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Shape Parametric Numerical Analysis of Thermal-Hydraulic Performance and Fluid Flow in Circular Pipes with Perforated Plate Inserts: Impact of Plate Number, Thickness, and Hole Geometry

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ABSTRACT

The thermal-hydraulic performance and fluid flow of circular pipes with inserts in the shape of a perforated plate have been investigated using three-dimensional computational fluid dynamics (CFD). The effect of the geometry of the holes, thickness of the plates, and number of plates on the pressure drop characteristics and heat transfer augmentation is regularly studied. ANSYS Fluent 2020 R2 was used to do simulations for Reynolds values between 500 and 2000. Circular and rectangular hole configurations were investigated for thicknesses between 5 and 20 mm and plate arrangements of one, two, and three plates. The accuracy of the numerical model was assessed by comparing the results with another paper's, with the greatest difference being 2.56%. The results indicate that the heat transfer performance increases with the number of plates: For Re = 1500, the Nusselt number increases by approximately 7–8% with an increase in plates from 1 to 3. However, with this improvement, there's a visible increase in pressure drop. The thicker the plate, the greater will be its resistance to flow, but there will be little effect on the heat transfer. Based on Performance Evaluation Criterion (PEC) analysis, the thinner the plate and the lower the Reynolds number, the better the thermo-hydraulic performance. The conclusions presented are helpful for the design of industrial thermal systems and small heat exchangers.

1. INTRODUCTION

An important component of thermal and flow engineering is convective heat transfer in internal flow systems. Heat exchangers, cooling systems, and energy conversion devices are all directly impacted. Enhancement of heat transfer is one of the most important fields of research because interaction of fluid flow and thermal transport determines the overall efficiency in these systems. Since circular tubes are commonly used in industries, much has been done to improve their thermal capabilities [1]. Methods of enhancing heat transfer can be classified into two groups: active and passive. Whereas passive methods aim at modifying geometry of the flow, or the properties of the flow with no other energy input, active methods demand some sort of external energy input, like vibration, or electromagnetic forces. An example of this enhancement is the production of swirl flow by the use of twisted tape inserts which has been shown to enhance rate of heat transfer dramatically, but at the cost of rising pressure drop [2]. Real thermal systems using passive methods have been extensively employed due to its ease of use, low cost and simplicity. A whole host of flow-disturbing apparatus, baffles, wire coils, ribs, twisted tapes, etc. have been well explored as passive methods of augmentation. These devices improve the mixing of fluids and increase heat exchange by developing secondary flows and strengthening turbulence, as well as disturbing the thermal barrier layer [3]. Recent numerical studies of perforated tube–plate heat exchangers have also proved the potential of perforated configurations to improve convective heat transfer [4]. Unlike solid inserts, perforated devices allow some fluid flow through their structure, which enhances mixing between fluid layers and generates local jet flows. Experimental studies have also searched in heat transfer improvement by different perforated tube inserts [5]. This action modifies the parameters of the pressure drop along with an enhanced convective heat transfer. Mehta et al. [6] demonstrated that the pressure drop is also increased but slanted perforated disk inserts significantly increase the Nusselt number due to improved turbulence and fluid mixing. According to Alwatban [7] porous vertical baffles enhance the fluid mixing and heat transfer efficiency but at the expense of enhancing resistance to flow. Also, in contrast with conventional layouts, modified baffle structures enhance significantly turbulence intensity and redistribution of the flow, resulting into enhanced thermal performance under numerical simulations [3, 8]. The shape of the channel and the working fluid significantly influence the form of entropy, the resistance to flow, and the thermal performance [9]. Moreover, studies of open-cell materials and perforated plates have confirmed that flow structure and porosity has a strong influence on the properties of heat transfer and pressure drop [10]. Perforated fins are also capable of significantly

enhancing thermal performance, both by enhancing the heat transfer mechanisms and by increasing the effective heat transfer area, according to experimental and numerical studies [11]. Cicek et al. [12] computational studied triangular perforated offset-strip fins and presented that the flow resistance and heat transfer enhancement were significantly affected by the geometry of the perforated fins. Recent advances in perforated geometries have been used to further demonstrate the importance of structural design in controlling thermal-hydraulic performance. By modifying the behaviour of the boundary layer, and increasing turbulence, perforated-serrated fin designs enhance heat transfer [13]. These works illustrate the significance of geometry in optimization of the heat transfer augmentation techniques. Moreover, the effects of channel geometry and nanofluids on the heat transfer efficiency have been studied numerically.These comprehensive studies leave a big gap in the literature. Most of the existing research focuses on single strategies of augmentation, e.g. twisted tapes, perforated fins or baffles, without systematically investigating the overall impact of significant geometric aspects of perforated plate inserts. In particular, the effects of whole shape, plate thickness, and plate number on overall thermal-hydraulic performance of circular tubes have not been studied in detail. Moreover, the previous studies often concentrate on enhancing heat transfer without providing a reasonable evaluation regarding the related pressure drop that is essential to realistic engineering design. The present research, therefore, aims to address these limitations by performing a rigorous numerical study of fluid flow, and heat transfer in circular pipes with perforated plate inserts. The findings of this research present design suggestions and quantitative data towards the maximization of perforated plate patterns in heat exchanger designs. Perforated structures have been investigated as operative passive heat transfer-increasing techniques in the latest investigations. A tangential perforated ring turbulator, as discussed by Chauhan et al. [14] played an important role in enhancing the thermo-hydraulic performance of double-pipe heat exchanger by improving the mixing of fluid and heat transfer between the inner and outer pipesMoreover, Aktas et al. [15] highlighted the need to balance the heat transfer enhancement and pressure-drop penalty in their experimental analysis of the thermal and hydrodynamic performance of plate-fin heat exchangers. Aoua et al. [16] investigated the influence of anisotropic parameters on flow structure and heat transfer enhancement in a partially porous channel, demonstrating that porous layer characteristics significantly affect fluid behavior, thermal distribution, and overall heat transfer performance. Although this research shows that perforated geometries can be viable enhancing strategies, the collective impacts of plate number, plate thickness, and overall geometry of a circular pipe have not been studied.

2. RESEARCH GAP AND NOVELTY

Despite the extensive studies of passive heat transfer enhancement techniques such as twisted tapes, perforated fins, porous baffles, and perforated inserts, most previous studies have focused on one enhancement technique or on a single geometric parameter, and have not investigated a wide range of design parameters [2-6, 13-18]. Recent studies have confirmed the significance of the geometry of the perforation, the intensity of the disturbance in the flow, and the configuration of the structure on the improvement of thermo-hydraulic performance. The use of perforated offset-strip fin designs, perforated fins, and perforated disk inserts, for example, has been found to be very effective for enhancing heat transfer through disruption of the fluid boundary layer and better mixing [6, 14-17]. However, a systematic study on the three parameters – plate number, plate thickness and hole shape – on thermo-hydraulic response of circular pipes is lacking. In addition, there has been little previous attention given to the trade-off between pressure-drop penalties and heat transfer enhancement, which is an important consideration in the design and optimization of real thermal systems [4, 16, 17]. Thus, by conducting a thorough three-dimensional computational fluid dynamics (CFD) analysis of circular pipes with perforated plate inserts, the current study fills this research gap. The effects of plate number (one, two, and three plates), the plate thickness (5, 10, and 20 mm), and hole geometry (rectangular and circular) are studied simultaneously over the Reynolds number range (500–2000). In this work, this important geometric parameter is evaluated in a unified parametric way, and designs are obtained that are able to achieve the best compromises between hydraulic losses and heat transfer enhancement. The results provide helpful guidelines for designs of process piping in industry, cooling systems, and compact heat exchangers where it is desired to optimize pumping power usage and thermal efficiency.

3. NUMERICAL STUDY

A three-dimensional steady state CFD model was used to investigate the convective heat transfer and fluid flow of a smooth circular pipe and perforated plate-inserted pipes. The numerical simulations were performed with ANSYS Fluent. To determine the influence of plate number on thermal-hydraulic performance, three perforated plates, as well as one and two, were incorporated into the computational domain. Six plate thicknesses (5, 8, 10, 15, 18, and 20 mm) were considered to determine the geometric effects in a systematic manner. Moreover, two hole geometries, circular and rectangular holes with the same porosity, were studied as demonstrated in Figures 1 and 2. To tackle the laminar and transitional flow regimes of compact heat exchanger applications, the Reynolds number at the intake was varied between 500 and 2000. The Reynolds number range selected (500 to 2000) is the range of laminar and transitional flow conditions that often occur in energy recovery devices, cooling channels, chemical process pipework, electronic cooling systems and small heat exchangers. These operating conditions are well suited to a number of industrial applications as a result of maintaining reasonable pumping power requirements whilst maintaining consistent flow characteristics. Engineers interested in enhancing heat transmission while minimizing pressure drop penalties can be interested in the thermal-hydraulic performance of inserts in the form of perforated plates in this Reynolds number range and gain valuable design information from the analysis. The fluid under study was water, which was believed to have a constant thermophysical behaviour, be incompressible, and Newtonian fluid. This is a common assumption in a study of the same sort and valid in the temperature range considered. It was decided to model the flow with the laminar viscous model due to the diameter of the pipes used in the study and very low Reynolds numbers. The governing equations of mass,

momentum, and energy conservation were solved by a pressure-based solver. Numerical accuracy was enhanced through second-order spatial discretization algorithms, and the SIMPLE algorithm was applied in the pressure velocity coupling. The effects of heat transfer were represented by switching on the energy equation. The following are the set of boundary conditions: The constant pressure outlet condition was imposed at the outlet, and a uniform velocity profile at the inlet with the assigned Reynolds number was imposed. A steady heat flux situation was imposed on the wall of the pipes and the perforated plates. The no-slip boundary conditions were applied to all solid surfaces. To ensure convergence and numerical stability, appropriate under-relaxation factors, i.e., 0.3 of pressure, 0.7 of momentum, and 1.0 of density, body forces, and energy, were used. Convergence was achieved when the measured outlet temperature and pressure dropped to constant values and the residuals of the continuity, momentum, and energy equations reduced to values less than 10^{-1}. A grid independence analysis was done to ensure that the numerical results are not dependent on the mesh size. The reliability of the existing simulations was also ensured through the validation of the numerical model with the reported findings of Taskesen et al. [9] that showed that there was a good agreement with the findings within the acceptable error margin This numerical framework, in comparison to the previous studies that examined these factors independently, allows a comprehensive evaluation of the combined impacts of the plate number, plate thickness, and hole geometry on the enhancement of heat transfer as well as the pressure drop and provides more insight into the mechanism of the flow.

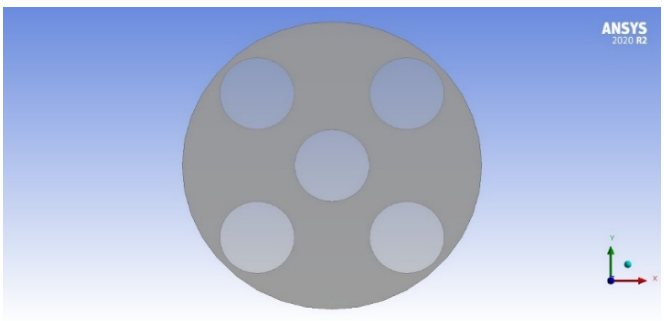


Figure 1. The circular shape of plate's holes

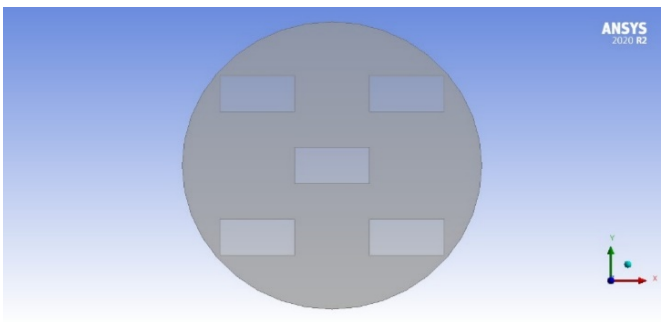


Figure 2. The rectangular shape of the plate's holes

3.1 Boundary conditions

Measurement of the length and diameter of the pipe is according to Taskesen et al. [9]. Figure 3 indicates the geometry (without a plate, one plate, two plates, and three plates). All the boundary conditions are presented in Table 1.

Table 1. Parameters of the model

Description	Value
Length of pipe	1.5 m
Diameter of pipe	16 mm
Number of plates	1, 2, and 3
Number of holes	5
Types of holes	Circular and rectangular
Thickness of plate	(5, 8, 10, 15, 18, and 20) mm
Fluid used	Water
Density of water	998.2 Kg/m^3
Reynolds number	500, 800, 1000, 1500, 1600, 1800, and 2000
Element size	0.001
Sphere radius for mesh plate	0.02 m, 0.03 m
Number of iterations	5000
Heat flux	6000 W/m^2
Temperature inlet	300

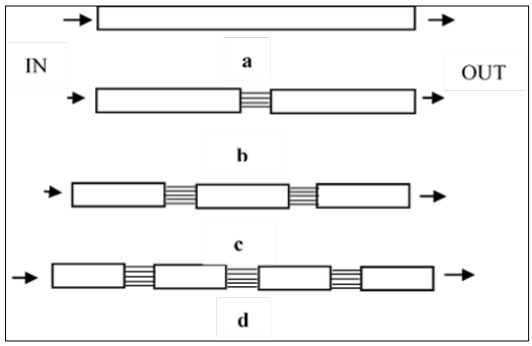


Figure 3. Geometry model: a- pipe without plate b-pipe with one plate c- pipe with two plates d- pipe with three plates

3.2 Mesh generation

ANSYS Fluent 2020R2 was used to make the mesh using the meshing model. In Figures 4 and 5, a tetrahedron mesh having a body size of 0.001 m was used. The size of the plate face sphere of influence and the sphere radius depend on the thickness of the plate; plate thicknesses of 5, 8, and 10 mm have a sphere radius of 0.02 m; plate thicknesses of 15, 18, and 20 mm have a sphere radius of 0.03 m.

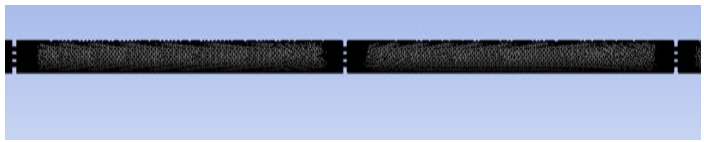


Figure 4. Mesh of pipe with three plates

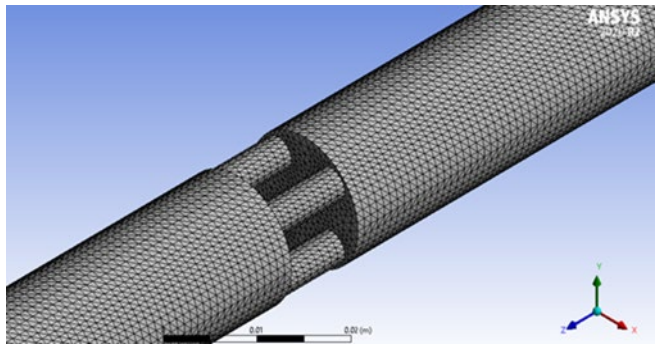


Figure 5. Magnifier photo for mesh of one plate

Table 2. Validation of mesh

Trial No.	Element Size (m)	Number of Elements	ΔP (Pa)	ΔP Error (%)	(T_{out}) (°C)	Error (%)
1	0.0005	20,465,749	41	—	307.82	—
2	0.001	2,596,253	40	2.44	307.85	0.0097
3	0.002	410,819	38.5	6.10	307.91	0.0198
4	0.003	189,717	37	9.76	307.99	0.0269

3.3 Grid convergence study

A grid independence study was conducted to ensure that the numerical results are independent of mesh size and to determine the optimal mesh resolution. Four different mesh sizes, ranging from 189,717 to 20,465,749 elements, were tested. The outlet temperature and pressure drop were monitored as key parameters to evaluate mesh sensitivity. The relative error between successive meshes was calculated using the following equation:

$$Error\% \ = \left| \frac{\emptyset_{fine} - \emptyset_{coarse}}{\emptyset_{fine}} \right| * 100 \qquad (1)$$

where, ϕ represents the monitored parameter (e.g., outlet temperature or pressure drop), and the subscript "fine" refers to the finer mesh, while "coarse" refers to the compared mesh. The results of the grid independence analysis are presented in Table 2. The difference in the pressure drops across the finest and medium mesh was found to be approximately 2.44%, which is within acceptable numerical tolerance, and the change in outlet temperature was found to be less than 0.03%. The mesh consisting of 2,596,253 elements was selected to be used in all the simulations due to it offering an appropriate trade-off between the computational cost and the solution accuracy.

The mesh with 20,465,749 elements was numerically the most accurate, but was slightly better than the mesh with 2,596,253 elements. The outlet-temperature deviation remained at less than 0.03%, and the difference between the two meshes was 2.44% in terms of the pressure drop. The other way around, however, caused a significant increase in the simulation time and processing cost. The mesh with 2,596,253 elements was selected as the best grid because it is mesh independent and has much fewer processing requirements.

3.4 The governing equations

The governing equations in three-dimensional, laminar, steady-state, incompressible flow with conjugate heat transfer have been solved using the finite volume method. It was believed that the working fluid was characterized by constant thermophysical parameters and was Newtonian. Such assumptions are valid in the case of moderate temperature variations and the range of Reynolds numbers (500–2000). Heat conduction was considered in the solid plates, and conservation laws of mass, momentum, and energy were applied to the fluid region [6, 18].

Continuity equation

$$\nabla . u \ = \ 0 \qquad (2)$$

Energy equation (fluid region)

$$\rho c_p (u . \nabla T) \ = \ k_f \nabla^2 T \qquad (3)$$

Energy equation (plate region)

$$k_s \nabla^2 Ts \ = \ 0 \qquad (4)$$

Momentum equation

$$\rho (u . \nabla) u \ = \ -\nabla p + \mu \nabla^2 u \qquad (5)$$

Heat flux continuity at the fluid–solid interface

$$kf \left(\frac{\delta Tf}{\delta n} \right) \ = \ ks \left(\frac{\delta Ts}{\delta n \delta} \right) \qquad (6)$$

At the fluid–solid interface, both temperature continuity and heat flux continuity were enforced to accurately model conjugate heat transfer between the fluid and perforated plates, which is consistent with previous studies on perforated structures and heat transfer enhancement [10].

Performance Evaluation Criterion (PEC)

The overall thermal-hydraulic performance was evaluated using the PEC, defined as [4, 14, 15]:

$$\mathrm{PEC} \ = \ \frac{\left(\mathrm{Nu_{modified}} / \mathrm{Nu_{smooth\ pipe}} \right)}{\left(\mathrm{f_{modified}} / \mathrm{f_{smooth\ pipe}} \right)^{1/3}} \qquad (7)$$

The finite volume method of ANSYS Fluent was employed to discretize and numerically solve the above equations. The flow field, temperature gradient, and the properties of enhancement of heat transfer and pressure drop in pipes that have in them inserts of perforated plates can all be correctly predicted courtesy of this formulation.

3.5 Validation of the model

Taskesen et al. [9] conducted a numerical study of the heat transport as well as the flow properties of water and nanofluids in pipes of different shapes. As observed in Figure 6 and Table 3, the validation results show there is a good agreement between the current numerical results and the data reported by Taskesen et al. [9]. The present numerical model was checked by comparing the change in Nusselt number with Reynolds number to the results that were reported by Taskesen et al. [9]. The largest variance of about 2.56% confirmed the accuracy and reliability of the current model, which is within an acceptable range of numerical simulations.

Table 3. Percentage error validation

Re	Nu (Taskesen)	Nu	Error (%)
500	6.3	6.2	1.59%
1000	7.8	7.6	2.56%
1500	8.9	8.7	2.25%
2000	9.5	9.4	1.05%

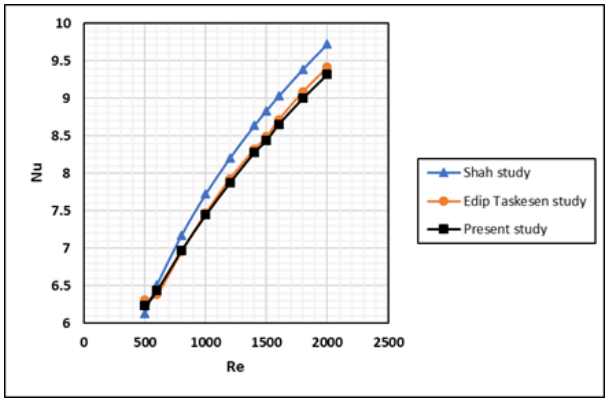


Figure 6. Model validation

4. RESULTS AND DISCUSSION

4.1 Effect of the number of plates on heat transfer

The dependence of the Reynolds number on the Nusselt number of the different numbers of perforated plates is shown in Figures 7 and 8. It can be seen that the Nusselt number increases with an increase in the number of plates, in the entire range of Reynolds number (500–2000). A quantitative analysis shows the Nusselt number increases by about 7.2% between a single plate and three plates at a Reynolds number of 1500, reflecting a positive gain of about 7.2. Besides, the Nusselt number increases by an average of (3–4)% with each additional plate. The perforated plates' introduction of flow disturbances is responsible for this improvement. Periodic discontinuities in the flow are induced by the increased number of plates, which create jet-like fluxes through holes and promote the development of recirculation zones downstream of each plate. These aspects enhance the convective heat transfer rate by enhancing fluid mixing and producing a more effective disturbance to the thermal boundary layer. Also, the viscosity between the primary flow and the secondary flow structures formed by the perforations increases with the increase in Reynolds number, thus resulting in a more pronounced enhancement of the performance of the heat transfer. These findings are consistent with previous studies on perforated inserts and flow enhancement techniques, which have found that an increased number of flow-disturbing elements will enhance the performance of the heat transfer by enhancing turbulence and mixing [6, 9]. The perforated plates create an interruption of flow, which is the main reason for the increased heat transfer. As more plates are added, more jet-like flows are produced through the perforations, thus mixing the fluids and more effectively breaking the thermal barrier layer. As the value of the convective heat transfer coefficient increases, the Nusselt number will increase. Also, the fluid accelerates and decelerates in every successive perforated plate, which enhances momentum exchange between the near-wall area and the core flow. This enhancement becomes more apparent at higher Reynolds numbers, resulting in better thermal performance for higher fluid-structure interactions through the perforated structures.

4.2 Flow structure analysis

Figure 9 shows the streamline distribution of the magnitude of velocity inside the pipe with perforated plates at Re = 800 and a plate thickness of 8 mm. The three figures (a–c) are used to represent different plate layouts along the pipe. Figure 9(a), illustrating the streamlines in a downstream direction along the axis of the pipe, shows a rather uniform flow just upstream of the perforated plate. The shaped restrictions of the plate give rise to a slight distortion of streamlines, as one approaches the perforated area. The passage of the fluid through the holes in Figure 9(b) will result in the fluid accelerating locally. Minor changes in the streamline paths can be observed along the plate region, thus pointing out local disturbances despite the overall flow remaining largely laminar. The flow later recovers, and the streamlines again take a near uniform distribution further down the stream as shown in Figure 9(c). This implies that disturbances caused by the perforated plate are relatively localized and distributed throughout the length of the pipe. The small scale distortion and local acceleration of the flow assist in enhancing mixing at the micro scale level even in the case where there are no significant recirculation zones. These effects can be attributed to the improvement in heat transfer performance seen in Section 3.1, and are sufficient to destabilize the thermal barrier layer.

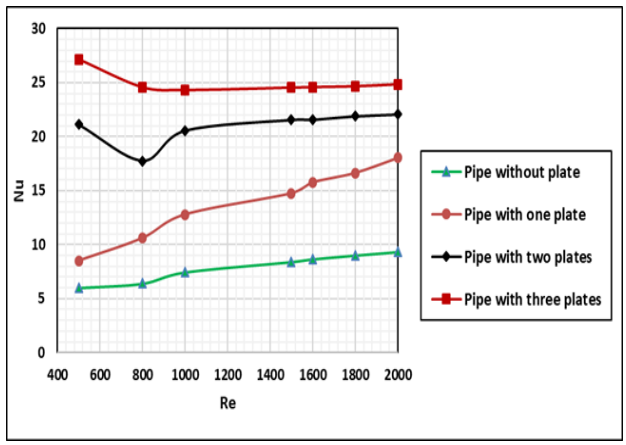


Figure 7. Nu versus Re at different numbers of plates with rectangular holes and a thickness of 10 mm

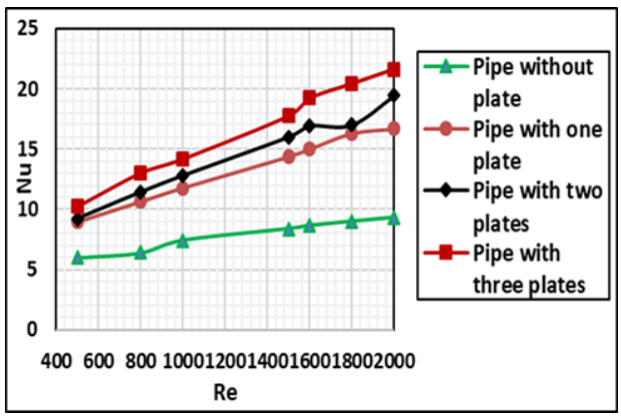
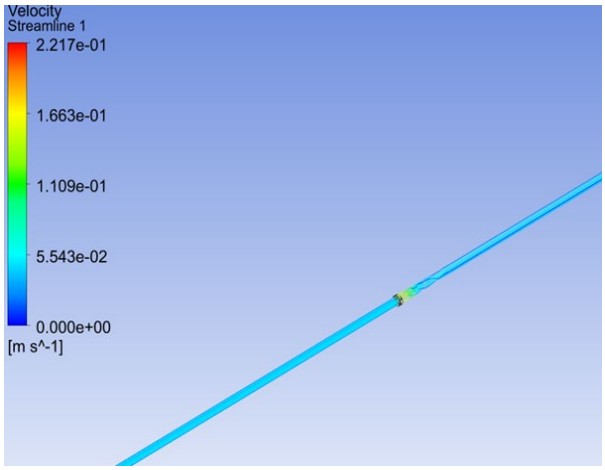


Figure 8. Nu versus Re at different numbers of plates with circular holes and a thickness of 10 mm

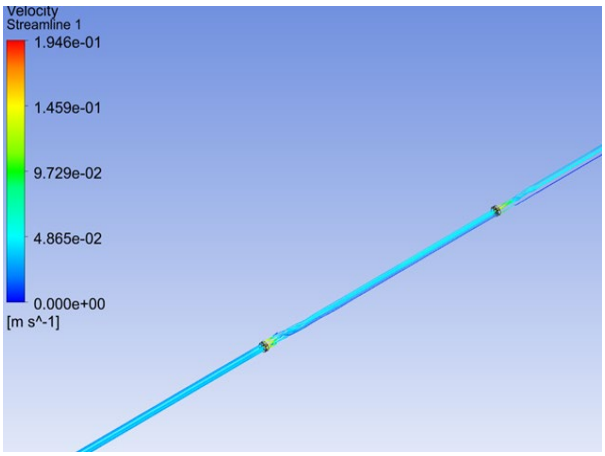
4.3 Effect of plate thickness on heat transfer performance

The reason for the less significant effect of the plate thickness in the current arrangement is because of the major heat transfer enhancement mechanism. The main ways that the perforated plates enhance thermal performance are through

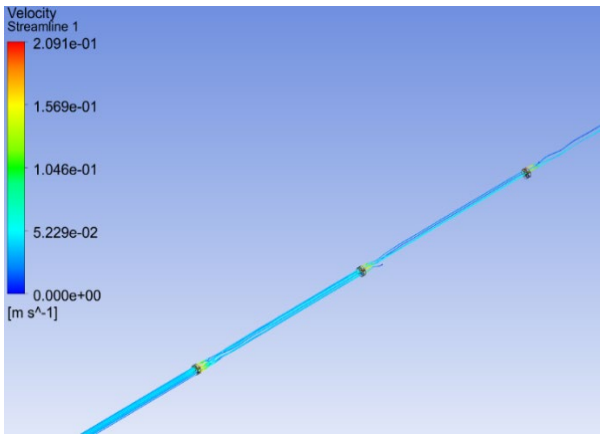
fluid mixing, jet creation, local flow acceleration, and thermal boundary-layer disruption caused by the perforations. In Figures 10 and 11, the increase in the thickness of the plate from 5 mm to 20 mm does not significantly change the heat transfer process, due to the fact that most of the heat transfer mechanisms are determined by the shape of the holes and the flow conditions. However, the thicker the plate, the longer the path to get through the holes and the higher the pressure losses and flow resistance will be. Consequently, in the range of investigation, the thickness of the plates significantly influences the hydraulic performance and not so much heat-transfer enhancement.



(a)



(b)



(c)

Figure 9. Streamlines of velocity magnitude illustrating the fluid flow structure in the pipe with a perforated plate, Re 800, and thickness 8 mm

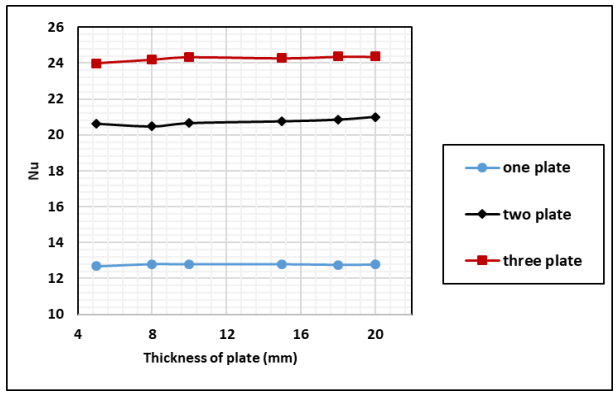


Figure 10. Nu versus thickness of plates for rectangular holes at Re = 1000

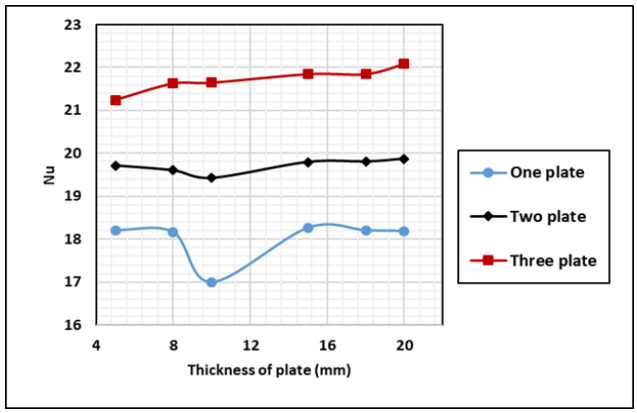


Figure 11. Nu versus thickness of plates for circular holes at Re = 1000

4.4 Effect of the shape of the plate on heat transfer

In the case of perforated plates with circular and rectangular holes and at thicknesses of 10 mm and 20 mm, Figure 12 illustrates the variation of the Nusselt number with the Reynolds number. In both hole shapes, it is determined that the Nusselt number increases with the increase in Reynolds number, which is in agreement with the enhancement of convective heat transfer as the Reynolds number rises. A comparison of the two geometries reveals that under equal operating conditions, the rectangular holes can give a comparatively higher Nusselt number than the circular holes. This improvement is due to the sharp edges of the rectangular perforations that cause more disruptions to local flows and promote more mixing near the plate region. However, at the range of Reynolds numbers investigated, the discrepancy between the two geometries is rather small. In most cases, it can be found that the thermal performance of both geometries has similar values, implying that the overall enhancement of heat transfer is dependent on the interaction between flow conditions and plate arrangement, as well as on the shape of the hole. Also, in both geometries, increasing the plate thickness beyond 10 mm, to 20 mm, is not a significant change to the Nusselt number, which is consistent with findings discussed in Section 3.3. All in all, there is a slight difference in heat transfer performance between rectangular holing and square holing under all operating conditions, but that difference is not significant enough to justify a conclusion that one of the geometries is always superior under all working conditions.

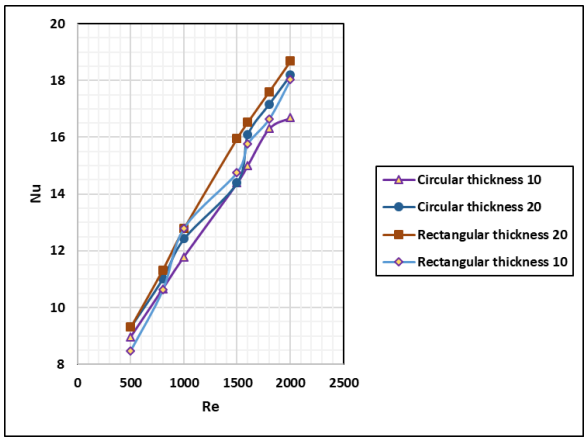


Figure 12. Nu versus Re for two shape holes for one plate and different thicknesses

4.5 Effect of the number of plates on fluid flow

Figures 13 and 14 illustrate how the pressure drop (ΔP) varies with the flow velocity in the case of varying numbers of perforated plates with circular and rectangular holes, respectively. Alterations in the behavior of laminar flow in a smooth pipe are reflected in the observation that, under the condition of no plates, the drop in pressure is relatively small and changes smoothly with velocity. In all configurations of the pressure loss, the loss increases at a tremendous rate with increasing velocity when perforated plates are introduced. The rationale behind this is the fact that the plates not only block the flow, but they also force the fluid through the holes, providing additional resistance to the flow. More so, the pressure drop slowly rises with the increase in the number of plates. The three plate structure demonstrates the greatest pressure reduction of the examples analyzed, with the two plate and single plate structures coming in second and third, respectively. This tendency can be attributed to the cumulative influence of multiple flow constraints that make the fluid repeatedly accelerate and decelerate as it passes through the plates. These recurring disruptions are the cause of higher pressure losses and an increase in velocity gradients, shear stresses, and energy dissipation. Momentum loss is further increased downstream of each plate by partial recirculation and localized flow separation. As was covered in earlier sections, there is a penalty associated with the rise in pressure drop, but there is also an improvement in heat transfer performance. As a result, choosing the right number of plates should be based on striking the best possible balance between pressure loss and heat transfer enhancement.

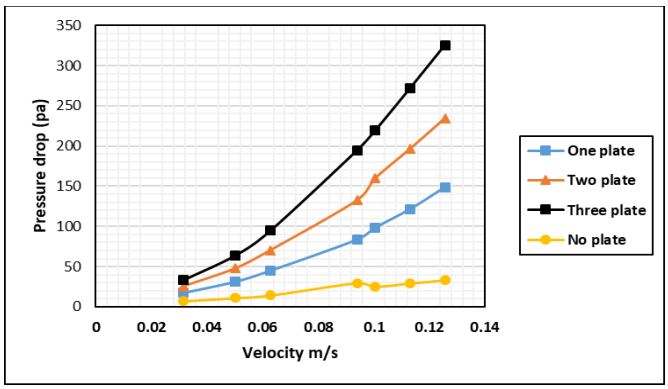


Figure 13. Pressure drop versus velocity for circular holes and a thickness of 5 mm at different numbers of plates

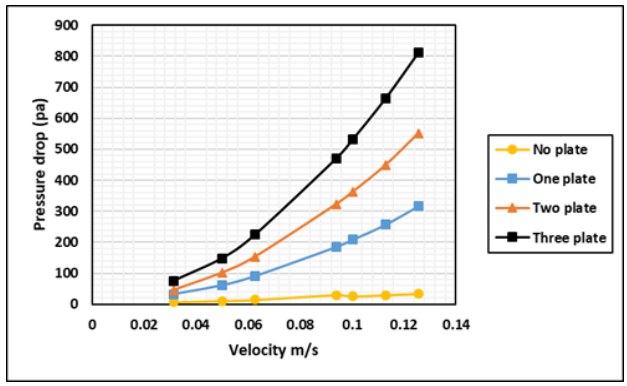


Figure 14. Pressure drop versus velocity for rectangular holes and a thickness of 5 mm at different numbers of plates

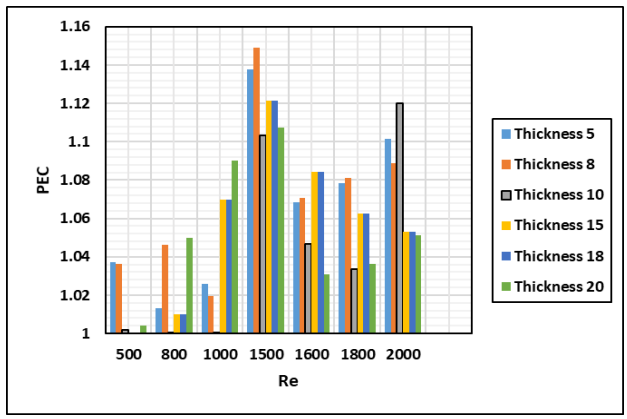


Figure 15. Performance evaluation criteria (PEC) for circular holes for different thicknesses for two plates

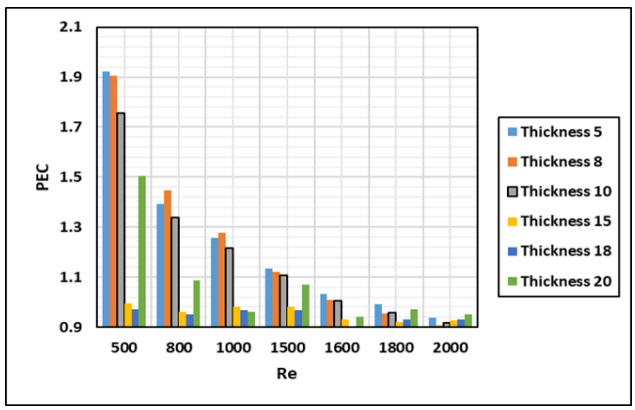


Figure 16. Performance evaluation criteria (PEC) for rectangular holes for different thicknesses for two plates

4.6 Performance evaluation criteria

4.6.1 Performance Evaluation Criterion for circular and rectangular

Figures 15 and 16 illustrate how the PEC changes with the Reynolds number, in the case of two-plate designs with the plate thickness varying between circular and rectangular holes, respectively. The results indicate that Reynolds number and plate thickness are significant factors in influencing PEC. The losses due to pressure decrease with increasing Reynolds number while the heat transfer increases to a lesser extent. Generally, the thinner the plate, the lower the hydraulic resistance, yet the higher the PEC value, when thermal resistance is considered adequate. This means that lower plate

thicknesses can be used to achieve a better balance between pressure drop penalties and increased heat transfer. The results indicate that thinner plates and moderate Reynolds numbers are optimum conditions to achieve good thermo-hydraulic performance. Compared to larger plate thicknesses (1520 mm), intermediate plate thicknesses (58 mm) offer higher PEC values for both plate geometries. This means that the thinner plates are more apt to strike a good balance between pressure drop and heat transfer enhancement. Although the performance of the heat transfer varies modestly, PEC reduces with the increase in plate thickness owing to the increase in flow resistance. The PEC also attains relatively optimum values at relatively moderate Reynolds numbers (Re = 1000-1600), indicating a reasonable operating range to achieve better thermal-hydraulic performance. The counteracting influences of pressure loss and the enhancement of heat transfer can explain the pattern observed. The perforated plates add resistance to flow even when they cause local flow disturbances, which help to mix and transmit heat. At smaller thicknesses, higher PEC values are achieved due to the fact that the rising heat transfer is more than the pressure penalty. However, as the thickness increases, the pressure drop dominates, which leads to the PEC reducing. In general, the results indicate that the best performance is achieved at lower plate thicknesses and moderate Reynolds numbers, where an effective balance between friction loss and heat transfer enhancement is achieved.

4.6.2 Effect of plate numbers on the Performance Evaluation Criterion

Figure 17 illustrates the variation of the PEC of circular holes in plate numbers with different plate thicknesses. The findings revealed that designs that have one plate usually have higher values of PEC as compared to those designs using two or three plates. The trend can be attributed to the fact that adding the first plate is associated with a significant enhancement in the heat transfer as well as an increase in pressure drop by only a slight margin. However, when an increasing number of plates are added, the losses of pressure increase significantly and the increase in heat transmission is only slight. Since the number of frictional penalties exceeds the number of thermal benefits in multi-plate systems, summation of frictional penalties to thermal benefits leads to a decrease in the total PEC. A similar tendency with respect to the rectangular holes is illustrated in Figure 18. In similar cases, rectangular holes are usually characterized by a low PEC value when compared to circular holes. This could be attributed to the sharper edges of the rectangular perforations, which enhance dissipation of energy and reinforce local flow disruptions, leading to greater pressure drops. Rectangular holes may improve local mixing, but the overall thermal-hydraulic performance is decreased due to the corresponding increase in flow resistance. Owing to this, a more beneficial tradeoff between pressure drop and heat transfer increase is typically achieved with the round hole configuration.

4.7 Engineering implications and economic considerations

The reported results are beneficial for design suggestions for realistic thermal systems. The more plates with perforations, the higher the heat transfer efficiency; however, the higher the pressure drop. During the system's lifetime, a greater pressure drop would require increased pumping power and increase operating costs. So, heat transfer enhancement should not be the sole criterion for selecting the optimum plate configuration. The current findings indicate that single-plate layouts generally have the highest PEC, which is a combination of the hydraulic losses and thermal augmentation. Higher Nusselt numbers can be attained by using numerous plates, but the extra heat-transfer benefit could not outweigh the increased energy consumption and pumping needs. Moreover, they have lower flow resistance but similar heat transfer properties, so thinner plates are advantageous in both hydraulic and manufacturing aspects. Thus, for small heat exchangers, cooling systems and industrial process pipework, designs that can achieve a reasonable level of thermal efficiency, maintain reasonable pressure drops and are optimised for operating costs might prove to be the most economical.

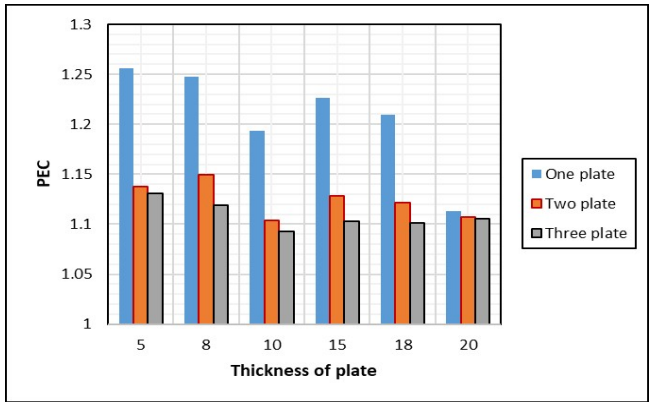


Figure 17. Performance evaluation criteria (PEC) for circular holes at different numbers of plates at Re = 1500

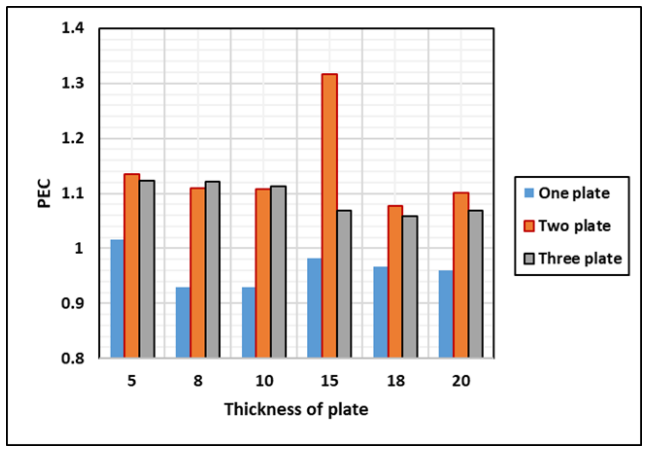


Figure 18. Performance evaluation criteria (PEC) for rectangular holes for different numbers of plates at Re = 1500

5. CONCLUSIONS

This work explored the effects of plate number, plate thickness, and hole form on fluid flow and heat transfer in a circular pipe with perforated plates through the use of three-dimensional CFD. The results showed that increasing the number of plates enhances the efficiency of heat transfer. As an example, the Nusselt number changed by approximately 7–8% when the number of plates increased by one to three at Re = 1500. But, since the plates give more resistance to the flow, this enhancement comes with a significant increase in pressure drop. In the range studied (520 mm), the effect of plate

thickness on heat transmission was observed to be very small, with only slight variations in the Nusselt number. But the larger the thickness, the larger the pressure losses, which negatively impact the overall thermal-hydraulic performance. The geometry of the holes is also very important in determining system performance. Due to the more intense local flow disturbances, rectangular holes generally yield a slightly higher rate of heat transfer. However, this is associated with increased pressure drop that might reduce the overall efficiency. Plate configuration and flow conditions both affect ideal performance, according to the PEC study. When the heat transfer enhancement exceeded the pressure penalty (low to moderate Reynolds numbers), the maximum PEC values were observed. There was also a decrease in PEC due to higher friction losses in multi-plate layouts, but the best performance was achieved in a single-plate system with round holes. In general, it can be concluded that the ideal design requires a trade-off between pressure drop and heat transfer improvement, where the thickness of plates and the shape of holes play secondary roles in the optimal design within the studied range, and the number of plates and hole shape are the most significant factors.

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NOMENCLATURE

$\nabla \cdot u$	Velocity divergence m/s^1
μ	Viscosity (Pa. s)
A	Surface area of the cross-section of pipe (m^2)
C_p	Specific heat (kJ kg-1 K-1)
h	Average convective heat transfer coefficient (W/ m^2.K)
K	Thermal conductivity (W/m .K)
Kf	Thermal conductivity of fluid
L	pipe length (m)
N_U	Nusselt number
q''	Heat flux (W/m^2)
Re	Reynolds number
T_b	Bulk temperature (K)
T_S	Temperature of solid K
T_w	Wall temperature (K)
V	Average fluid velocity (m/ s)
ρ	Density (kg/m^3)
$\partial T/\partial n$	Temperature gradient normal to surface K/m
$Nu_{modified}$	Nusselt number for modified pipe
Nu_{smooth}	Nusselt number for smooth pipe
f	friction factor
$f_{modified}$	friction factor for modified pipe
f_{smooth}	friction factor for smooth pipe
PEC	Performance Evaluation Criterion