Research Article

Younes Menni, Houari Ameer, Shao-Wen Yao*, Mohammed Amine Amraoui, Mustafa Inc, Giulio Lorenzini, and Hijaz Ahmad*

Computational fluid dynamic simulations and heat transfer characteristic comparisons of various arc-baffled channels

<https://doi.org/10.1515/phys-2021-0005>

received November 20, 2020; accepted January 09, 2021

Abstract: In this analysis, the baffling method is used to increase the efficiency of channel heat exchangers (CHEs). The present CFD (computational fluid dynamics)-based work aims to analyze the constant property, steady, turbulent, Newtonian, and incompressible fluid flow (air), in the presence of transverse-section, arc-shaped vortex generators (VGs) with two various geometrical models, i.e., arc towards the inlet section (called arc-upstream) and arc towards the outlet section (called arc-downstream), attached to the hot lower wall, in an in-line situation, through a horizontal duct. For the investigated range of Reynolds number (from 12,000 to 32,000), the order of the thermal exchange and pressure loss went from 1.599–3.309 to 3.667–21.103 times, respectively, over the values obtained with the unbaffled exchanger. The arc-downstream configuration proved its superiority in terms

of thermal exchange rate by about 14% than the other shape of baffle. Due to ability to produce strong flows, the arc-downstream baffle has given the highest outlet bulk temperature.

Keywords: Thermal exchange rate, pressure loss, baffling method, turbulent forced-convection, numerical solution

1 Introduction

The enhancement of the efficiency of heat exchangers (HEs) by the deflector insertion technique in the fluid flow domain has many engineering and industrial applications. By using numerical simulations, Pirouz et al. [1] used the Lattice Boltzmann Method (LBM) to explore the thermal behavior in a channel heat exchanger (CHE) equipped by upper and lower wall-inserted baffles. The influence of several geometrical variables of helical baffles on the overall performances of HEs was explored by Du et al. [2]. Based on the concepts of permeability and porosity, You et al. [3] introduced a numerical approach for shell-and-tube heat exchangers (STHXs). They used the distribution of turbulence kinetic energy, dissipation rate of energy, and heat source to estimate the effect of tubes on fluid. Eiamsa-ard and Promvong [4] explored the efficiency of a CHE having grooves on the lower wall. Also, Afrianto et al. [5] predicted the hydrothermal characteristics of liquid natural gas (LNG) in a HE. Additionally, for several Reynolds numbers (from 4,475 to 43,725), Ozceylan et al. [6] explored the influence of the presence of rings near the wall and their clearance on the overall performances of a CHE. Furthermore, the improvements in heat transfer rates that are yielded by the baffling technique in a pipe heat exchanger (PHE) have been reported by Nasiruddin and Siddiqui [7]. Moreover, via the CFD software FLUENT, Zhang et al. [8] provided details on a PHE with overlapped helical baffles. Also, under laminar flow conditions and with the help of CFD software,

* **Corresponding author: Shao-Wen Yao**, School of Mathematics and Information Science, Henan Polytechnic University, Jiaozuo 454000, China, e-mail: yaoshawen@hpu.edu.cn

* **Corresponding author: Hijaz Ahmad**, University of Engineering and Technology, Peshawar, Pakistan, e-mail: hijaz555@gmail.com

Younes Menni: Unit of Research on Materials and Renewable Energies, Department of Physics, Faculty of Sciences, Abou Bekr Belkaid University, P. O. Box 119, Tlemcen 13000, Algeria, e-mail: menniyounes.cfd@gmail.com

Houari Ameer: Department of Technology, University Centre of Naama, P. O. Box 66, Naama 45000, Algeria, e-mail: ameur@cuniv-naama.dz

Mohammed Amine Amraoui: Faculty of Technology, University Djillali Liabes Sidi-Bel-Abbès, BP 89 22000, Sidi-Bel-Abbès, Algérie, e-mail: amraoui_mohammedamine@yahoo.fr

Mustafa Inc: Department of Mathematics, Science Faculty, Firat University, Elazig, Turkey; Department of Medical Research, China Medical University Hospital, China Medical University, Taichung, Taiwan, e-mail: minc@firat.edu.tr

Giulio Lorenzini: Department of Engineering and Architecture, University of Parma, Parco Area delle Scienze, 181/A, Parma 43124, Italy, e-mail: Giulio.lorenzini@unipr.it

Figure 1: Baffled ducts under inspection (a) duct with in-line arc-upstream, (b) duct with in-line arc-downstream baffles. The dimensions (L , H , L_{in} , a , b , and c) are selected from [43].

Sripattanapipat and Promvonge [9] analyzed the hydrothermal characteristics through a 2D CHE. They used baffles with diamond shape and inserted in a staggered arrangement. Additionally, Santos and de Lemos [10] were interested in a channel having baffles constructed from porous and impermeable materials. Via numerical analysis and for a tube heat exchanger (THE), Xiao et al. [11] used different values of Prandtl number and helical tilt angles for helical baffles. Mohsenzadeh et al. [12] analyzed the forced convective thermal transfer through a horizontal CHE. They explored the influence of baffle clearance and Reynolds number. Under turbulent flow conditions, Valencia and Cid [13] used the CFD method to study the hydrothermal details in a CHE provided with square bars in the streamwise direction of flow. In another study, Promvonge et al. [14] determined the hydrothermal fields in a 3D CHE-inclined V-shaped discrete thin VGs (vortex generators).

By experiments, Zhang et al. [15] explored the hydrothermal fields of STHes equipped by helical baffles. Also, Wang et al. [16] were interested in the flow through a channel provided with pin fins. Additionally, Dutta and Hossain [17] were interested in a CHE having perforated baffles under various inclination angles. In another study, Ali et al. [18] studied the thermal transfer phenomenon from the outer surface of horizontal cylinders. Furthermore, Wang et al. [19] characterized the hydrothermal fields through a channel having periodic ribs

on one wall. Moreover, Rivir et al. [20] achieved measurements of the flow details in a CHE equipped with transverse ribs on the sidewalls. They explored the effect of three geometrical cases: single rib, staggered multiple ribs, and in-line multiple ribs.

Other interesting studies are available in the literature for various fluid flow situations and different numerical solutions, for example, see Kazem et al. [21], Rashidi et al. [22], Kumar et al. [23–25], Ghanbari et al. [26], Goufo et al. [27], Shafiq et al. [28], Ilhan et al. [29], Basha et al. [30], Baskonus [31], Guedda and Hammouch [32], Ahmad et al. [33–36], and Menni et al. [37–42]. Most of the studies carried out focused on the HE in order to improve its performance. Enhancing heat transfer is the goal of most researches in both numerical and experimental studies. So, the topic is very interesting, which led us to search a technique to increase the effectiveness of the exchanger. This work is a numerical analysis of a constant property, steady, turbulent, Newtonian, and incompressible fluid flow (air) inside a rectangular duct heat exchanger. Two types of transverse, solid-type, and arc-shaped obstacles are attached on the lower hot wall in in-line arrays, namely, an arc towards the inlet section (called arc-upstream) and an arc towards the outlet section (called arc-downstream). Both obstacle forms give a different structure to the flow, which enhances heat transfer in different amounts. The study compares the best performance which identifies the effective model.

2 Physical model under analysis

A numerical simulation was reported on the turbulent flow and forced-convection of an incompressible Newtonian fluid (air) with constant thermal physical properties and flowing inside a channel (Figure 1). Two transverse, solid, in-line obstacles having various geometrical configurations, i.e., arc towards the inlet section of the channel, called arc-upstream (Figure 1a), and arc towards the exit section of the channel, called arc-downstream (Figure 1b), were fitted into the duct and attached on the bottom wall to lengthen the trajectory of the fluid and increase the heat exchange surface.

Two new models of VGs are suggested in the present study, namely, arc-upstream and arc-downstream baffles. The values of height (H) and length (L) of the channel are 0.146 and 0.554 m, respectively. The first arc-shaped baffle is attached on the lower wall at $L_{in} = 0.218$ m from the entrance section of the channel, while the 2nd arc-shaped obstacle is inserted at $c = 0.142$ m from the first obstacle. The baffles height (a), thickness (b), and attack (θ) are 0.08 m, 0.01 m, and 45° , respectively. Air is used as a working fluid and Reynolds numbers are changed from 12,000 to 32,000.

3 Modelling and simulation

For the steady state with neglected radiation thermal exchange mode, the equations of mass, momentum, and energy are expressed as [7]

$$\nabla \cdot \vec{V} = 0, \quad (1)$$

$$\rho(\vec{V} \cdot \nabla \vec{V}) = -\nabla P + \mu \nabla^2 \vec{V}, \quad (2)$$

$$\rho C_p(\vec{V} \cdot \nabla T) = \lambda \nabla^2 T. \quad (3)$$

where, $\vec{V}$ is the speed vector (m s^{-1}); ρ is the fluid density (kg m^{-3}); P is the pressure (Pa); μ is the fluid dynamic viscosity ($\text{kg m}^{-1} \text{s}^{-1}$); C_p is the fluid specific heat ($\text{J kg}^{-1} \text{K}^{-1}$); λ is the fluid thermal conductivity ($\text{W m}^{-1} \text{K}^{-1}$); and T is the temperature (K). The standard k -epsilon [44] is also shown in this study in order to simulate the turbulent flow, as it is characterized by the kinetic energy (k) and dissipation rate (ϵ) equations as follows [38]:

$$\frac{\partial}{\partial x_j}(\rho k u_j) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k + \rho \epsilon, \quad (4)$$

$$\frac{\partial}{\partial x_j}(\rho \epsilon u_j) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\epsilon} \right) \frac{\partial \epsilon}{\partial x_j} \right] + C_{1\epsilon} \frac{\epsilon}{k} G_k - C_{2\epsilon} \rho \frac{\epsilon^2}{k}. \quad (5)$$

where, μ_t is the fluid turbulent viscosity ($\text{kg m}^{-1} \text{s}^{-1}$); and C_μ , $C_{1\epsilon}$, $C_{2\epsilon}$, σ_k , and σ_ϵ are the model constants. The present thermal and hydrodynamic limit conditions are expressed as [45]:

At $x = 0$

$$u = U_{in}, \quad (6a)$$

$$v = 0, \quad (6b)$$

$$T_{in} = 300 \text{ K}, \quad (6c)$$

$$k_{in} = 0.005 U_{in}^2, \quad (6d)$$

$$\epsilon_{in} = 0.1 k_{in}^2. \quad (6e)$$

At $y = \pm H/2$

$$u = v = 0, \quad (7a)$$

$$k = \epsilon = 0, \quad (7b)$$

$$T_w = 375 \text{ K}. \quad (7c)$$

At $x = L$

$$P = P_{atm}, \quad (8a)$$

$$\frac{\partial \phi}{\partial x} = 0. \quad (8b)$$

where, ϕ stands for the dependent variables u , v , k , T , and ϵ . The calculations are achieved with the finite volume technique [46], SIMPLE algorithm [46], and the Quick scheme [47]. The Nu_0 and f_0 values of the unbaffled channel were verified [38] by comparing them with the correlations of Dittus–Boelter [48] and Petukhov [49], respectively. The comparison demonstrated that there is a quantitative agreement between the CFD data and the experimental relationships results [48,49].

4 Results and discussion

4.1 Stream-function field

Figure 2 illustrates the streamlines for the different arc-baffle configurations, i.e., arc-upstream and arc-downstream. For the two studied geometrical models, the flow is uniform until reaching the 1st arc-baffle. Then, recirculating flows are formed at the baffled region, referenced here as ‘zone A.’ The size of these vortices is significant in the case of arc-upstream type baffles. The

Figure 2: Stream-function fields (ψ) for both cases under investigation: (a) arc-upstream baffled channel, (b) arc-downstream baffled channel, $Re = 12,000$ (ψ values in kg s^{-1}).

sharp edge of the baffle presents a point of detachment, the so referenced here as ‘zone B.’ The flow is then detached from the arc-baffle, resulting thus in a depression behind baffles. Furthermore, recirculation areas (zones ‘C’ and ‘D’) are formed behind baffles, where the widest recirculation zone is observed with the arc-downstream baffle. It is surrounded by iso-surfaces that take elliptical shapes.

4.2 Mean velocity field

Figure 3 reports the mean velocity fields for different configurations of arc-baffles, i.e., arc-upstream and arc-downstream. The velocities are weak next to the left side of the first arc-baffle, region ‘A,’ for both models of arc-obstacles studied. The velocity magnitudes are also low behind the second arc-VG. The velocity is important at the edge of the first and the second arc-shaped baffles, zone ‘E.’ In this same region, the fluid velocity for the second obstacle type (arc-downstream baffle) reaches up to 3.64 m/s , followed by that of the first type (arc-upstream baffle), 3.28 m/s . In the regions between both the first and the second arc-baffles, zone ‘C,’ the airflow velocity is significant in the arc-downstream model than that of the other model.

4.3 Axial velocity field

Figure 4 illustrates the impact of the variation of the arc-VG geometry on the axial velocity field. For both kinds of

Figure 3: Mean velocity fields (V) for both cases under investigation: (a) arc-upstream baffled channel, (b) arc-downstream baffled channel, $Re = 12,000$ (V values in m s^{-1}).

baffles, the streamlines are parallel in the unbaffled areas of the duct. However, the velocity magnitudes are almost negligible in the downstream areas of arc-baffles, zones ‘C’ and ‘D,’ which is caused by the presence of recirculating flows. An increase in the velocity is observed in the space between the tip of arc-VG to the upper wall of the exchanger, referenced here as ‘zone E.’ This is due to the presence of the arc-shaped VG and the abrupt modification in the flow direction.

The fluid flow is accelerated just after the first arc-VG until reaching $274\text{--}346\%$ of the inlet velocity, depending on the shape of VGs. It should be noted that the VG configuration has a significant influence in the zones ‘C,’ ‘D,’ and ‘E,’ which is mainly caused by the modification in the streamlines. The arc-upstream baffle increased the axial velocity by about 2.747 times over the inlet velocity U_{in} (Figure 4).

Figure 4: Axial velocity (u) for various arc-baffle configurations: (a) arc-upstream, (b) arc-downstream, $Re = 12,000$ (u values in m s^{-1}).

Figure 5: Fields of the transverse velocity (v) for various arc-baffle models: (a) arc-upstream, (b) arc-downstream, $Re = 12,000$ (v values in $m\ s^{-1}$).

4.4 Transverse velocity field

The distribution of y-transverse velocity is plotted in Figure 5a and b for the arc-upstream and arc-downstream baffles, respectively. Positive and negative velocity gradients are remarked at the tip of the 1st arc-baffle (zones 'B') and 2nd arc-baffle (zone 'F'), respectively.

4.5 Dynamic-pressure field

The dynamic-pressure fields for both arc-obstacle configurations are plotted in Figure 6. As illustrated in this figure, the values of the dynamic-pressure coefficient are weak near the VGs due to the presence of vortices.

However, the dynamic-pressure augments in the regions between the baffles tip and the upper wall of the exchanger, where the maximum values are located between the 1st and 2nd arc-baffles (zone 'E'), which is resulted from the high airflow velocities. In addition, the highest amount of P_d depends on the shape of arc-VG, where it is lower by about 18% for the arc-upstream baffle than that for the other case.

4.6 Dimensionless axial velocity profiles

The curves of the dimensionless axial velocity (U/U_{in}) just after the two baffles are plotted in Figure 7. As observed, the reattachment length for the arc-downstream baffle is greater than that for the arc-upstream-baffle, regardless of the value of Re .

Figure 6: Fields of the dynamic pressure (P_d) for various arc-baffle types: (a) arc-upstream, and (b) arc-downstream, $Re = 12,000$ (P_d values in Pa).

4.7 Thermal fields

The thermal fields illustrated in Figure 8 show that the baffled region is the most heated. The temperature drops in the areas between the tip of arc-obstacle and the surfaces of the duct, which is due to the high fluid velocity and interaction between the fluid particles in these regions.

A comparison of the outlet fluid temperature is provided in Figure 9, where the most considerable values of the temperature are reached with the arc-downstream baffle. Because of its ability to produce strong flows, the arc-downstream-shaped VG is more advantageous than the other model.

4.8 Heat transfer

The results of the ratio (Nu_x/Nu_0) are summarized in Figure 10 for both arc-shaped baffles. Both the arc-deflectors push the flow towards the upper part of the duct, which allows further absorption of the thermal energy from the heated surface. The lowest value of the Nu_x/Nu_0 is observed on the upstream side of the first arc-baffle, while the highest amount is remarked on the opposite side of the 2nd arc-baffle. This figure shows also that the Nu_x/Nu_0 is considerable in the downstream area of the 1st arc-VG. This augmentation is yielded from the efficient mixing by vortices, which corresponds to high rates of the local thermal exchange. For both shapes of baffles, the values of (Nu_x/Nu_0) are similar at the positions between (0 m) and (0.2 m). However, there is an important increase in the (Nu_x/Nu_0) in the case of

Figure 7: Influence of arc-baffle orientation on the length of recirculation cells vs Re. (a) at $x = 0.315 \text{ m}$ (downstream of the 1st VG) (b) at $x = 0.435 \text{ m}$ (downstream of the 2nd VG).

arc-downstream type baffle from the position (0.2 m) until the outlet of the duct.

Figure 11 presents the change of the average ratio (Nu/Nu_0), where a proportional increase is observed according to Re. The maximum Nu/Nu_0 is reached with the arc-downstream case. Compared to the unbauffed exchanger and for $Re = 12,000$ – $32,000$, the average Nu gains for the arc-upstream and arc-downstream baffles are 159–284% and 187–331%, respectively. In addition,

Figure 8: Temperature contours (T) for (a) arc-upstream, (b) arc-downstream baffles at $Re = 12,000$ (T values in K).

Figure 9: Outlet fluid temperature profiles for (a) arc-upstream, (b) arc-downstream VGs, at $Re = 12,000$.

Figure 10: Normalized local Nusselt number on upper wall of the channel for various arc-baffles, $Re = 12,000$.

Figure 11: Normalized average Nusselt number with Re for various arc-baffles.

and at the highest Re, the arc-downstream baffle overcomes the other shape of baffles by about 14% in terms of thermal exchange rates (Nu/Nu_0) than that reached with the arc-upstream (Figure 11).

4.9 Friction loss

The variation of the normalized skin friction coefficient (C_f/f_0) on the top wall of the duct is provided in Figure 12. From this figure, both shapes of the baffles give the same trends of C_f/f_0 . Also, an increased C_f/f_0 is observed in the region between the arc-baffles ($0.228 \text{ m} < x < 0.37 \text{ m}$). The arc-upstream and arc-downstream baffles provided, respectively, an increase in C_f by about 73 and 117 times over the unbaffled exchanger. Furthermore, the use of arc-downstream baffles gives higher thermal exchange than that of the other model by about 37%.

Figure 12: Variation of Normalized skin friction coefficient along upper channel wall for various arc-baffles, $Re = 12,000$.

Figure 13: Variation of Normalized friction factor with Re for various arc-baffles.

The changes of the friction factor ratio (f/f_0) vs Re are shown in Figure 13. A proportional increase is observed in the values of $Re(f/f_0)$. In addition, and compared to the smooth duct, the arc-upstream and arc-downstream baffles provided, respectively, an increase in (f/f_0) by about 3–16 and 4–21 times when Re has been changed from 12,000 to 32,000. This means that the arc-downstream baffle generates greater friction loss than the arc-upstream baffle by around 23.266%, at the highest Re.

4.10 Effect of the arc-shaped baffle

Finally, the results of the thermal performance factor (TEF) are summarized in Figure 14. As observed, the

Figure 14: Changes in the thermal enhancement factor vs Re.

Figure 15: Performance comparison with numerical data for various baffles at $Re = 32,000$.

TEF tends to augment with the rise of Re for both shapes of VGs under inspection. At $Re = 32,000$, the optimum value of the TEF is about 1.138 and 1.212 for the arc-upstream and arc-downstream-shaped baffles, respectively. Accordingly, the highest TEF is found with arc-downstream baffle, which is estimated to be higher than that of the arc-upstream baffles by about 6%. The effect of arc-downstream baffles can also be highlighted based on literature data. In the presence of the following conditions: $L = 0.554$ m, $L_{in} = 0.218$ m, $H = 0.146$ m, $D_h = 0.167$ m, $a = 0.08$ m, $b = 0.01$ m, and $Re = 32,000$, their performance has been compared with many previously realized baffles [42]. The relative difference of results shows a remarkable improvement in the presence of an in-line downstream arc-baffle pair by about 7.019, 3.958, 3.130, 3.580, 6.572, 10.815, 7.536, 1.715, 11.485, 10.971, and 8.221% over the upstream-arc, rectangular (simple), triangular, trapezoidal, corrugated, plus, S, V, W, T, and Γ -shaped one-baffle channel, respectively (Figure 15).

5 Conclusion

A numerical inspection has been conducted on the characteristics of the turbulent convection of air flowing in

a baffled rectangular exchanger. Two shapes of arc-baffles were considered, namely, the arc-upstream and arc-downstream shapes. These obstacles were inserted on bottom wall of the exchanger in in-line arrays. The result analysis shows a reinforcement in fluid dynamics with a considerable enhancement in heat exchange in the case of the arc-downstream second obstacle due to the secretion of very strong cells on their back sides, which also caused a significant increase in skin friction coefficients, especially at high flow rates. This second configuration of the arc-baffle (arc-downstream) proved its superiority in terms of thermal exchange rate by about 14% than the other shape of baffle. At $Re = 32,000$, this optimal model of the arc-baffle showed an increase in the enhancement factor by about 7.019, 3.958, 3.130, 3.580, 6.572, 10.815, 7.536, 1.715, 11.485, 10.971, and 8.221% compared to the cases of one baffle, i.e., upstream-arc, rectangular (simple), triangular, trapezoidal, corrugated, plus, S, V, W, T, and Γ , respectively.

Funding: This work was supported by the National Natural Science Foundation of China (No. 71601072) and Key Scientific Research Project of Higher Education Institutions in Henan Province of China (No. 20B110006).

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