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Study of Baffles Arrangement Influence on the Natural Convection into a Heated Square Channel

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ABSTRACT

The present experimental paper is officiated to study the heat characteristics and performance of air flow through a square cross-sectional heated channel under natural heat convection conditions. The influence of baffles arrangement and perforated baffles on the rate of heat transfer through a channel are studied. Four cases of a heated channels are studied, one is a plain channel and other three channels with a baffle namely, three inline baffles, three staggered baffles and three staggered perforated baffles arrangements. The outer tested channels surfaces are electrically heated with a constant surface heat flux condition. All tested channels are fabricated by cold forming with constant dimensions using aluminum plate has thickness of 1.0 mm. The heat performance is assessed for all tested channels. The results of present paper are approved than the available data in the previous paper and a good convergence are noticed. Obtained experimental results appear that a three staggered perforated baffles arrangement is a best selection as it improves heat transfer rate in expression of Nusselt number, it higher about 22% to 27% than that the plain channel for same conditions.

1. INTRODUCTION

One of important passive approaches for heat transfer improvement is baffles used. The baffles come in different configurations and arrangements. Baffles are an integral parts of heat exchanger design, industrial process, some of household stoves, chemical reactors and solar heat collectors.

Demartini et al. [1] focused experimentally and numerically on the investigation of turbulent air flow into a rectangle cross sectional channel with two baffles. They used finite volume technique to solve of governing differential equations. They compared obtained results with experimental data. They noted the biggest variations in pressure and velocity obtains near of baffle-plates regions. Nasr et al. [2] investigated numerically and experimentally the turbulent flow into a rectangular cross sectional duct with inclined perforated baffles. They used two baffles, one (upstream baffle) is attached to the top wall of duct while the second is attached to the lower wall. They utilized Reynolds numbers ranged from 71,000 to 122,500. They observed that the surface coefficient of pressure recovery is powerfully affected by geometry and position of baffles in the rectangular duct. Benzenine et al. [3] investigated numerically the turbulent air flow into the channel using two various configurations of baffles like trapezoidal and rectangular. They obtained the velocity profiles for all configurations and for various sections like, upstream, downstream and between baffles, in addition, they obtained the friction factors for various sections and Reynolds numbers They concluded the trapezoidal baffle type gives higher velocity values compared with rectangular baffle type. Alwan [4] introduced an experimental and numerical investigating for natural convective into a cubic enclosure with pair of baffles at different inclinations angles (0°≤baffles angles≤150°). He utilized (k-ε) model in the numerical solution. He concluded the bigger values of average Nusselt number obtains when baffles angles are equal 0° while smallest values at 90°.

Rashidi et al. [5] analyzed numerically the thermal and hydrodynamic parameters of nano-fluid flow into a square sectional duct added with transverse twisted baffles. They used a finite volume technique to simulate forced heat convection and determined the best design of system. They study the influence of pitch intensity for values from 180° to 540° and nanoparticles volume fraction for values about 0 to 0.05 on the nano-fluid flow and forced heat convection. They concluded that the baffle with pitch of 360° gives maximum value of convection heat transfer coefficient while the baffle with pitch of 540° gives minimum pressure drop. Also, they noted that the heat entropy generation decreases with increasing the nanoparticles volume fraction. Javadi et al. [6] simulated numerically the effect of rib configurations for turbulent flow into a duct. They used different Reynolds numbers from 20,000 to 60,000 with six rib configurations namely, square, rectangular, isosceles and rectangular trapezoidal, equilateral and non-equilateral triangles. They used a shear stress transport (k-omega) turbulence model for the simulations. They concluded that the ribs with no inclined sides like square and rectangular configurations create more heat irreversibility. In addition, the equilateral triangle shape of ribs has the biggest frictional entropy generation for all ranges of the Reynolds number. Palaniappan et al. [7] achieved a numerical study of heat convection in ventilated square cavity with insulated baffles. The right and left surfaces of cavity are conserved at the high temperature while the top and bottom surfaces of cavity are adiabatic. They used finite difference method to solve the governing Navier Stokes equations. They studied the influence of baffles size, baffles location, Rayleigh number, Reynolds number on the fluid flow and isotherms. They noted that the increase of baffles size and its variable locations gives a major effect on the heat performance of opened cavity. The rate of heat transfer is decreased with baffles size increasing. Razavi et al. [8] performed a numerical simulation on the effect of oriented perforated baffles on thermal and hydrodynamics behavior into a rectangular channels. They used different baffles angles, baffle type, baffles ratios and Reynolds numbers. They noted that the 135°-perforated baffle type is the best selection as it optimizes rate of heat transfer in expression of Nusselt number and minimize the friction factor. Sadeq and Shehab [9] investigated experimentally of heat convective for air flow inside an annuli. Two types of inner finned tube are utilized namely, rectangle and trapezoidal extended surfaces. They observed Nusselt number values of trapezoidal tube type are largest compared with other types. Boonloi and Jedsadaratanachai [10] studied numerically the heat performance of fluid flow into a tube included different sizes of V-baffles using finite volume approach with a SIMPLE algorithm. They studied the modified type of V- baffle (combination of V-baffles and V-orifices) and placed on inner surface and core of tube. They used range of Reynolds numbers (100-2,000). They found the best heat enhancement factor in the tube with a modified V-baffles about 3.92. Menni et al. [11] investigated numerically the baffling approach to increase the heat performance of air flow into the channel using computational fluid dynamics (CFD). They utilized two types of arc baffles into the channels such as arc-upstream and arc-downstream baffles for wide range of Reynolds number from 12,000 to 32,000. They observed that the arc downstream baffle type gives best heat performance as heat exchange rate about 14.0% with other baffles types. Boonloi and Jedsadaratanachai [12] simulated numerically the heat performance of fluid flow into a rectangular duct with six different types of wavy baffles for Reynolds numbers from 100 to 2,000.

They used different baffle heights (h=0.05 to 0.3 of duct width). They found the best heat enhancement factor in the duct with a modified wavy baffles about 3.70 at height (h=0.3 of duct width). Other researchers investigated and analyzed the thermal performance and fluid properties of flow into a channels and enclosures (rectangular and square cross-sectionals) with different baffles configurations, heights and positions [13-18].

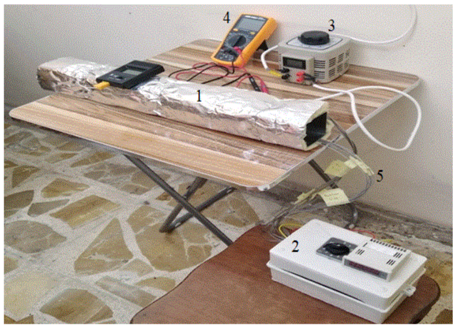
The present paper concentrates on the experimental study and analysis for the thermal characteristics and heat performance of air flow into a baffled square cross-sectional channel under natural convection and constant surface heat flux conditions. Besides, focuses on the effect the baffles arrangement (in-line and staggered) with and without circular perforations.

2. EXPERIMENTAL ANALYSIS

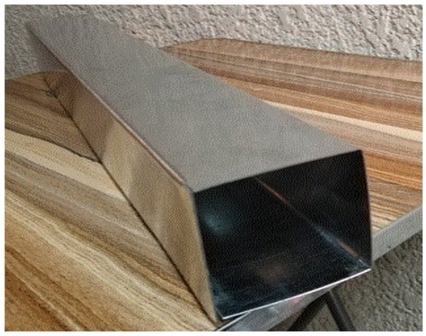
2.1 Experimental test rig

Figure 1 showed the test rig. It is manufactured to achieve the current paper tests. Test rig includes, the test section (square channel, heating element, thermocouples, electrical and thermal insulations), selector-switch with temperatures reader, multi-meter, voltage variac.

Four square cross-sectional channels namely, plain, in-line arrangement three baffles, staggered arrangement three baffles and staggered arrangement three perforated baffles shown in Figure 2 utilized in the tests are manufactured from aluminum sheet plate using a cold forming with fixed dimensions (channel length of 550 mm, side of 50 mm and thickness of 1.0 mm). The baffles are made from aluminum plate has thickness of (1.0 mm) and fixed in the channel with several arrangements. The sheet electrical heater of (1,000 watt) is used to heat the outer surfaces of channel under constant heat flux condition. The heating element is a best heat insulated by glass wool with aluminum foil layer of (50 mm) thickness. Sixteen calibrated thermocouples model K are used, four thermocouples four each side of square channel. They are fixed on the outer surfaces of the channel with pitch of (150 mm) from first thermocouple which away from channel inlet (50 mm) as shown in Figure 2a to register surface temperatures for axial distance of channel. Plus, two thermocouples are used to register the inner and outer bulk temperatures through channel.



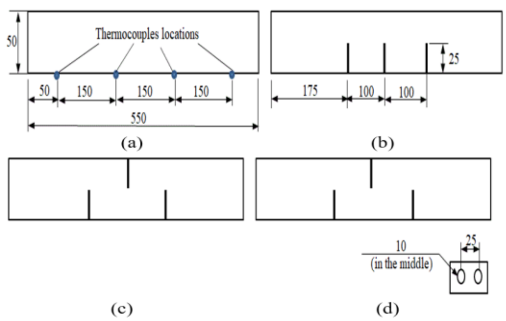
(a) Experimental test rig



(b) Aluminum square channel

1. Test section; 2. Selector switch; 3. Voltage variac; 4. Multi meter; 5. Thermocouple wires

Figure 1. Photos of test rig



(a) plain (b) in-line arrangement three baffles (c) staggered arrangement three baffles (d) staggered arrangement three perforated baffles

Figure 2. Square channels with various baffles arrangement and thermocouples locations

2.2 Experimental procedure

Many tests for four cases of square channels (plain, in-line arrangement three baffles, staggered arrangement three baffles and staggered arrangement three perforated baffles) are performed to investigate and analyze the effect of baffles arrangement and perforated baffles on the rate of heat transfer through an open ended square channels at an uniform heat flux. Four several heat fluxes such as (500, 1,000, 1,500 and 2,000 W/m^2) are used. The steps of tests as follows:

(1) Adjust the connections of a smooth channel. Then, the AC source is turn on.

(2) Adjust the voltage and current using voltage regulator to obtain the heat flux 500 W/m^2.

(3) Readings of thermocouples have been recorded after 30 min. when two readings of temperatures changed within 0.5°C. Then, surfaces temperatures, inlet and output air temperatures, current and voltage value are recorded.

(4) Again, the same steps for another heat fluxes, channels cases.

(5) Repeat the above steps for other channels cases.

2.3 Modeling of data

The input power (Q_i) to the heating element can be evaluated [19]:

$$Q_i = VI \tag{1}$$

The rate of convective heat transfer (Q_c) can be evaluated as [19]:

$$Q_c = Q_i - Q_l \tag{2}$$

where, I, V are intensity of current and voltage respectively; Q_l is a thermal loss due to heat conduction and heat radiation. They are very small so that neglected.

Local convective heat transfer coefficient (h_x) between the inside channel surfaces and bulk air temperatures (T_b) is [19]:

$$h_x = \frac{q}{T_{sx} - T_b} \tag{3}$$

where, q is the surface heat flux, and evaluated as [20]:

$$q = \frac{Q_i}{A_s} \tag{4}$$

T_b is bulk air temperatures into the channel and computed as average between the inlet (T_{bin}) and outlet (T_{bout}) bulk air temperatures [20]:

$$T_b = \frac{T_{bin} + T_{bout}}{2} \tag{5}$$

Define Nusselt number as, Local Nusselt Number (Nu_x) [21, 22]:

$$Nu_x = \frac{h_x\, D_h}{k} \tag{6}$$

And, average Nusselt number (Nu_a) [21, 22]:

$$Nu_a = \frac{h_a\, D_h}{k} \tag{7}$$

where,

$$h_a = \frac{q}{T_{sa} - T_b} \tag{8}$$

$$T_{sa} = \frac{\sum_{i=1}^{n} T_{sxi}}{n} \tag{9}$$

The average Rayleigh number (Ra_a) can be evaluated as equation [22]:

$$Ra_a = Gr_a\, Pr = \frac{g\, \beta\, Pr(T_{sa} - T_b)\, D_h^3}{\nu^2} \tag{10}$$

where, D_h is hydraulic diameter of the square channel; Pr is Prandtl number; Ra_a is average Rayleigh number; Gr_a is average Grashof number; k, g, ν and β are air thermal conductivity, acceleration of gravity, kinematic viscosity and the heat expansion volumetric coefficient respectively.

3. EXPERIMENTAL UNCERTAINTY ANALYSIS

The uncertainties and errors came from measurements, readings, calibrations and instruments types can be computed using the analytical method, the Nusselt number is a function of three independent variables as follows [23]:

$$Nu = f\ (V, I, \Delta T) \tag{11}$$

Then, the uncertainty in Nusselt number value can be computed as [23]:

$$e_{Nu} = \pm \left[\left(\frac{\partial Nu}{\partial V} e_V \right)^2 + \left(\frac{\partial Nu}{\partial I} e_I \right)^2 + \left(\frac{\partial Nu}{\partial \Delta T} e_{\Delta T} \right)^2 \right]^{1/2} \tag{12}$$

where, e_V=±0.05 is the error in voltage; e_I=±0.005 is the error in AC current; $e_{\Delta T}$=±0.01 is the error in temperature difference.

$$\Delta T = T_{Sx} - T_b \tag{13}$$

$$\left(\frac{\partial Nu}{\partial V} \right) = \frac{I\, D_h}{A_S\, \Delta T\, k} \tag{14}$$

$$\left(\frac{\partial Nu}{\partial I} \right) = \frac{V\, D_h}{A_S\, \Delta T\, k} \tag{15}$$

$$\left(\frac{\partial Nu}{\partial \Delta T} \right) = \frac{V\, I\, D_h}{A_S\, (\Delta T)^2\, k} \tag{16}$$

The relative error can be computed as:

$$Relative\ error = \frac{e_{Nu}}{Nu} \times 100 \tag{17}$$

Maximum relative error in Nu number is less than 0.5%.

4. RESULT AND DISCUSSION

To validate the current experimental data, a comparison between current data with Oosthuizen and Bishop correlation [24] for plain channel is achieved. Figure 3 illustrate that the current data a good agreement in behavior than Oosthuizen and Bishop correlation.

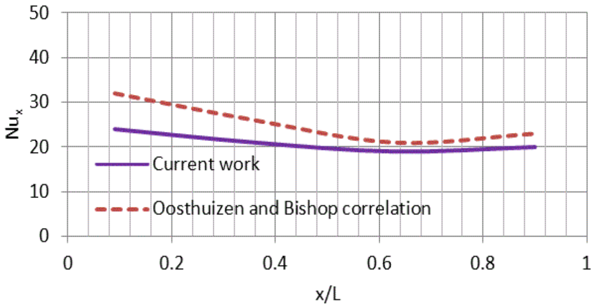


Figure 3. Data validation

Data obtained from the tests of natural convection for air flow into an open ended square channels are analyzed. Four several square channels with different baffles arrangements like plain, in-line arrangement three baffles, staggered arrangement three baffles and staggered arrangement three perforated baffles are studied and analyzed.

Behaviors of the temperature difference ($\Delta T = T_{Sx} - T_b$) along the channel axial distance (x/L) for plain channel at several heat fluxes are observed in Figure 4, the temperatures difference (ΔT) gradually increase as incrementing the axial-distance (x/L) from a channel inlet and arrive the maximum values near the channel and then a little decrements because the effect of bouncy as a consequence to flow cold air enter from the channel inlet to channel outlet under effect of natural convective. Increase of the boundary layer starts from channel inlet and developing along channel until fill the channel and consequently, increasing the surface temperature. Figure 5 show the influence of baffles arrangements on the temperature difference (ΔT) along the axial distance (x/L) comparing than plain channel at heat flux (q=2,000 W/m^2). It is clear that the smallest values of surface temperature at channel contains staggered baffles with perforated. This because the staggered baffles with perforated increase breaks the progressing of boundary layer. And so increase the turbulence of air flow.

The influence of baffles arrangements for channels with in-line arrangement three baffles, staggered arrangement three baffles and staggered arrangement three perforated baffles plus, plain channel on the local Nusselt number (Nu_x) with the axial distance (x/L) for several wall heat fluxes shown in Figures 6 and 7. They are clear that the local Nusselt number (Nu_x) decrements as the channel axial distance (x/L) increments to reach a smallest values near of channel outer and then, it a small increments due to the bouncy effect on air flow consequently, decreases the heat convection coefficient (h_x). Additionally, the three staggered perforated baffles arrangement is a best selection as it enhances the convective heat transfer rate as a local Nusselt number (Nu_x), it greater about (22% to 27%), (18% to 20%) and (8% to 13%) than those the plain, inline and staggered channels respectively for same conditions.

Figure 8 illustrated the effect of average Nusselt number (Nu_a) on Rayleigh number (Ra_a) for different channels cases. It's evidenced the values of average Nusselt number (Nu_a) increases than Rayleigh number (Ra_a) increasing for all cases and the bigger values at three staggered perforated baffles arrangement.

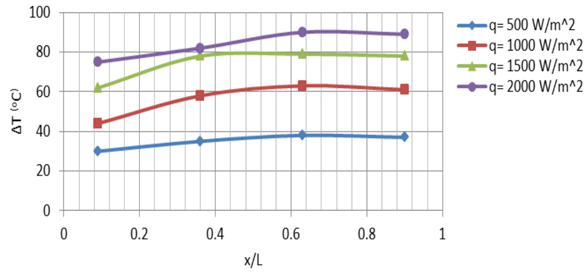


Figure 4. Effect of surface heat flux (q) on the temperature difference (ΔT) along the channel axial distance (x/L) for plain channel

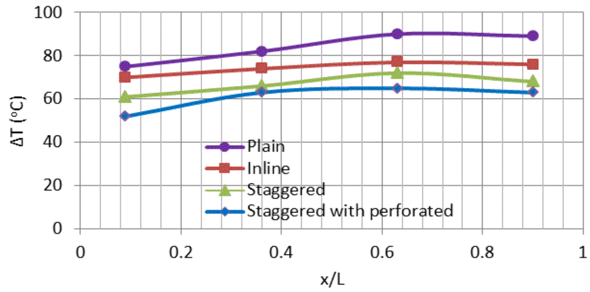


Figure 5. Effect of baffles arrangements on the temperatures (ΔT) along axial distance (x/L) for wall heat flux (q=2,000 W/m^2)

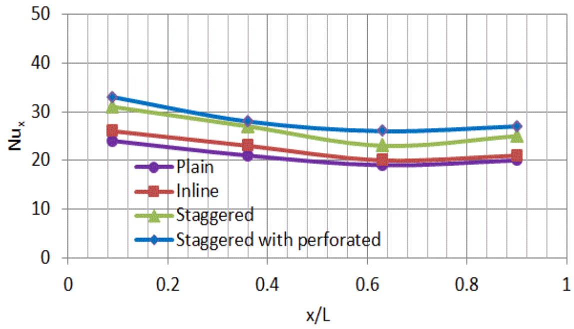


Figure 6. Effect of baffles arrangements on the Nusselt number (Nu_x) with axial distance (x/L) at heat flux (q=1,000 W/m^2)

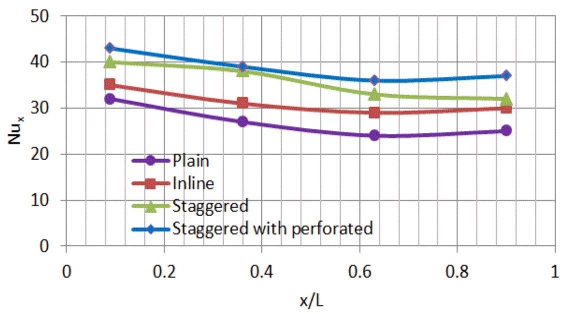


Figure 7. Effect of baffles arrangements on the Nusselt number (Nu_x) values with axial distance (x/L) at heat flux (q=2,000 W/m^2)

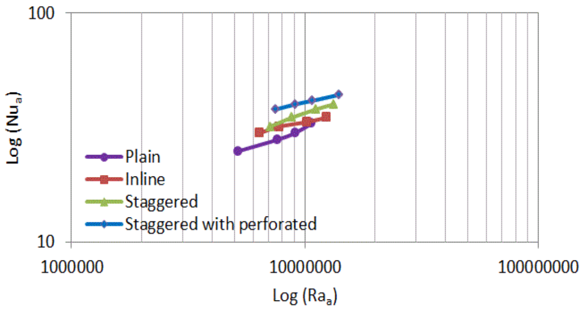


Figure 8. The behaviors of Nusselt numbers (Nu_a) versus Rayleigh numbers (Ra_a) for several baffles arrangements at heat flux (q=2,000 W/m^2)

5. CONCLUSIONS

In this experimental study, the natural convective for air flow into a heated square channel are studied and analyzed for four cases, one is a plain channel and other three channels with a baffle like, three inline baffles, three staggered baffles and three staggered perforated baffles arrangements. The main conclusions can be drawn as:

(1) Three staggered perforated baffles arrangement is best selection as it improves heat transfer rate in terms of Nusselt number, it higher about (22% to 27%), (18% to 20%) and (8% to 13%) than those the plain, inline and staggered channels respectively for same conditions.

(2) Values of Nusselt number (Nu_x) gradually decrements along channel axial-distance and then a lightly incrementing near of channel exit for all cases.

(3) Surface temperatures gradually increases along the axial distance of channel and then, there is a slightly decreasing near the channel exit for all channel cases.

(4) Values of the average Nusselt number (Nu_a) increase with increasing average Rayleigh number (Ra_a).

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NOMENCLATURE

A_c	cross sectional area of channel, m^2
D_h	square channel hydraulic diameter, m
g	gravity acceleration, m/s^2
Gr	grashof number
h	coefficient of convection heat transfer, W/m^2. K
I	current, A
k	thermal conductivity, W/m. K
Nu	Nusselet number
Pr	Prandtl number
q	surface heat flux, W/m^2
Q_c	natural heat convection, W
Q_i	heat input, W
Q_l	heat losses, W
Ra	Rayleigh number
T_b	bulk temperature, ℃
T_s	surface temperature, ℃
V	voltage, V

Greek symbols

β	heat expansion coefficient, K^{-1}
v	kinematic viscosity, m^2/s

Subscripts

a	average
in	inlet
out	outlet
x	local