Enhancement of Heat Transfer in Perforated Circular Finned-Tube Heat Exchangers: A Numerical Investigation

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Abstract. The effects of employing perforations of circular shape in circular finned-tube heat exchangers (CFTHEs) are numerically investigated in the present study using Computational Fluid Dynamics (CFD) software. Simulations performed on the validated CFD model shows considerable increase in the Average Nusselt Number for perforated circular finned-tube heat exchangers (PCFTHEs) as compared to their solid counterpart, with only a slight increase in Euler Number for $10{,}000 \leq Re \leq 50{,}000$. Diameter of perforations (d_p) is kept equal to 6mm and optimum range of number of perforations per fin (n_p) by analysis is found to be 16 to 20. For this range, the performance of PCFTHEs improves with the increase in number of perforations per fin. The maximum increase in Average Nusselt Number is 24% for d_p =6mm and n_p =20 at Re=10,000. The minimum increase in Average Nusselt Number is 3% for d_p =6mm and n_p =14 at Re=40,000. This increase in Average Nusselt Number on introducing perforations is due to (i) flow of localized air jets through the perforations, and (ii) reduction in recirculation region caused by the separation of flow. This improvement in the performance of CFTHEs on introducing perforations is achieved with the maximum and minimum reduction in weight of 41% and 29% respectively.

1. Introduction

Circular finned-tube heat exchangers (CFTHEs) are most commonly used in the industries because of their simple, cost-effective and efficient design. In the past, many researchers have explored the effect of geometrical parameters and different finned-tube arrangement on the performance of CFTHEs. In order to further enhance the performance of CFTHEs, various modifications in fins are still being explored. One of those modifications is creating porosity or slots in fins, which along with improving the performance of CFTHE’s will lead to reduction in cost and weight of the material. Everywhere in this study CFTHEs means the one without perforations and PCFTHEs is the one with perforations.

Most of the researches regarding perforation technique are performed on heat sinks. Heat sink is nothing but a type of heat exchanger only that uses a fluid medium, often air to dissipate the heat generated by an electronic or a mechanical device. Ehteshum et al. [1] and Dhanawade [2] carried out experimental studies on perforated plain fins in turbulent flow. Ehteshum et al. found that fins with circular perforations show heat transfer enhancement, lower pressure drop and higher effectiveness with increase in the number of perforations. Dhanawade and Dhanawade studied the effect of lateral circular perforations in plain fins. They concluded that optimum diameter of perforations depend on the applied heat flux density. Most of the researchers concluded that increase in the number of perforations lead to the reduction in length of recirculation zone and size of wakes around the fin.

Some other investigations were done on heat sinks in order to study the effect of perforations in pin fins. At first, researchers like Dhumne et al. [3] and Sahin et al. [4] performed experimental investigations to analyze the effect of single perforations in pin fins. Sahin et al. found that circular perforations lead to heat transfer enhancement in blocks of square cross-section arranged in inline arrays. Dhumne et al. examined the effect of circular perforations in staggered tube arrays. These studies concluded that single perforations are helpful in increasing the heat transfer and reducing pressure drop as compared to their equivalent solid systems. Al-Damook et al. [5] carried out experimental and numerical studies on pin fins having multiple perforations of circular shape; concluding that as the number of perforations increase, there is a monotonic increase in the rate of heat transfer and monotonic decrease in pressure drop. It was also found that the location of perforations had very less effect on the performance of pin fins. Later on another computational study by Al-Damook et al. [6] was performed on pin fins having single rectangular, slotted and notched fin perforations. It was found that the increase in size of rectangular perforations follow the same trend as before i.e. heat transfer increases and pressure drop decreases .

One of the most significant experimental investigation on PCFTHE's was carried out by Lee et al. [7]. They examined the effect of perforations in circular finned-tubes on the performance of CFTHE's. The increase in air side heat transfer coefficient is found to be 3.31% and 3.55%, whereas increase in pressure drop across the heat exchanger is 2.08% and 0.68% for the 4-hole and 2-hole perforated circular finned-tube cases, respectively. Jang et al. [8] found that the staggered arrangement is better as compared to in lined one for increasing the average heat transfer coefficient and decreasing pressure drop in plain fins. As per the recommendations of Nir [9] and Webb [10] the heat transfers correlation of Briggs and Young [11] and Robinson and Briggs's [12] correlation of pressure drop should be preferred for a staggered tube arrangement.

1.1. Problem definition

After going through the literature it can be concluded that most of the investigations regarding fin perforations are done for plain and pin fins. The effects of perforations in circular finned tubes on the air-side performance of CFTHEs are being explored computationally in the current study. The objective of the study includes finding Optimum range of number of perforations per fin when their diameter is kept as 6mm. This can be later used to see the effects of varying number of perforations on the thermal hydraulic performance of PCFTHEs.

2. Numerical Method

This paper consists of numerical study of CFTHE that is simulated on CFD software ANSYS FLUENT 13.0 [13] and the analysis is limited to the outside of the tubes. Assumptions: (i) Three-dimensional turbulent fluid flow over an array of perforated and un-perforated circular finned-tubes is considered to be steady and incompressible, and (ii) Dry air is considered as the cooling medium and condensation effects are neglected. The finned-tubes are placed in a staggered arrangement and dry air flow perpendicular to the tubes in order to increase the heat transfer performance. Model symmetry is taken into consideration in order to decrease its complexity as well as the computational time required. Continuity, momentum and energy equations are used to govern this steady and incompressible flow inside a heat exchanger. RNG k-ε turbulent model is opted and relaxation factor of 0.8 is taken as initial iteration value. In solver steady time, 3D axisymmetric with absolute velocity formulation is taken and solver is taken a simple pressure based. Second order upwind discretization setup is considered for energy and momentum differential equations. The relaxation factor of momentum and energy equation is 0.7 and 1 respectively. The convergence criteria for residuals of momentum and energy equation are 1x10-3 and 1x10-6 respectively.

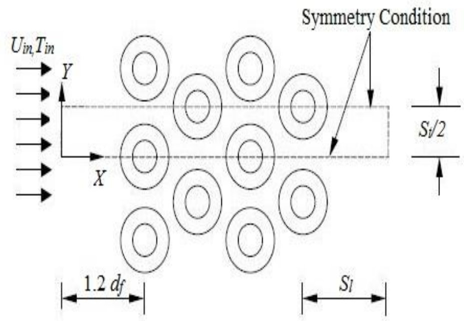
2.1. Model description and computational domain

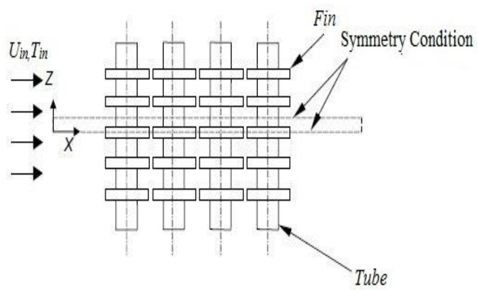
Table 1 shows specifications of the model borrowed from a heat exchanger which is already being used as an oil cooler in the industries, to see the feasibility of the study in real world applications. The heat exchanger consists of 4 rows and 5 columns, along with several fins that are integral part of the tube in each row as can be seen in Figure 1. The fin pitch is taken constant for the model since there will be no hindrance in flow due to the formation of frost as the cooling medium is dry air. For simplicity while modeling a perforated circular finned-tube heat exchanger, circular holes or perforations are done at the mid portion of the fin surface and are distributed symmetrically along the same. The thickness of the tube is considered to be 1mm, which is of no significance in this study, since the tube is considered to be at a constant temperature in all simulations. The simulations are done for the inlet velocity in the range, 3 m/s to 17 m/s.

Table 1. Dimensions of the heat exchanger model

Outside diameter of tubes, *d*	29 mm
Tube thickness, *t*	1 mm
Diameter of fin, d_f	51 mm
Fin thickness, δ	0.53 mm
Fin pitch, *p*	2.30 mm
Fin spacing, *s*	1.77 mm
Transverse tube pitch, S_t	55.56 mm
Longitudinal tube pitch, S_l	48.12 mm
Number of tube rows in flow direction, *n*	4

Computational domain considered in the current study is shown by dashed lines along with the symmetry conditions in Figure 1. In Figure 1, it can be seen that lines of symmetry are passing through the centres of fin thickness and the mid plane between two adjacent fins. *Y* direction is perpendicular to the flow of stream in *X* direction. The length of fluid domain at the upstream boundary is $1.2\times d_f$ from the first row centre line and that at the downstream boundary is equal to longitudinal tube pitch from the centre line of the last row in order to avoid approximation effects in the flow boundary, if any and eliminate periodic vortex street in the wake.





(a) Front view (b) Top view

Figure 1. Finned-tube staggered arrangement (Highlighting computational domains).

2.2. Grid independence

The grid size used for the simulations should be such that it not only increases the accuracy of the results but also decreases the computational time as well. Keeping this in mind the number of grid elements taken in the study is near about 278000.

2.3. Material of construction

Depending on the application, either copper or aluminium is selected as the material for heat exchanger because of their optimum thermal properties and corrosion resistance. For this particular study on compact heat exchangers, aluminium is used as the material for both fin and tubes owing to its low cost and less weight. The density of aluminium is taken as 2700 kg/m3.

2.4. Boundary conditions

Following boundary conditions are needed to be given to carry out the simulations: (i) Velocity boundary condition at the inlet (i.e. from 3m/s to 13m/s) with a constant temperature of 300K and turbulent intensity of 15%, (ii) Pressure boundary condition at the outlet with gauge pressure set as zero, (iii) No slip-condition at the tube surface which is kept at a constant temperature of 598K and heat transfer takes place by convection, (iv) No slip-condition at the fin surface and heat transfer takes place by the combination of conduction and convection, and (v) At the plane of symmetry; in normal direction, both the velocity components and temperature gradients are assumed to be zero.

2.5. Post processing

In addition to visualizing the flow inside the computational domain, CFD analysis is used to determine different dimensionless numbers in order to compare the performances of different heat exchangers under consideration. The average heat transfer coefficient ($\bar{h}$) is assumed to be constant and uniform and is evaluated as the area-weighted average of heat transfer coefficient over fins and tubes.

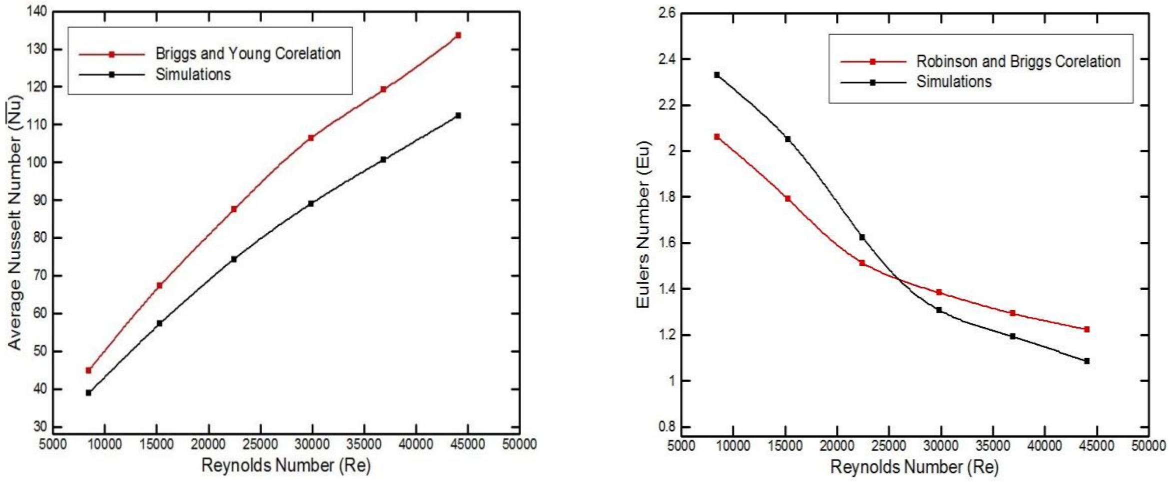
3. Results and Discussion

3.1. Validation

The numerical model of CFTHE used here is validated with the help of reliable correlations to establish its accuracy. The different correlations used here are as follows:

$$\overline{Nu} = 0.134\, Re^{0.681} Pr^{\frac{1}{3}} \left(\frac{s}{h_f}\right)^{0.2} \left(\frac{s}{\delta}\right)^{0.1134} \quad - \text{by Briggs and Young} \tag{1}$$

$$Eu = 18.93\, Re^{-0.316} \left(\frac{S_t}{d}\right)^{-0.927} \left(\frac{S_t}{S_d}\right)^{0.515} n \quad - \text{by Robinson and Briggs} \tag{2}$$



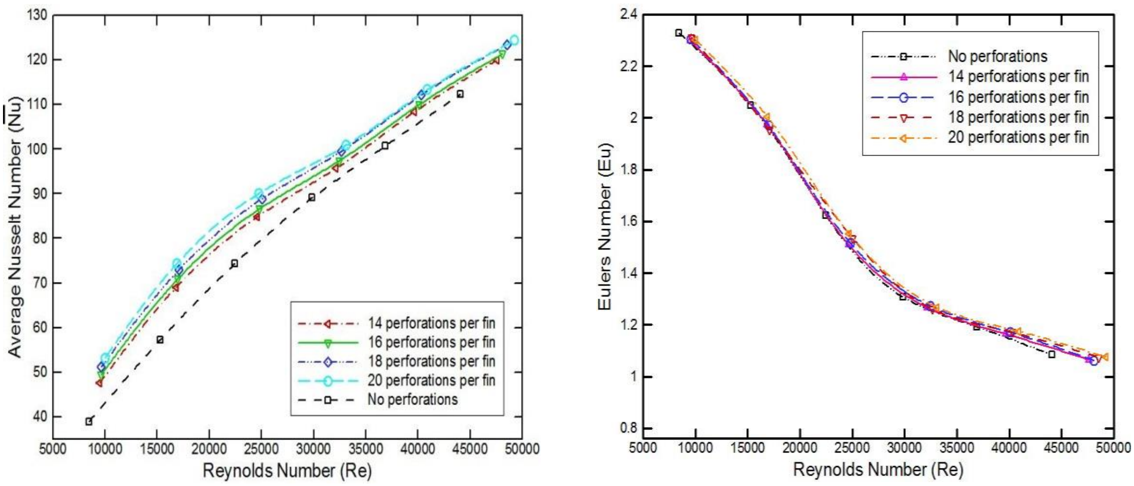
(a) (b)

Figure 2. Comparison of the results obtained by simulations and correlations of (a) Average Nusselt Number (b) Euler Number for CFTHEs.

As can be seen in Figure 2, results for both simulations and correlations follow the same trend i.e. the Average Nusselt Number monotonically increases whereas Euler Number monotonically decreases with Reynolds Number. This is because as the Reynolds Number increases in turbulent flow the intense mixing of air enhances heat and momentum transfer between its particles, which in turn increases the convection heat transfer coefficient and the friction force thus resulting in increase in the Average Nusselt Number and decrease in Euler Number. 100% of the numerical results for Average Nusselt Number and Euler Number are correlated within the deviation of ±15%. The deviation in correlated and simulated results is because the temperature and pressure distribution inside the heat exchanger in reality of unsteady state conditions will be different from that visualized in simulations assuming steady state.

3.2. Effect of perforations on the performance of CFTHEs

The first thing to be noted is that PCFTHEs follow the same trend as CFTHEs i.e. the Average Nusselt Number monotonically increases whereas Euler Number monotonically decreases with Reynolds Number. For diameter of perforations equal to 6mm, optimum range of the number of perforations per fin is found to be 14 to 20. It can be seen in Figure 3 that for this range, the Average Nusselt Number increases as number of perforations per fin increase, with only a slight increase in the Euler number, for all inlet velocities considered.



(a) (b)

Figure 3. Variation of (a) Average Nusselt Number (b) Euler Number; for fins with varying number of perforations per fin and d_p= 6mm.

Quantitative comparison of the performance of PCFTHEs with different number of perforations per fin at different Reynolds Number is shown in Figure 4. Following observations can also be noted: (i) Percentage increase in Average Nusselt Number for PCFTHEs with the given number and diameter of perforations as compared to their solid counterpart is more at lower velocities i.e. lower Reynolds Number than higher velocities i.e. higher Reynolds number, and (ii) Percentage increase in Euler Number for PCFTHEs with the given number and diameter of perforations as compared to their solid counterpart is more at higher velocities i.e. higher Reynolds number than lower velocities i.e. lower Reynolds Number.

For diameter of perforations equal to 6mm, the maximum increase in Average Nusselt Number is about 24% at Re=10,000 for PCFTHE with 20 perforations per fin and minimum increase in Average Nusselt Number is 3% at Re=40,000 for PCFTHE with 14 perforations per fin. For diameter of perforations equal to 6mm, the minimum increase in Euler Number is 0.19 % at Re=10,000 for

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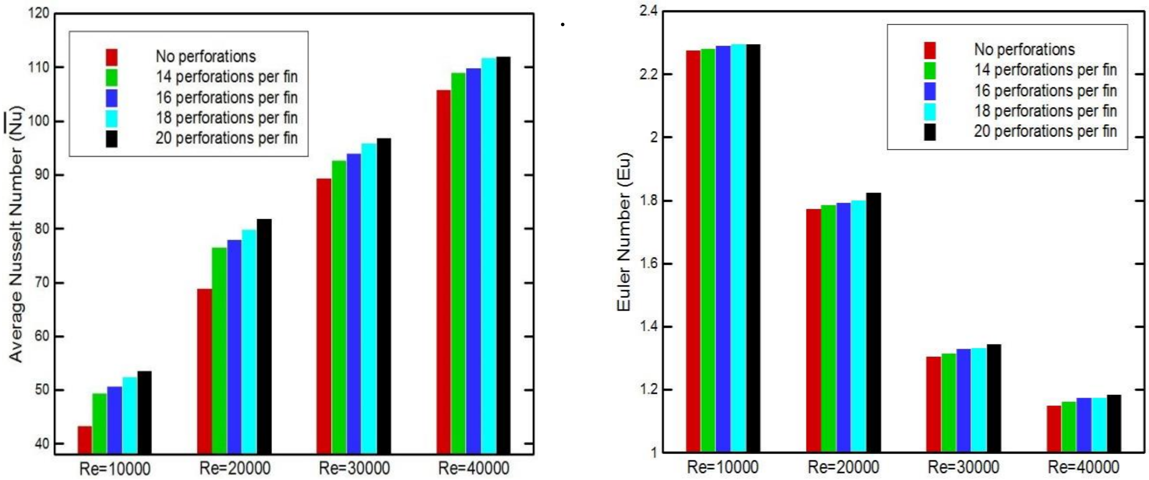
PCFTHE with 14 perforations per fin and maximum increase in Euler Number is 3% at *Re*=40,000 for PCFTHE with 20 perforations per fin.

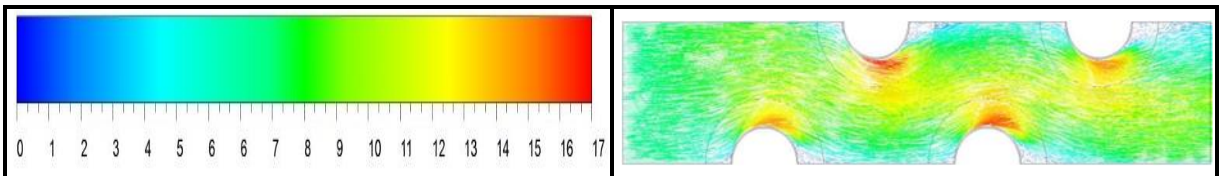


(a) (b)

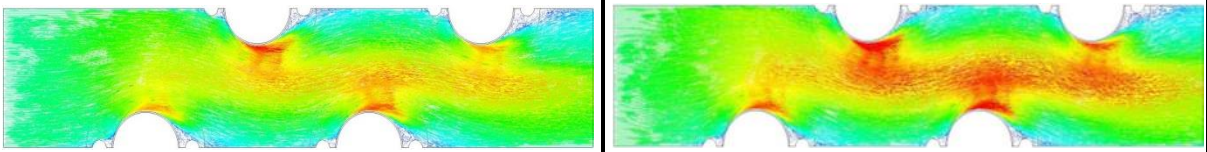
Figure 4. Comparison of (a) Average Nusselt Number (b) Euler Number; for solid fin and fins with different number of perforations at various Reynolds Number.

The increase in Average Nusselt Number on introducing perforations in CFTHEs is because of the flow of localized air jets through the perforations which decreases the size of recirculation zones that form behind solid fins, thus increases heat transfer by increasing shear induced mixing. In addition to this, because of perforations; the flow separation is delayed on surfaces of finned-tubes, thus resulting in the reduction of recirculation region near its surfaces. Both of these phenomena results in the increase in velocity inside the heat exchanger, which can be seen in Figure 5. It can also be concluded that perforations placed before the point of separation does not contribute much to the enhancement of heat transfer, since they do not play any part in the reduction of recirculation region caused by the separation of flow. With increase in the number of perforations per fin, the velocity inside the heat exchanger increases (predominantly between the tubes) which is quite clear from the ‘red region’ in Figure 5. This increase in velocity can be attributed to the increase in intensity of localized air jets flowing through the perforations and further reduction in size of the recirculation zones, thus resulting in increase in the Average Nusselt Number with the number of perforations per fin.

Because of the reduction in recirculation region at the finned tube surfaces, Euler Number for PFTHEs should ideally be less than that for CFTHEs. Although this doesn’t happen because the excessive flow disturbances produced by multiple perforations surpasses the benefit obtained by the reduction in the recirculation region, thus resulting in the increase in Euler Number. The same phenomenon is responsible for the increase in Euler number on increasing the number of perforations per fin.



(a) fin with no perforations



(b) 14 perforations per fin (c) 20 perforations per fin

Figure 5. Contour of velocity vector for PCFTHEs with different number of perforations per fin and d_p = 6mm.

3.3 Reduction in weight obtained by the introduction of perforations in CFTHEs

Table 2. Percentage reduction in weight for PCFTHE as compared to CFTHE for varying number of perforations per fin and d_p = 6mm.

Number of perforations per fin	Percentage reduction in weight per fin
14	28.64%
16	32.72%
18	36.81%
20	40.91%

Along with increase in heat transfer, weight reduction is yet another important advantage of creating perforations in CFTHEs. The maximum and minimum reduction in weight of CFTHEs because of perforations is about 41% and 29% respectively as can be seen in Table 2. As a consequence of weight reduction PCFTHEs are a lot cheaper as compared to CFTHEs

4. Conclusions

The present study investigated the effects of creating perforations on air-side performance of circular finned-tube heat exchangers. The optimum range of number of perforations for d_p=6mm is found to be 14 to 20. The maximum and minimum increase in Average Nusselt Number for d_p = 6mm is 24% and 3% for 20 perforations per fin at *Re*=10,000 and 14 perforations per fin at *Re*=40,000 respectively. This increase in Average Nusselt Number on introducing perforations is due to (i) flow of localized air jets through the perforations, and (ii) reduction in recirculation region caused by the flow separation. As far as the Euler number is concerned, the maximum and minimum increase for d_p=6mm is 3% and 0.19% for 20 perforations per fin at *Re*=40,000 and 14 perforations per fin at *Re*=10,000 respectively. This increase in Euler number is because of the excessive flow disturbances due to perforations. The maximum and minimum reduction in weight of CFTHEs is about 41% and 29% respectively. Further investigations are recommended to study the effect of different perforations shape such as square, triangular etc. on air-side performance of CFTHEs.

Nomenclature

$\bar{h}$	Average heat transfer coefficient, W/m^2-K	n_p	Number of perforations per fin
K_a	Thermal conductivity of air, W/m-K	$\overline{Nu}$	Average Nusselt Number, $\overline{Nu} = \frac{\bar{h}d}{K_a}$

δ	Fin thickness, mm	Eu	Euler Number, $Eu = \frac{\Delta p}{\rho u_{max}^2}$
d_f	Diameter of fin, mm	Re	Reynolds Number, $Re = \frac{u_{max} d}{\nu}$
S_d	Diagonal tube pitch, mm	Pr	Prandtl Number
X,Y,Z	Cartesian coordinates, m	**Greek letters**	
h_f	Fin height, mm	ν	Kinematic viscosity, m^2/s
u_{max}	Maximum velocity of air in fluid domain, m/s	P	Density, kg/m^3
p_{in}	Area weighted average of pressure at inlet of fluid domain, Pa	s	Fin spacing, mm
p_{out}	Area weighted average of pressure at outlet of fluid domain, Pa	μ	Dynamic viscosity of air, kg/m-s
Δp	Pressure drop in the heat exchanger ($\Delta p = p_{in} - p_{out}$), Pa	**Abbreviations**	
n	Number of tube rows in the direction of flow	CFTHEs	Circular finned-tube heat exchangers
d_p	Diameter of perforations, mm	PCFTHEs	Perforated circular finned-tube heat exchangers

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