooth surface
w	Wall

Abbreviations

DPPI	Decreasing pore density
GPM	gradient porous material
IPPI	Increasing pore density
MAE	Mean absolute error
$\overline{Nu}$	Average Nusselt number
PPI	Pore per inch
Re_{Dh}	Reynolds number based on channel hydraulic diameter
SB	Solid copper baffle

ЕКСПЕРИМЕНТАЛНО ИСТРАЖИВАЊЕ ТОПЛОТНО-ХИДРАУЛИЧКИХ ПЕРФОРМАНСИ КАНАЛА СА ПРЕГРАДАМА ОД ГРАДИЈЕНТНЕ МЕТАЛНЕ ПЕНЕ

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Метална пена је нови материјал који се недавно користио у преградама као алтернатива чврстим преградама за смањење отпора протока. Међутим, преграде од металне пене су праћене ниском ефи–касношћу преноса топлоте. Да би се превазишао овај проблем, у овом истраживању је предложен нови дизајн преграда од бакарне пене, названих преграда са градијентом густине пора бакарне пене. Густина пора се или повећава или смањује према зиду. Дакле, експериментална испитивања су спро–ведена у квадратном каналу и равномерно загрејана на доњем зиду испитног дела. На његове зидове су постављене преграде од бакарне пене са фиксном порозношћу од 95%. Преграде су наизменично фиксиране на дно и на врх зидова у степенастом

режиму. Резултати су одређени за различите врсте бакарне пене (10 и 20) пора по инчу (ППИ), а густина пора градијента је била или са смањењем (ДППИ) 10/20 ППИ или повећањем (ИППИ) 20/10 ППИ са степен резања прозора од 25% и константан топлотни ток од 4,4 кВ/м2. Рејн–олдсов број је промењен са 3,8к104 на 5,4к104. За поређење утицаја типа пенастог метала коришћени су подаци за конвенционалне бакарне преграде. Добијени резултати су открили побољшање термо–хидра–уличких перформанси за преграде са градијентом густине пора реда опадајућег ДППИ (10/20 ППИ) већим од свих модела преграда од бакарне пене.